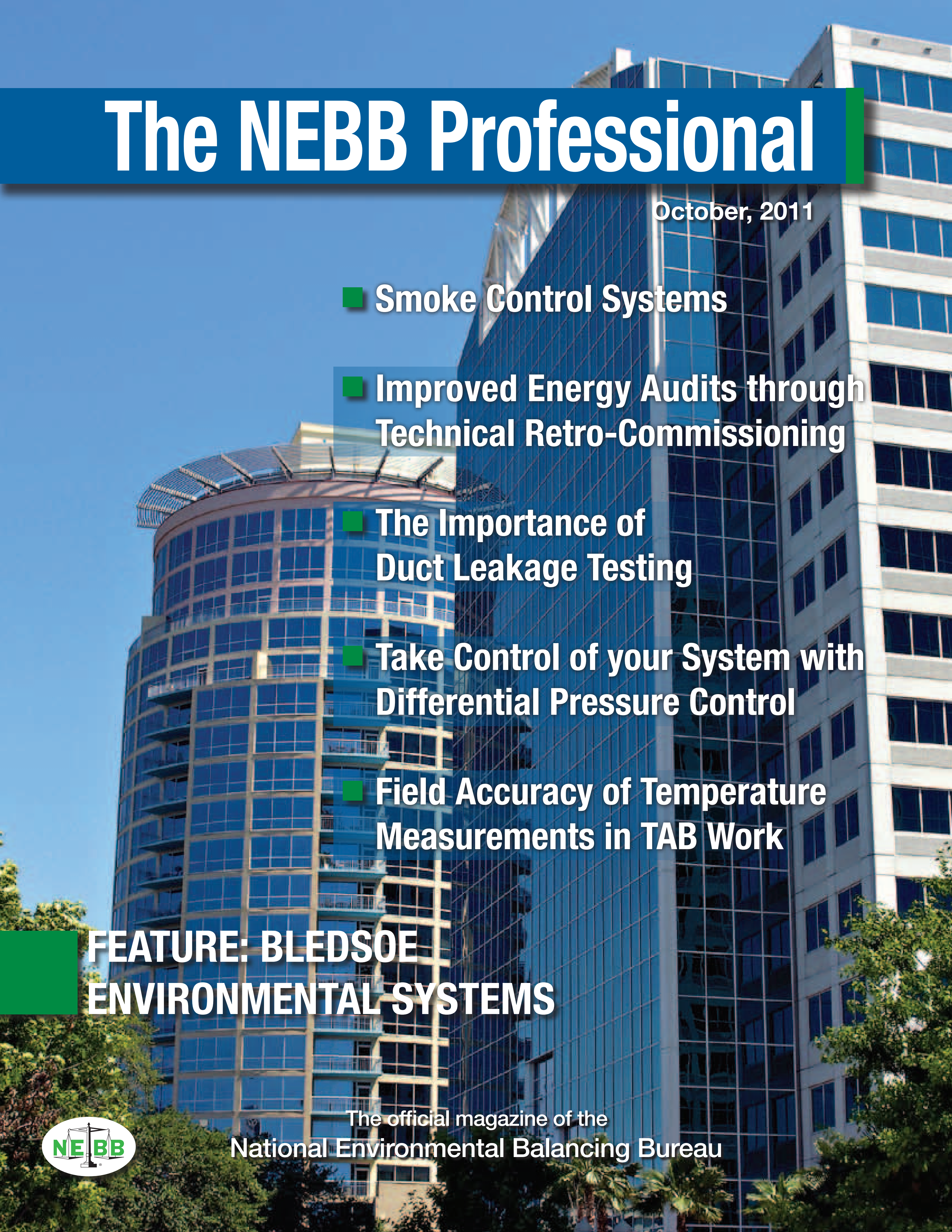

The NEBB Professional

October, 2011

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FEATURE: BLEDSOE ENVIRONMENTAL SYSTEMS

The official magazine of the
National Environmental Balancing Bureau

Letter from the NEBB President

As I write this note, I'm reflecting on this year as the NEBB President. We've accomplished a lot in 2011. Our membership has grown and so has our industry eminence as the leading international association of certified firms for testing, balancing, commissioning of building systems and cleanroom certification.

I would like to take this opportunity to thank everybody involved in supporting NEBB. A special thanks to the Executive Finance Committee, the Board of Directors, the Committee Chairman and all of the committee members for each of our disciplines, past presidents, chapter coordinators, vendors and members who participate in our meetings and keep this organization relevant. I would particularly thank our national staff for all of the work that they do for all of us on a daily basis. I truly value each and every one of you for your dedication and support.

My involvement with NEBB began in 1998 and it's been a truly rewarding journey. I would urge all members to explore opportunities to become more involved, as from my experience, you definitely get more out than you put in, whether its networking with specialists and peers in the industry, enhancing your knowledge about current issues, trends, and technologies or building lasting relationships with fellow members.

During the year I, along with the Board of Directors, had meaningful meetings with representatives from MCAA, ASHRAE, and RSES to increase NEBB's standing as the leading organization for each of our disciplines. I look forward to discussing how each of these relationships can add value to you and your business during the Town Hall meetings at the annual conference.

The 2011 NEBB Annual Conference is just around the corner. This year's conference has a great line-up of speakers and topics and promises to be our best conference yet.

I look forward to welcoming you at the 2011 NEBB Annual Conference!

Bill Neudorfer
President

Index of Articles

Competing Smoke Control Systems: A Case Study in Elevator Shaft Pressurization with Stair Pressurization

By: Daniel Picciano, PE (MN), CFPS

The Appleton Medical Center new Bed Tower included an addition to an existing building located in Appleton, Wisconsin. The design of the project occurred primarily in 2009, with construction being completed in the summer of 2010. As such, the applicable building code for the project was the 2008 Wisconsin Commercial Building Code (WCBC), which is based on the 2006 International Building Code (IBC). Further, as a health care occupancy, the Authority Having Jurisdiction (AHJ) also enforced the 2000 edition of NFPA 101 *Life Safety Code*.

The addition consisted of a new Bed Tower that is nine stories in height, not including a partial basement. The construction of the Bed Tower is noncombustible, fire-rated—Type I-B. The Appleton Medical Center proper is used as a hospital and medical office building, including emergency rooms, surgery wings, cancer center, etc.; and, the Bed Tower addition consists primarily of patient rooms. As such, the primary occupancy in the Bed Tower is Group I-2 (institutional). The main building and addition are provided with automatic fire sprinkler protection throughout.

The elevation of the top-most occupied floor of the Bed Tower exceeds 75 feet above the lowest level of Fire Department vehicle access; and, as such, the Bed Tower is classified as a "high-rise" building. Prescriptively, the IBC requires high-rise buildings to be provided with additional protection for egress stairs and elevator shafts against the migration of smoke vertically throughout the building under a fire condition. The IBC allows various options for providing the necessary protection of egress stairs and elevator shafts, including both passive and active means that are capable of satisfying the intent.

For taller egress stairways, "smokeproof enclosures" are required. Traditional methods of providing smokeproof enclosures are fairly well-established in the construction industry and have been utilized over the years, via both passive and active means. Although the code allows for passive means of providing smokeproof enclosures, active systems are often more preferable from an architectural standpoint. For example, exterior vestibules – an available option for passively providing smokeproof enclosures,– is likely infeasible for egress stairs that are interior to the building footprint. As such, it is very common for egress stairways to be provided with mechanical stair pressurization as a means of complying with the smokeproof enclosure requirements of the IBC.

Elevator shafts, on the other hand, have historically been protected using passive means – such as enclosed elevator lobbies, additional (UL listed) doors, etc. Enclosed elevator lobbies tend to be the most cost-effective solution, even though they are oftentimes not desirable. With the 2006 revision to the IBC, however, an additional option was included to allow mechanical pressurization of the elevator shaft in lieu of providing enclosed elevator lobbies. For the first time, elevator shafts were permitted to be protected with active systems, similar (in concept) to the stair pressurization systems commonly utilized in egress stairs. Unlike stair pressurization, however, elevator shaft pressurization systems are relatively new to the design and construction industry; and, as such, present unique challenges.

In the Bed Tower, the architectural desire was to achieve protection of the two egress stairs that served the entire Bed Tower via mechanical stair pressurization; and, similarly, provide elevator shaft pressurization in one of the two elevator shafts. Given that both the stair pressurization systems and the elevator shaft pressurization system would activate simultaneously under certain fire scenarios, standardized

Image courtesy of The Boldt Company

calculation methods were unavailable to provide design guidance and assure that the performance criteria of each pressurization system could be satisfied during simultaneous operation. Although the specified performance criteria for stair pressurization and elevator shaft pressurization differ in the IBC, both sets of criteria must be satisfied during simultaneous operation of the pressurization systems. So-called "competing" smoke control systems often require computer simulation modeling in order to estimate the design capacities and parameters for each pressurization system, since the available empirical equations typically apply to stair pressurization or elevator shaft pressurization – separately.

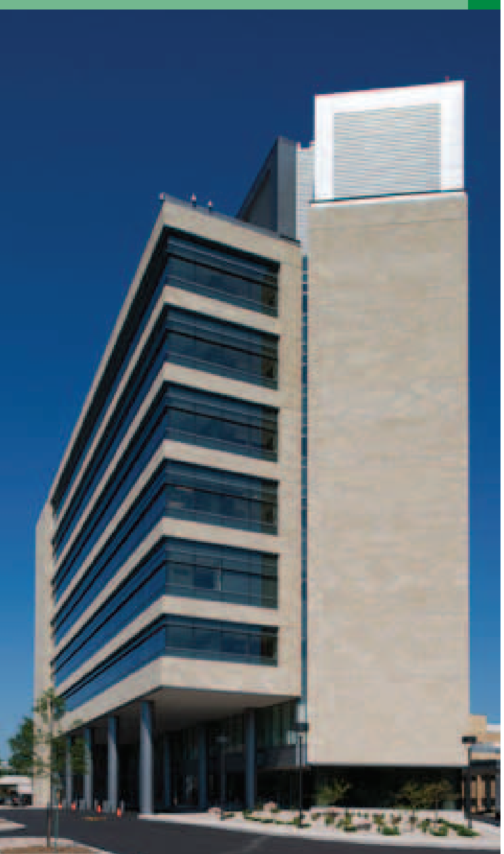


Image courtesy of The Boldt Company

Summit Fire Consulting utilized CONTAM for modeling the expected airflows in the Bed Tower under smoke control conditions. CONTAM is a computer simulation program that is available from the National Institute of Standards and Technologies (NIST) that was originally developed as a multi-zone model to analyze airflow, contaminant transport, personal exposure, etc., for building ventilation systems. The program has been adapted for applications involving pressurization smoke control systems, which rely on maintaining airflow criteria at distinct points in the building.

Data was input into CONTAM to create a computer model of the building. Input included the following: Stairwell enclosure and door locations, elevator shaft and door locations, wall areas, floor areas, roof areas, leakage areas and factors, shaft locations, building temperatures, expected exterior temperatures, and exterior door locations. In addition, conceptual wall routing and locations were input into the model, where such walls were expected to have an impact on the expected airflows – such as smoke barriers, separations from the existing hospital, etc. The primary intent of inputting the routing and locations of interior walls was to determine which areas (i.e., "zones") of the building communicate with each other for the purposes of leakage between zones. In this fashion, the airflow in the building is modeled from zone-to-zone, both horizontally and vertically.

Through a series of simulations, Summit Fire Consulting adjusted fan sizes, injection points, and other mechanical design features (such as relief vents) in order to estimate ranges of expected capacities that would accommodate the various design criteria of the "competing" smoke control systems. Such simulations included a wide range of exterior building temperatures, as well, in order to estimate their effect on the operation of the pressurization systems due to potential "stack effect" conditions in the stairwells and elevator shaft. In this fashion, the design of the pressurization systems incorporated the expected variations in exterior environmental conditions.

The design guidance developed for the pressurization systems was communicated to the mechanical engineer, the project team, and the AHJ for the project via a design report. The design report documented, in detail, the rational analysis conducted, proposed design criteria, code background and applicable navigation, etc., for the purposes of final approval by the AHJ and incorporation into the Construction Documents for the project. In addition, given that both the WCBC and NFPA 101 applied to the project, the Design Report documented the means of complying with both codes – and/or the intent of both codes where prescriptive compliance was not possible. Although the stair pressurization and elevator shaft pressurization systems could not satisfy all of the prescriptive requirements of the applicable sections of the WCBC and NFPA 101, the proposed design satisfied the overall level of fire-and life-safety that is intended by both the WCBC and NFPA 101.

Special inspection of the stair pressurization systems and elevator shaft pressurization systems are also required by the IBC. Such tests and inspections are to be carried out by a qualified agency, and be sufficient to "verify the proper commissioning of the smoke control design in its final installed condition." In addition to the design consulting previously provided, Summit Fire Consulting was also selected as "Special Inspector" for the stair pressurization systems and elevator shaft pressurization system in the Bed Tower.

The special inspection generally occurs over the course of the construction and installation process, and is recommended to include three primary phases: documentation review, equipment inspections, and sequence testing. Each phase of the Special inspection is utilized to confirm specific design and installation requirements for the pressurization systems that are detailed in Section 909 of the IBC and the Construction documents. Documentation review includes the review of pertinent shop drawings and product submittals to confirm that certain equipment requirements – such as listings for mechanical or fire alarm equipment – are satisfied. Equipment inspections, on the other hand, include actual field observations at key milestones during the course of construction to confirm that the installed equipment corresponds to the shop drawings and product submittals, as well as additional equipment requirements specified by the IBC – such as wiring installation requirements, pressure testing of ductwork, etc. Finally, sequence testing occurs near the completion of construction and involves physical testing of the activation features for the pressurization systems and airflow measurements.

All three phases of the special inspection were conducted over a six-month period of time, concluding with final sequence testing in the summer of 2010. Sequence testing involved multiple "pretests," during which the project team identified discrepancies between the installed condition and the approved design—in order to make any necessary modifications to ready the pressurization systems for a final demonstration with the AHJ. The scope of the final demonstration was ultimately at the discretion of the AHJ and involved limited sequence testing and airflow measurements. Ultimately, adequate performance of the pressurization systems was observed during the final demonstrations; and, upon completion and issuance of a special inspection report, a Certificate of Occupancy was issued for the new Bed Tower.

Given the relatively limited implementation of elevator shaft pressurization systems, compounded by the complexity of utilizing "competing" smoke control systems, the project team encountered many unique challenges throughout the design and construction process. For example, the CONTAM model idealized the Bed Tower as essentially a separate, isolated building. In reality, however, the separation of the Bed Tower from the existing Appleton Medical Center proper was not complete – as far as airflow and communication between the spaces is concerned.

Additionally, due to the nature of health care occupancies and the use of the Bed Tower, the integration of the fire alarm system with normal building tempering involved different sequences for different levels of the building as well as different areas within a single level of the building due to the compartmentalization of individual levels of the Bed Tower with smoke barriers. From a smoke control perspective, however, the desire was to *minimize* the number of possible activation sequences and "airflow" conditions under which the pressurization systems would operate. The desired sequencing in the Bed Tower created multiple "airflow" conditions which introduced factors of uncertainty into the design, as well as cause for additional testing to be conducted during the special inspection.

Finally, perhaps the greatest challenge in the design of the pressurization systems (and, thus, the subsequent balancing of the systems) was in the estimation of leakage areas and leakage factors for the building. For input into CONTAM, an approximation of the actual amount of leakage in the building construction is required. Such leakage is inherent to all building construction, to some extent, and can play a significant role in the sizing of fans utilized for pressurization systems. While some data is available in fire protection engineering literature for approximating expected building leakage, the available data is very limited and are only approximations for leakage based on the type of component (i.e., interior wall, exterior wall, roof assembly, floor assembly, etc.) and the qualitative type of construction (i.e., loose, tight, etc.). Slight errors in estimating the leakage expected for the building can cause significant errors in the actual airflow requirements to achieve the performance criteria of pressurization systems.

In the end, through mechanical balancing and sealing of visible leakage points, adequate airflow was observed during testing. In addition, simultaneous operation of the stair pressurization systems with the elevator shaft pressurization system was successfully tested and confirmed under multiple activation sequences and "airflow" conditions in the Bed Tower.

Even as soon as the 2009 edition of the IBC, modifications and revisions to the performance criteria for elevator shaft pressurization systems have been implemented, undoubtedly due to the growing pool of experience with respect to implementation of elevator shaft pressurization systems in building construction – transitioning from theory to reality. In a realm where active fire protection and life safety systems are becoming ever more prevalent, there is reason to anticipate that elevator shaft pressurization will continue to be refined in building construction as a reliable means of protecting elevator shafts. In the meantime, projects such as the new Bed Tower at the Appleton Medical Center will continue to pioneer the design and installation of such systems. ■

Daniel Picciano, PE, CFPS, is with Summit Fire Consulting in St. Paul, MN. He has worked in code consulting and smoke management design/inspection capacities on multiple notable projects in the upper Midwest. Daniel is licensed in the State of Minnesota in Fire Protection Engineering and can be reached at 651-251-1872 or dpicciano@summitfire.com.

Improved Energy Audits through Technical Retro-Commissioning

Dave McFarlane | Technical Commissioning, Inc.

Energy audit reports can be significantly improved when the Technical Retro-Commissioning (TRCx) process is used to obtain an in depth understanding of the operation of a building.

TRCx is the systematic process where a hands on approach is used to operate building systems in order to learn how systems are actually performing. The TRCx process corrects ventilation and energy related problems in existing buildings and is also used to obtain actual operating data for audit savings calculations.

There seem to be two approaches in auditing that are commonly used. These are Process Audits and Technical RCx Audits. The process RCx approach relies on the written work of others and includes review of the Test and Balance reports, the temperature control sequences shown in the contract documents, a review of the plans and specifications and a brief walk through of the facility to determine corrective action. These types of audits are called a Level I audit by the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE).

Audits of this type usually have simple corrective actions that in many cases can be quite predictable. The typical recommendations for Facility Improvement Measures (FIM's) have included the following predictable items:

- Changing out T-12 lighting for more efficient lighting
- Installing occupancy sensors to turn off lights when not in use
- Installing variable frequency drives on constant volume equipment
- Turning off equipment that is not in use
- Lowering temperatures in the winter and increasing temperatures in the summer
- Install weather stripping at doors
- Changing out single pane windows for double pane windows
- Adding insulation to the roof

We have observed boiler plate energy audits from some firms where the same format and wording is used with keywords being changed for each building.

Typical of Level I audits, implementation costs and projected savings are not calculated. These types of audits are less expensive and can produce savings in the 10% range.

The more accurate technical RCx audit relies on the RCX's team to determine the entire building operation. ASHRAE refers to these more stringent audits as level II or III audits. In the technical Level II audit, the RCx audit team checks the entire building and all of the energy using components. These audits will focus on the electrical and lighting system, building envelop and the entire HVAC operations.

The level III audit adds the component of energy modeling to determine savings outlined in the Level II report. Electrical systems are verified for proper grounding. Thermal imaging is used to determine loose connections on all electrical panels, disconnects and motor connections. Power quality analysis is used to determine low voltage, low power factor or phase load imbalance on the electrical systems. A small 10 % reduction in air or water flow will reduce the horsepower of the fan or pump by 33%. Revised motor horsepower's are calculated and smaller more efficient motors are installed where practical.

Lighting systems are evaluated for efficient use of ballasts, lamps, occupancy control, and daylight harvesting. A Level I audit may simply change all T-12 bulbs to T-8 or lower, The more technically advanced Technical Retro Cx Audit checks for lighting levels, modifies these levels to meet code or occupant required lighting levels and then changes the lamps to more efficient ballasts. A Level I audit may simply change the existing lamps but leave the space over or under lit and in the same condition as found.

The building envelope is evaluated for air infiltration, moisture intrusion, building pressurization, and glazing efficiencies. Where the Level I audit may look for cracks at doors and leaks at windows to determine areas of infiltration. The thorough TRCx audit will perform building pressurization tests on the envelop using blower door testing kits or using the actual air handling units themselves to pressurize the building in order to determine leakage.

In this test, the building is pressurized or evacuated to two or three different levels of pressure between 0.1 and 0.3 inches of water. The air infiltration or exfiltration at these pressures is measured to determine the actual air leakage. A statistical analysis of the data is completed to determine the air flow leakage in CFM leakage per square foot of building surface at 0.3 inches of water pressure.

Various industry specific sets of data are available at this pressure. These show that high performing buildings leaks less than 0.1 CFM per FT2 of building surface. Medium performing building leak in the 0.3 CFM per FT2 range and a poor building exceeds 0.4 CFM air leakage per FT2 of building surface. Thermal imaging of the building surface while under this test pressure shows where leakage is occurring. Once areas of infiltration are shown, proper sealing can be implemented. Leakage points are sealed and retested until an acceptable level of leakage is attained. The composition and construction of the windows is evaluated to determine if the window shading coefficients or window U-values can be improved. The RCX's team determines the current operation of the HVAC systems. The TRCx process looks for areas where air and hydronic flows, temperatures, pressures and run times can be reduced. The TRCx process looks for ways to decrease fan air flows and also remove restrictions that increase the static pressure in duct systems. A 10 % reduction in fan air flow reduces total horsepower by 33%. Reducing static pressure by removing restrictions or by simply lowering the static pressure set points reduces energy. A 10% reduction in static pressure at constant air flow reduces horsepower by 11%.

Many design engineers use the ASHRAE 99.4% weather data and then add safety factors of 10% or more to determine the design flows and equipment for the project. While this may be an acceptable approach to ensure the project has sufficient capacity, actual operating flows can be reduced. This reduction is completed by analyzing the actual heating and cooling load in buildings at the 98% ASHRAE weather data and then removing the safety factors from the various air and water flows. This exercise usually results in a significant reduction in the maximum flow requirements. It is important to reset the VAV terminal units to these lower values. Leaking VAV reheat valves or arbitrarily high Air Handling Unit (AHU) discharge temperatures will cause the VAV box to revert to the higher position if the boxes are not reset to the new revised conditions. Minimum position settings can usually be reduced and still maintain acceptable indoor air quality.

Toilet exhaust systems are an area that are often ventilated well above code required minimums. Savings can be generated by slowing the exhaust to code required minimums and shutting the fans off when the space is not occupied. Ducts that are not reinforced to the proper SMACNA standards can collapse. Collapsed duct impart a restriction on the duct system and usually leak at broken seams and joints. Open end caps, open duct collars, duct work that is collapsed or has split seams and unsealed ductwork provides opportunities to lower the fan speeds and reduce airflow once the problems areas are repaired. Restrictive duct fittings, elbows without turning vanes, elbows with debris on turning vanes and choked fittings all impose unneeded restrictions that increase static pressure, horsepower and energy usage.

In order to check out the duct and hydronic systems, ceiling tiles are removed to enable the entire duct system to be reviewed for proper sealing, duct reinforcement, and fitting construction. The inspection looks for missing end caps, open duct collars, collapsed ducts, split seems and major leak points. Systems are analyzed for proper zoning. Mixing interior and exterior spaces on the same terminal box causes both comfort issues and increased energy usage. The TRCx process finds and corrects leakage, restrictive fittings and zoning issues. The actual air, water flows, temperatures and pressures are measured. The actual water and air flows are compared to the new 98% ASHRAE design requirements. The percentage reduction in flows gives the information necessary to determine energy savings from flow reductions. Actual temperature measurements of air and hydronic heating and cooling temperatures will outline the potential for energy reductions. Heating systems can usually be reset between the temperature required at design heating days and the lower hot water temperatures that can be used during warmer outside conditions. This lower hot water temperature is a function of the type of boiler and the capacity of the heating coil.

A typical hot water reset schedule in northern climates may reset the water temperature from 180F at 0F to 140F or lower at 50F. A similar reset schedule for cooling would provide 45F chilled water at design conditions but then increase the chilled water temperature to 50F or higher depending on the cooling coils ability to provide properly dehumidified air at cooler outdoor conditions. Air Handling Unit discharge temperatures can be reset based on outdoor design conditions or the number of zones calling for heating or cooling. Typical outdoor reset schedules adjusting the mixed air or discharge temperature would have a discharge temperature of 55F at 65F OA and a 65F temperature at 0F OA. This reset is dependent on the ability of the interior spaces terminal boxes to handle the cooling load and the dehumidification capabilities coiling coils and space humidity requirements. The increased discharge temperature reduces the amount of reheat required.

After the restrictions outlined in the above paragraphs have been removed and the actual duct system pressure is known, the duct discharge pressure can usually be lowered until the box requiring the greatest static inlet pressure is controlling at the 90-95% open range. This pressure is then set as the maximum static pressure required for the air handler and is typically found in the summer. This pressure can usually be reset to a lower static pressure when boxes go to the heating mode in the winter. In order to set the lower static pressure limit, all boxes that require heating are set to a call for heat. The static pressure required for the most restrictive box to meet the reduced heating air flow becomes the lower duct static pressure set point. This duct discharge pressure reset is also verified by the number of boxes calling for cooling. Duct Static pressure is reset between the high and the low setting based on the number of VAV dampers at 100% open.

Many control sequences will specify that the hydronic pumping systems have a pipe differential pressure set at 6-10 PSIG. In many cases this is excessive. Similar to air systems, hydronic systems are set up so that the most restrictive coil valve is controlling at the 90-95% open position at the lowest required static pressure. Lowering system pressures reduces energy costs. The technical RCX team also verifies the temperature control operation of the system and sequences. Actual run times, control response times and trends of all loops are recorded for two weeks to one month to confirm normal system operation before changes are made.

One of the easiest changes that can be made to provide immediate savings is to compare the run times of equipment to the actual occupancy of the building. Systems do not need to run for 2-3 hours after the building is closed. Fresh air dampers do not need to be open when exhaust systems are shut off in the unoccupied mode. Turning boilers, chillers, pumps, fans and air handlers off when the building is unoccupied and shutting outside air and relief air dampers saves energy. The team compares the sequences that are in operation to known energy efficient control sequences. Occupied and unoccupied times are compared to the actual building occupancy. Building pressurization sequences are reviewed to determine that the building is adequately pressurized to the +0.02 to 0.05 inch of pressure. Return fan tracking sequences used to control building pressurization are notoriously poor control sequences for maintaining proper building pressurization. In cold climates, buildings under a negative pressure will be drafty due to window leaks or leaks at the floor-wall and roof -wall joints. In colder climates, occupants will compensate for these drafts by placing electric plug in heaters at their feet. These plug loads increase electric energy.

The current control sequences are compared to the reset schedules outlined above. The energy savings seen in these pressure and temperature reductions can be accurately calculated. Perhaps most importantly the technical RCx team physically verifies each HAVC system control loop for proper operation. This ensures that all dampers, valves, VFD's and other equipment opens and closes 100%, controls to set point and responds to changes in an acceptable amount of time. Valves that do not close 100% leak either hot or cold water into the coils. This leakage is an energy waster due to extra pumping energy and additional reheat if the cooling coil leaks or additional air flow when the heating coil valves leak. Dampers that do not open fully create a restriction that wastes energy. Dampers that do not shut fully allow hot or cold outside air into the space in the unoccupied mode. Technical RCx Auditing is a hands on approach to building systems evaluation conducted by professionals skilled in the science of engineering, design and building operation. Our firm has a good track record of producing energy savings in the 30+% range by tuning systems reducing pressures, temperatures and run times. ■

The Importance of Duct Leakage Testing

Wade Sand | TSI Incorporated

Air duct leakage should be a concern to everyone that touches a building or home, from the design engineer, architect, and installer to the commissioning agent, building owner, and occupants. An Inefficient and ineffective duct work system for supply and return air is not sustainable and will not perform to its design. Instead, it will increase costs and energy usage, decrease system performance and create indoor air quality issues.

Leaks in forced air duct systems are recognized as a source of energy waste in both new and existing buildings and homes. Studies indicate that duct leakage can account for as much as 25% of the total energy loss, and it may have a greater impact on energy use than air infiltration through gaps or leaks in the building envelope.

Leaky ducts can increase air conditioning and heating bills, reduce equipment capacity and performance, and may result in indoor air quality (IAQ) problems. While the consequence of leaky ducts can be found in many areas, some of the most common outcomes include:

- Building depressurization from duct leaks and imbalanced duct systems can cause combustion products to vent back into the building.
- Supply duct leakage causes conditioned air to be dumped into the crawlspace, open plenums, and ceilings instead of distributed back to through the system.
- Return duck leakage pulls outside air into the duct system reducing efficiency and capacity.
- In humid climates, moist air being drawn into return leaks can and will reduce the dehumidification capacity of the HVAC system and create a clammy environment indoors.
- Leaks in the return ductwork draw air into the building from the ceilings, open plenums, crawlspaces and bring with it dust, mold spores, insulation fibers and other contaminants.

Duct leakage testing involves pressurizing the duct system with a calibrated fan and measuring the air flow through the fan and its effect on pressurizing the duct system. A tighter duct system means that less air is required from the fan to create sufficient duct system pressure. Duct leakage measurements are used to diagnose and demonstrate leakage problems, estimate efficiency losses from duct leakage, and certify the quality of duct system installation.

Maintaining an efficient and effective duct system is imperative, as a building’s HVAC systems can contribute to nearly 40% of a buildings annual energy usage. Even counteractive, energy-efficient solutions, such as high efficiency equipment and the latest design techniques, are futile if the core duct system is not operating effectively. Testing and verifying the duct system is a very important aspect in the buildings efficiency in order to ensure system optimization. ■

Take Control of your System with Differential Pressure Control

Bhumika Lathia | *Tour & Andersson, North America*

The ultimate goal of every heating and cooling installation is to provide the desired indoor climate at the lowest possible energy cost.

Today's advanced control technology means that, in theory, achieving this goal is possible. In practice however, even the most sophisticated controllers don't always perform as promised. The result is lower levels of comfort at a higher level of energy expenditure. Only through balancing can potential problems be identified and rectified.

The reason for these problems is often that the controllers are not being allowed to do their job. Indeed they can only do their job as specified if the three key conditions for hydronic control are fulfilled.

These key conditions are:

- The design flow must be available at all terminals.
- The differential pressure across control valves must not vary too much.
- Flows must be compatible at system interfaces.

The best way to attain these three conditions is to perform a balancing procedure. The reason being that balancing ensures that the plant actually performs as specified by the designer and operates the way the designer intended.

Balancing reveals and removes a number of threats to the functionality of the installation. Threats which can range from incorrectly implemented balancing calculations to assembly errors like incorrectly installed check valves, and blocked filters.

The balancing procedure means you can immediately reveal the effects of any disturbances; identify the reason for them and take corrective measures. In simple terms, carrying out a balancing procedure gives the plant the optimum capability to provide the desired indoor climate at the lowest possible energy cost.

Variable flow systems are becoming more and more popular, mainly due to the following advantages compared to constant flow systems:

- The pumping costs are reduced
- The return temperature is optimized in heating/cooling systems

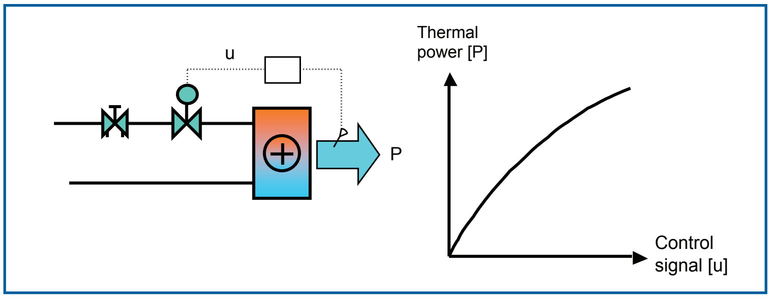
But there is one major disadvantage to variable flow systems: The differential pressures in the plant also may vary considerably during the operation. Typical symptoms due to differential pressure variations are:

- Continuous oscillation of room temperature
- Room temperatures not reaching the required set points at low loads
- Maintenance problems with control valves and actuators, due to fatigue from hunting
- Higher energy costs than expected due to unfavorable control settings to avoid instability

Sophisticated controllers cannot achieve their theoretical performance unless the conditions for their operation are correct. These conditions are governed by the design of the hydronic system. In simple words, control cannot compensate for poorly designed systems, which is why system must be designed as controllable as possible.

Circuit characteristic

One important measure of hydronic design quality is the circuit characteristic. The figure below represents a typical hydronic circuit for an air heating/cooling coil. The circuit characteristic is the relationship between the control signal and the resulting thermal power from the coil. It determines the controllability of the system.

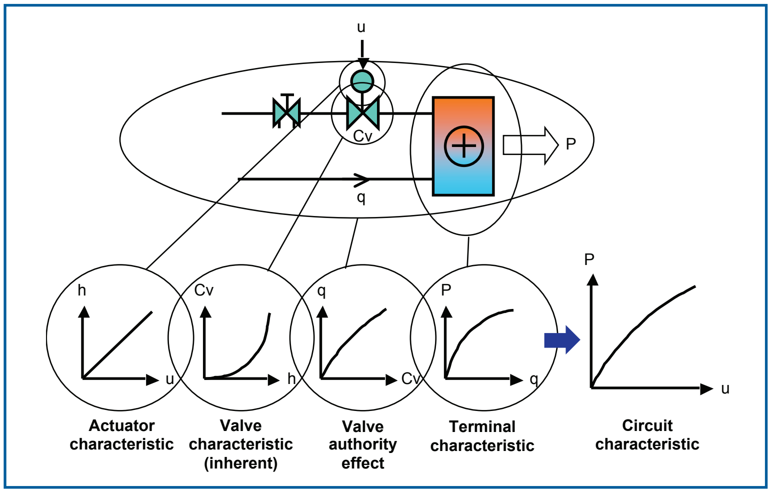


The steeper the slope of the circuit characteristic curve, the higher is the risk for control instability and as a result, the control becomes more difficult. As can be seen in the graph above, at a low slope, any actions from the control will result in marginal changes of the thermal output, making the system quite indifferent. However, at a high slope, even tiny alterations of the control signal will result in large changes of the thermal output, making the system sensitive and possibly unstable. To avoid instability problems, which effectively will ruin the control function, a low set value of the gain (corresponding to a wide P-band) is required in the controller. However, the result of a low controller gain is less control accuracy and slower response to disturbances. It is therefore crucial to avoid steep slopes of the circuit characteristic. In such perspective, the ambition should be to achieve a linear circuit characteristic since it will minimise the slope across the entire control range.

Circuit characteristic compound

The circuit characteristic consist of:

- Actuator characteristic
- Inherent valve characteristic
- Terminal characteristic
- Valve authority



Symbol Legend: u – Incoming control signal from controller to actuator
h – valve lift/valve opening
Cv – valve capacity
q – water flow
P – thermal power
u – control signal.

The actuator characteristic shows the relationship between the incoming control signal (u) from the controller to the actuator and the resulting valve lift (h). Usually, the characteristic is linear but for advanced actuators, the characteristic curve may be quite nonlinear.

The inherent valve characteristic, which shows the relation between valve opening and valve capacity (Cv value), depends solely on the mechanical design of the control valve. There are few different types of valve characteristics on the market; the most common ones are the *linear* and the *equal-percentage*, or actually *equal-percentage modified* (EQM) characteristics.

The terminal characteristic may vary a lot depending on design, size and temperatures but is definitely nonlinear. A typical characteristic gives 50% power at 20% flow and 80% power at 50% flow, quite the opposite shape compared to an EQM valve characteristic. This is also the reason why EQM is usually preferred, when choosing control valve, since it counteracts the nonlinearity of the terminal.

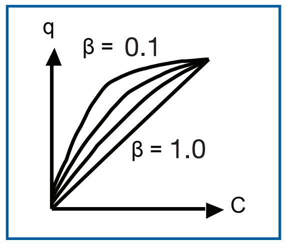
Valve authority is a measure of the change in differential pressure across a control valve during operation. The flow through a control valve depends on the differential pressure across the valve and its Cv value. The Cv value is given by the inherent valve characteristic for any valve opening. If the differential pressure is constant during operation, the relationship between Cv and water flow would be completely linear. However, in variable flow system, the differential pressure varies during operation, which means that the relationship becomes more or less nonlinear. The "magnitude" of the nonlinearity is expressed by the valve authority:

$$\boxed{\beta = \frac{\Delta p\, V_{design}}{\Delta p\, V_{shut}}}$$

β = *Valve authority [-]*

$\Delta p\, V_{design}$ = *Differential pressure across fully open control valve at design flow [psi]*

$\Delta p\, V_{shut}$ = *Differential pressure across fully shut control valve [psi]*



A high value of valve authority means that the differential pressure is close to constant and the relationship between Cv value and water flow becomes quite linear. A low value, on the other hand, means that the differential pressure will increase a lot when the valve closes, resulting in large nonlinearity between Cv value and flow. The lower the valve authority is, the more nonlinear the curve becomes.

Simply by looking at the compound of the circuit characteristic, it is quite clear that a low valve authority will make the circuit characteristic curve unfavourable. This is why the second condition must be fulfilled; too much variation in differential pressure across a control valve leads to low authority, distorted circuit characteristic and poor control. In addition, large variations in differential pressure will lead to interactivity between circuits, making control even more difficult.

Design and minimum valve authority

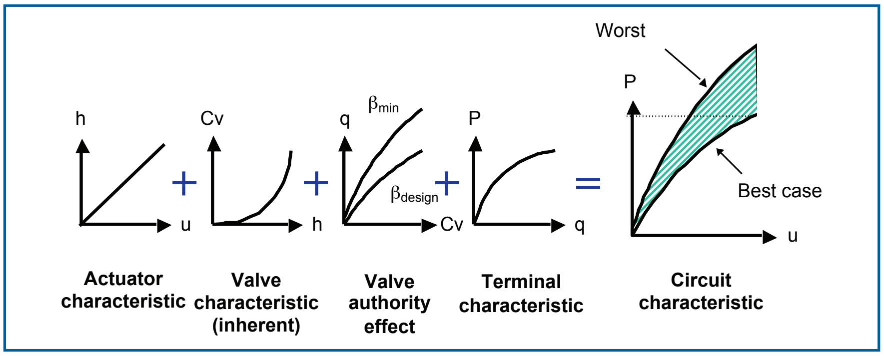
The available differential pressure across the hydronic circuit is transferred to the control valve once it shuts, which means that size, design and control of the system determines the differential pressure across the control valve fully shut (denominator in the expression above). Hence, the circumstance of the system at any given time determines the available differential pressure across the circuits, which means that the valve authority varies during operation. If, for instance, only one control valve shuts in a system while the others are fully open, the differential pressure across that specific valve will become significantly lower than if all control valves shut at the same time. This leads to two different definitions of valve authority; *design authority* and *minimum authority*. For two-way control valves in variable flow systems, these definitions are as follows:

$$\beta_{design} = \frac{\Delta p\, V_{design}}{\Delta H_{design}} \qquad \beta_{min} = \frac{\Delta p\, V_{design}}{\Delta H_{max}}$$

β_{design} = *Valve authority at design condition [-]*
ΔH_{design} = *Available differential pressure across circuit at design condition [psi]*
$\Delta p\, V_{design}$ = *Differential pressure across fully open control valve at design flow [psi]*
β_{min} = *Minimum valve authority [-]*
ΔH_{max} = *Maximum differential pressure across circuit during operation [psi]*

Both the design and minimum authority definitions should be regarded when designing a system, since the level of the valve authority will vary somewhere between the design authority (highest possible level) and the minimum authority (lowest possible level) during operation.

The implication of varying valve authority on the circuit characteristic during operation is shown in the figure below. The best case represents the characteristic of the considered circuit for design authority, corresponding to a situation where all other control valves in the system are maintained fully open. The worst case, however, represents the circuit characteristic for minimum authority, corresponding effectively to a situation where all other control valves are maintained fully shut. The latter case results in much higher differential pressure across the circuit and, thus, steeper slope of the circuit characteristic and also substantial overflow when the control valve is fully open.



Hydronic design

The impact of the chosen control valve on the circuit characteristic and, hence, on the controllability of the system, is quite obvious since both the valve characteristic as well as the valve authority depends on the control valve selection. When choosing control valves both of these aspects must be taken care of.

- The design flow must be obtained for the control valve fully open in design conditions.
- In order to facilitate control, the valve characteristic should match the terminal nonlinearity.
- To maintain a favourable circuit characteristic, the valve authority must not be too low.

In order to prevent the valve authority from distorting the circuit characteristic too much, the lowest values of design and minimum authority are:

$$\beta_{design} \geq 0.5$$
$$\beta_{min} \geq 0.25$$

The design authority for a control valve should not be less than 0.5, which effectively means that the design pressure drop through the fully open (two-way) control valve at design flow should at least be equal to half of the available differential pressure across the circuit at design condition. The purpose of this guideline is to make sure that the circuit characteristic at its best becomes quite close to linear, assuming that the inherent characteristic of the valve is appropriately chosen.

The second condition states that the minimum authority should not fall short of 0.25, which sets the lowest level of the circuit characteristic, when it is at its worst. Evidently, this condition is of much importance since it effectively settles the limit for the control to handle.

Besides choosing the control valves carefully, there are other design measures to implement in order to avoid low authority:

- Avoid large pipe pressure drops
- Use variable speed pumps
- Use Δp stabilization valves when needed

Even if the control valve is selected with great care, there might arise situations where the valve authority becomes too low anyway, simply because it depends not solely on the control valve sizing but on the design of the rest of the system as well. An effective measure in such perspective is to install differential pressure stabilization valves. These might improve the prerequisites for problem free control radically.

Conclusions

The objective of any HVAC plant is to provide a comfortable indoor climate whilst minimising costs and operational problems.

In theory, modern control technologies make this objective possible. In practise, however, not even the most sophisticated controllers perform as promised. The reason is often that the conditions necessary for good control are not fulfilled.

One such condition is that the differential pressure across control valves must not vary too much. Valves like differential pressure control valves can be used to maintain almost constant differential pressure and thus maintain constant flow regardless of pressure fluctuations in other parts of the system. Differential pressure control valves can be very useful in modular applications, retrofit applications etc. The reason is to maintain the valve authority at a sufficient level in order to prevent the circuit characteristic from distorting too much and, thus, avoiding control problems. ■

Field Accuracy of Temperature Measurements in TAB Work

Andrew P. Nolfo, PE | Former NEBB Technical Director

The purpose of this article is to discuss the measuring and reporting of temperatures at heat transfer equipment during the Testing Adjusting and Balancing (TAB) effort.

Historical perspective

In 2005, the National Environmental Balancing Bureau (NEBB) published the 7th Edition of the Procedural Standards for Testing Adjusting and Balancing of Environmental Systems. This 7th Edition represented a departure from all previous editions in many ways. The Procedural Standard's use of the ***SHALL, SHOULD*** and ***MAY*** language define the minimum requirements of a project that is to be performed for a NEBB Certified TAB Report. For a report to be certified as a NEBB TAB Report, all of the ***SHALL*** requirements must be included in the report. ***SHALL*** is defined to indicate a mandatory requirement to be followed strictly in order to conform to the standards and procedures and from which no deviation is permitted. ***SHOULD*** is defined to indicate a course of action that is preferred but not necessarily required. ***MAY*** is defined to indicate a course of action that is permissible as determined by the NEBB Certified TAB Firm.

One of the changes in the 7th Edition pertains to the requirement for performing and reporting all temperature measurements at heat transfer equipment such as boilers, chillers, coils, etc as part of the minimum TAB scope of services. Previous editions of the Procedural Standard had always included measurement and reporting of temperatures as an integral part of a NEBB TAB Report. This current edition has made this work a ***MAY*** item in lieu of a ***SHALL*** item.

Contractual issues

Measuring and reporting temperatures ***MAY*** be included at the discretion of the NEBB Certified TAB Firm or it may be required if the design engineer specified that temperature measurements and reporting were part of the TAB scope of services. While the requirements are defined in the NEBB Procedural Standard, as with any contract project, the design engineer of record prepares a specification which normally includes a section covering the TAB work. In a specification, the engineer delineates the contract scope of work for the TAB services. The requirements of the contract documents will always take precedence over the requirements in the NEBB Procedural Standard. This issue is clearly stated in the Procedural Standards. As with any of the ***SHALL, SHOULD*** or ***MAY*** items, the design engineer can dictate the actual requirements for any project. Thus if an engineer requires a project to be NEBB Certified, he is free to specify any and all requirements that may be in complete agreement with the minimum (***SHALL***) NEBB requirements or he is free to completely define his own project requirements including such items as temperature measurements.

Capacity testing requirements

While capacity testing requirements covers the historical and contractual status of the issue, it may be appropriate to discuss the reasoning behind the decision to make this change from previous editions. The main reason is really quite simple and relates to the accuracy of field temperature measurements and the ability to utilize these measurements as an indication of heat transfer capacity

(cont.)...

in the field when comparing field capacity to factory or rated capacity performance.

There are many designers, commissioning providers, owners and operational personnel that are under the misconception that normal TAB data such as fluid flows and fluid temperatures which are identified in a final certified TAB report can be easily used to equate the performance of a heat transfer device in the field to manufacturer's rated capacity performance. Nothing could be further from the truth.

Let's examine how equipment capacity performance is obtained by the manufacturer. The capacity performance of almost all HVAC equipment is rated by some standardized testing procedure. This applies to fans, coils, chillers, boilers, cooling towers, air terminal devices, air devices, control valves, etc. While many of the manufacturers of HVAC equipment and components utilize independent or third party testing agencies, some manufacturers perform capacity testing in their own factory. Under either scenario, the testing procedure is usually performed in accordance with an industry standard such as ASHRAE, ARI, AMCA, Hydronic Boiler Institute, Cooling Tower Institute, etc. Many of the testing protocols are jointly sponsored or co-authored, and many testing protocols are also American National Standards Institute (ANSI) standards.

While the exacting requirements of capacity performance testing vary with the type, size and features of the various pieces of HVAC equipment, there are some common threads that must be understood before we proceed:

- All capacity performance testing is performed in idealized laboratory conditions.
- Testing is performed in strict accordance with either industry accepted protocols or manufacturer's standard practices.
- Instrumentation requirements necessary to measure and report data such as temperature, fluid flow, pressures, electrical characteristics, etc are industrial-grade, laboratory quality digital instrumentation that usually involves data logging of measurements over an extended period of time.
- Capacity performance of heat transfer equipment is determined under steady-state, or quasi steady-state, heat transfer conditions that must exist over an extended period of time.
- Capacity performance testing protocols are utilized to guarantee consistency in the performance of the equipment in the laboratory to control manufacturing tolerances and to be used when comparing performance of similar equipment from various manufacturers. The testing protocols are NOT meant to replicate the performance of the equipment in the field.

While the last bullet is probably the most important, one other item needs to be stated. The first four items are NOT present in the real world conditions of the field. As an example, fan measurements in the field will never plot on a manufacturer's fan performance curve. They were never meant to. Statements in the AMCA publications 201, 203 and 210 clearly address this issue. The same is true of capacity performance ratings of heat transfer equipment.

Accuracy of temperature measurements

All of the agencies that certify TAB firms require a minimum level of instrumentation for various functions. The requirements identify measurement type, appropriate instrumentation, accuracy, range, resolution and the minimum calibration requirements of that instrumentation to be used in field when performing TAB work. While each of the agencies publishes their own procedural standards, TAB work is also addressed in ANSI/ASHRAE Standard 111-2008, *Measurement, Testing, Adjusting, and Balancing of Building HVAC Systems*.

One of the unique features of the ASHRAE document is the method that the standard describes the instrumentation requirements. For many of the instruments, the standard discusses the various standard TAB instruments, the uses, limitations, calibration requirements and the accuracy of field measurements. I repeat, the title of the paragraph for each instrument is: Accuracy of Field Measurements. Instrumentation accuracy is NOT the accuracy of the measured quantity in the field.

A thermometer, a manometer or electrical test meter may be rated by the manufacturer as having a stated accuracy. That is the accuracy of the instrument. Most analog instrument is rated as a percent of full scale or percent of reading. Most digital instrumentation is normally rated by two components: absolute accuracy and range accuracy. A typical digital micromanometer used for measuring airflow velocity pressures (re: velocities) may be rated as ±3% of reading and ±5 feet per minute (fpm). That means that the absolute accuracy will be within 3% of the measured value and the range accuracy will be within an additional 5 fpm. As an example, a measurement of 824 fpm for a duct traverse using this instrument means the actual velocity could be within the range of 795 fpm to 853 fpm. That would equate to an overall accuracy of ±3.5%, which is excellent. Now let's take a digital thermal anemometer with the same rated accuracies measuring airflow velocity at the face of a fume hood. For this example, our measurement is 96 fpm. Using the same values as the previous example, the actual velocity could be within the range of 88 to 104 fpm. That would equate to an overall accuracy of ±8.3%, which may be acceptable. In either example, the values identified are instrument accuracy, not field accuracy.

The ASHRAE standard discusses the accuracy of the instrument to measure a correct value when used by a technician in field conditions. This terminology of accuracy of field measurements means that an average technician, standing on a 10' ladder performing a duct traverse will NOT be the same as the instrument accuracy. It may be the same as the instrument accuracy but it will normally be less (sometimes, much less) than the instrument accuracy. As a rule, the average accuracy of any field measurement will be ±10%. That is why the ASHRAE Standard, the NEBB Procedural Standards, the AABC National Standards and TABB Standards all reference ±10% as the acceptable tolerance on field measurements. It's the standard. Yet we see may designers specifying unobtainable tolerances on TAB work of such as ±5% or -0% - +10%. The "standard" is ± 10%. If a facility design needs tighter tolerances to design criteria, change the design not the standard.

One last issue when taking temperature readings that goes directly to the subject of accuracy is temperature uniformity of the fluid flow. For water systems, the temperature of the fluid is usually thoroughly mixed and of the same uniform temperature as it enters or leaves the heat transfer device. That is certainly not the case with air temperatures. The most common misnomer in the HVAC industry is the term "mixing box" as a standard part of an air handling unit. If the outdoor air enters the "mixing box" at 20°F and the return air stream is at 75°F, it would not be unusual to measure mixed air temperatures anywhere from 20°F to 75°F within the mixing box regardless of the amount of either airstream entering the chamber. Mixing boxes are poor "mixers". Additionally, other items within the air handling unit such as filters, coils and even fans, offer little assistance in causing the airstream to thorough mix into a homogenous, uniform temperature. When these conditions occur it may require that a temperature traverse be performed and a weighted airflow volume/temperature calculation be performed. Unfortunately, performing an airflow traverse inside a mixing box is most unreliable due to the fact that uniform velocity profile seldomly occurs in a section with the type of turbulence encountered in a mixing box. All of these items compound the issue of trying to determine the actual temperature.

Accuracy of capacity calculations from field measured data

Let's examine an issue that would appear to be relatively simple – determining capacity performance from field measurements for a hydronic heating coil at a VAV air terminal unit. The following measurements must be made in order to determine capacity:

- **Volumetric airflow**
- **Entering and leaving air temperatures (dry-bulb)**
- **Hydronic flow**
- **Entering and leaving water temperatures**

While the above listing appears to be pretty simplistic in nature, let's examine each component.

Volumetric airflow

There are several methods to determine airflow at a VAV box. The most accurate would be a traverse. The traverse could be performed at the inlet or the outlet. In order to obtain what ASHRAE and AMCA define as an ideal velocity pressure profile, the duct traverse should in a straight section of duct with at least 2.5 diameters of straight duct. For a 10" VAV box, the traverse should be performed in a straight, hard section of duct at least 25" upstream of the inlet. Let's assume the outlet duct is 12" x 10" which is an equivalent of 12.4". A downstream duct traverse should be performed in a straight section of duct that is 31" long before any fittings, takeoffs, etc. While either of these conditions may be attainable, the accuracy of a traverse at either location would be within ±10% of the actual airflow. A traverse at a location other than ideal would be less accurate.

In addition to a traverse, the inlet static pressure could be used to determine the airflow. The manufacturer of the VAV box provides a pressure/airflow chart directly on the side of the VAV box. The accuracy of this chart is completely dependent on the inlet and outlet condition of the actual ducts (re: System Effect is not just at fans). Remember, this capacity chart is produced under the same rating conditions as all other HVAC equipment; idealized laboratory conditions, long straight run of hard, sealed duct on the inlet and outlet with no flex duct, fittings, takeoffs etc., none of which is representative of the field conditions. In order to equate field performance with factory ratings, the exact same testing parameters must apply. The last point on using the manufacturer's rated information is that we are assuming that the VAV box does not leak.

The outlets could be measured and the values added together to determine the airflow. Of course each outlet measurement has an accuracy of ±10% and we now must take into account any downstream leakage in the low velocity duct, spin-ins, flexible duct and connections to the air devices themselves. Again, the accuracy of the field measurement would be less than the traverse.

Entering and leaving airflow temperatures (dry-bulb)

We previously discussed the inability of mixing sections to properly mix two airstreams in an attempt to create a uniform temperature. As the supply air enters the VAV box, it is safe to assume that temperature uniformity has been attained. We can now take a single temperature of the air. The field accuracy of that temperature using a good digital thermometer should be within ±2%. Now we need to take the temperature of the air after the heating coil. For the sake of discussion, let's assume that the temperature of the air leaving the coil has uniformity and a single measurement can be utilized. Again, the field accuracy of this measurement will be within ±2%.

Hydronic flow

The hydronic flow can be measured in by the following methods:

- A flow meter (venturi, orifice plate, magnetic flow meter)
- A calibrated balancing valve
- An ultrasonic flowmeter
- Using rated equipment pressure drop
- Using a pump curve
- Performing an energy balance

The above list is in order of accuracy from greatest (flow meters) to the lowest (energy balance). Since we are focusing on a hot water heating coil at a VAV box, the most logical would be a calibrated balance valve, and the coil piping may also be provided with pressure and temperature ports at the coil. Since we are focusing on accuracy, we will assume that piping for this heating coil has a calibrated balancing valve on the return line. The type of calibrated balancing valve does not impact accuracy. Adjustable orifice, fixed orifice or self adjusting should all have the same relative degree of accuracy. The field accuracy of a hydronic flow measurement will be within ±10%.

Entering and leaving water temperatures

Again, there are several methods to determine the water temperatures. If the coil is provided with test ports, we could utilize the test ports to measure the entering and leaving temperature. The ports should be located immediately adjacent to the coil connections. This is how the coil rating was determined by utilizing temperature and pressure measurements directly at the connections. Again, the field must replicate the laboratory if we are to utilize performance data that was obtained in the laboratory. If the piping material was metallic (copper or steel), we could simply measure the surface temperature of both the supply pipe and the return pipe and subtract the values to obtain the differential temperature. While this method may not be as accurate as actual temperatures measurements using test ports, this method is employed by TAB firms on a regular basis. Taking temperature measurements with the appropriate thermocouple and digital electronic temperature instrument should have a field accuracy of ±2%.

• Heat transfer capacity – airside

Now we can calculate the amount of sensible heat being transferred to the airstream by using Equation 1.

Equation 1: Sensible Heat - Air

$$Q = Cp \cdot d \cdot 60 \cdot CFM \cdot \Delta T,$$

Where: Q = Heat Transfer (Btu/hr)
Cp = Specific Heat of the air (Std Air = 0.24 btu/lb-°F)
d = Density of the air (Std Air = 0.075 lb/ft³)
60 = Constant (60 minutes/hour)
cfm = Airflow volume (ft³/min)
ΔT = Temperature difference between the leaving air and the entering air (°F)

Note: At standard conditions: Cp • d • 60 = 1.08

• Heat transfer capacity – waterside

Now we can calculate the amount of sensible heat being transferred by the heating hot water by using Equation 2.

Equation 2: Sensible Heat - Hydronic

$$Q = Cp \cdot d \cdot 60 \cdot GPM \cdot \Delta T,$$

Where: Q = Heat Transfer (Btu/hr)
Cp = Specific Heat of the water (Std Water = 1.00 Btu/lb-°F)
d = Density of the water (gal) (Std Water = 8.33 lb/gal)
60 = Constant (60 minutes/hour)
gfm = Wate r flow volume (gal/min or gpm)
ΔT = Temperature difference between the entering water and the leaving water (°F)

Note: At standard conditions: Cp • d • 60 = 500

Example #1:
We will use the following example to summarize all of the above issues. A 10" VAV terminal unit with a hydronic heating coil has been designed with the following requirements: 650 cfm (max), 200 cfm (min), 55°F entering air temperature, 100°F leaving air temperature. The heating coil is designed to heat the air using 0.972 gpm of heating hot water entering at 180°F and leaving at 160°F. The air terminal unit is installed at sea level. The sequence of operation identifies that the heating coil is not energized unless the VAV box is at minimum position. This design represents a very normal set of criteria. The theoretical design heat transfer can be calculated as follows:

The design heating capacity is:
Air: Q = 1.08 • 200 • (100 – 55) = 9720 Btuh
The design heating capacity is:
Water: Q = 500 • 0.972 • (180 – 160) = 9720 Btuh

Thus we have an energy balance.

Field measurements indicate the minimum airflow is 210 cfm, the entering air temperature is 56°F and the leaving air temperature is 98°F. The measured water flow is 1.00 gpm, the entering water temperature is 179°F and the leaving water temperature is 161°F. Determine if the heating coil is providing its rated capacity.

Measured air heat transfer

Based on our data, the measured heat transfer (air) is:
Q = 1.08 • 210 • (98 – 56) = 9526 Btuh or 98% of design. However each of our measurements contains a degree of error: i.e. the field accuracy of the measurements. As

previously stated, the airflow measurement and each of the temperature measurements contain an accuracy of field measurements. So, the individual values could be within the following ranges:

Airflow:	210 cfm, ±10% = 231 cfm – 189 cfm
E.A.T.:	56°F, ±2% = 57.1°F – 54.8°F
L.A.T:	98°F, ±2% = 100.0°F – 96.0°F

Thus the actual heat transfer could be as low as:
Q = 1.08 • 189 • (96.0 – 57.1) = 7940 Btuh, or,
The actual heat transfer could be as high as:
Q = 1.08 • 231 • (100.0 – 54.8) = 11,276 Btuh.

Measured water heat transfer

Based on our data, the measured heat transfer (water) is: Q = 500 • 1 • (179 – 161) = 9000 Btuh or 93% of design. However each of our measurements contains a degree of error: i.e. the field accuracy of the measurements. As previously stated, the water flow measurement and each of the temperature measurements contain an accuracy of field measurements. So, the individual values could be within the following ranges:

Water Flow:	1 gpm, ±10% = 0.9 gpm – 1.1 gpm
E.A.T.:	179°F, ±2% = 182.6°F – 175.4°F
L.A.T:	161°F, ±2% = 164.2°F – 157.8°F

Thus the actual heat transfer could be as low as:
Q = 500 • 0.9 • (175.4 – 164.2) = 5040 Btuh, or,
The actual heat transfer could be as high as:
Q = 500 • 1.1 • (182.6 – 157.8) = 13,640 Btuh.

So, let's put all of this data into perspective and put all of the design, measured and calculated data into tabular form to make some sense of it all.

Heating Capacity	Btuh	% of Design	% of Measured
Design Value (Air)	9,720	-	102%
Measured Value (Air)	9,526	98%	-
Low Value of Field Accuracy (Air)	7,940	82%	83%
High Value of Field Accuracy (Air)	11,276	116%	118%
Design Value (Water)	9,720	-	108%
Measured Value (Water)	9,000	93%	-
Low Value of Field Accuracy (Water)	5,040	52%	56%
High Value of Field Accuracy (Water)	13,640	149%	152%

As can readily seen from the data, the actual sensible heat transfer to the airstream using the measured data can be accurately stated as being somewhere between 82% to 118% of design. The actual heat transfer of the water using the measured data can be accurately stated as being somewhere between 52% to 152% of design. Not very impressive is it? It's also not very accurate. Remember this is a simple VAV box with a single row hot water coil. If this would have been an 8 row chilled water coil with wet-bulb temperatures the results would have been even worse.

Conclusion

Taking and reporting temperature measurements associated with TAB work is not a valid method of comparing field performance to rated capacity. The only value in temperature measurements is to approximate the instantaneous heat transfer at that single point in time. And it has absolutely nothing to do with capacity. Utilizing field data to compare rated equipment capacity performance should not be done due to the inaccuracy of field measurements. Standards and protocols used in rating capacity performance are not meant to be performed in the field.

And finally, remember designing a project's HVAC systems is not an exact science. Performing TAB is not an exact science. That's why they are both identified as engineering. ■

FEATURE: Bledsoe Environmental Systems

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Bring your BEST

For the last 18 years, husband and wife team, Carolyn and Mike Bledsoe, have diligently built a successful business. It's located on a quiet, dead-end street in Camby and even though the street might be a dead-end, the business is certainly not – it is still growing. In fact, it is growing a whole new branch. Bledsoe Environmental has expanded to include Bledsoe Building Commissioning.

When building owners or tenants have a problem with air or water systems, they call Bledsoe. Under the Environmental banner, Bledsoe comes in and gathers data to troubleshoot the issues and make recommendations to resolve them. They test the air and water systems to assess what they are designed to do compared to the current output. Once they identify inefficiencies or deficiencies, they report their findings. It is up to the client to make the changes needed to improve the systems. It was that test and balance work that fueled the frustrations that led to the new branch of operations, Bledsoe Building Commissioning.

The Bledsoes discovered that the problem of systems not operating optimally was sometimes due to initial installation issues, and by the time the problems surfaced, the owners had to hire someone to replace or work on the system. If the systems had been assessed at the time of installation, the corrections could have been made immediately, saving money in operating and repair costs.

Building commissioning is a growing service industry where companies like Bledsoe are a part of the construction team from the onset of the building, renovation or remodel. Their role is to work in concert with engineers and contractors whose responsibility is to make sure the project is completed within budget and on time. Bledsoe's role is to make sure systems are functioning the way they are designed to function and operating the way they should. They can also make recommendations for sustainable or "green" products during the construction to reduce future energy use and maintenance costs.

"In building commissioning, if you get hired at the right time and are involved in the process all the way through, you can make so much more impact," said Carolyn. "We feel like we can actually help (the owners) get the best and most for their dollar. And upon completion, they have a fully functional, operational building."

"We can show an owner a comparison of with us and without our process; how we can lower energy costs, maintenance costs and fix things during the project rather than after the fact. Once they see the savings, they are not as apprehensive with the upfront costs. They see that it is a good ROI for them."

Quality drives the Bledsoe philosophy. Besides being LEED AP certified, the company and employees are all also certified by the National Environmental Balancing Bureau (NEBB). "NEBB sets the policies, procedures and standards for the test and balancing and building commissioning industry," said Carolyn.

Bledsoe Environmental has the only certified hands-on test lab in the Midwest for NEBB. The Bledsoes stepped up to establish the lab when NEBB mandated certification. At that time, the closest test facilities were in Maryland and Florida.

Even though the company operates nationally, Morgan County is home. "Neither of us wanted the rushed, fast-paced life more than we already have," Carolyn said. "It doesn't matter where we have our offices because we're going to be traveling anyway. We might as well be somewhere we are comfortable and somewhere we like."

"We have a passion for our business, Carolyn said. "It's not just a job. We want to bring quality to the process." That explains why the Bledsoes put so much time into training and their work.

However Carolyn says there is an area of business they neglected and that was a mistake.

She said, "In the beginning of any new business you're struggling to get jobs, do the work and make the customer happy. But you have to build relationships. We didn't see the true value of networking and contacts and utilizing other people's knowledge. We thought we were doing the right things by meeting the day-to-day demands.

The Bledsoes established a hands-on test lab for test and balance training for industry supervisors and technicians. This picture shows water pipes and ductwork in the test lab.

"We went years and years and learned the hard way. Eventually we met up with a man who was very helpful. We learned about the business world and what you have to do to immerse yourself into the right places and make the right contacts."

Her advice to new business owners: Surround yourself with good, knowledgeable people – and bring your best. ■

Bledsoe Environment/
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