TAB Journal

AABC

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- Electronically Commutated Motors
- Rooftop Units
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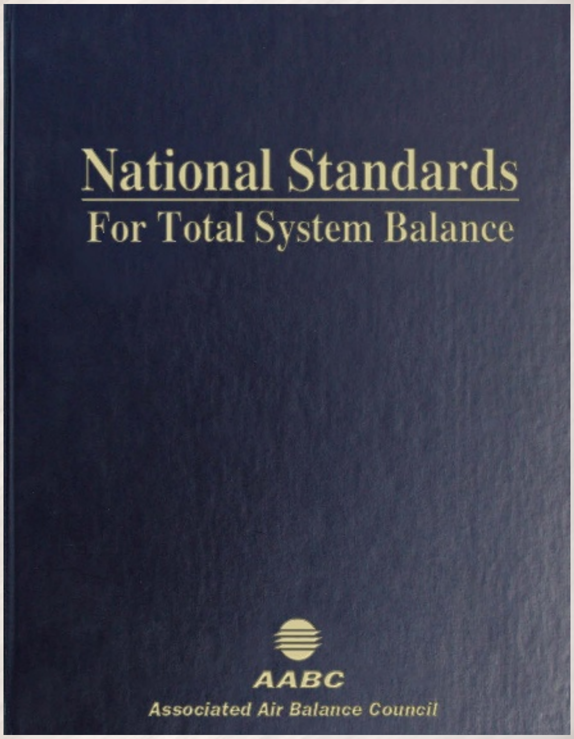

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From the Publisher

The summer 2014 issue of *TAB Journal* looks at the technical expertise of AABC test and balance engineers on topics related to the balancing of air and water systems. The feature story by David E. Gilbert, TBE, of Fluid Dynamics, Inc., showcases the balancing of hydronic systems equipped with a pressure independent characterized control valve (PICCV).

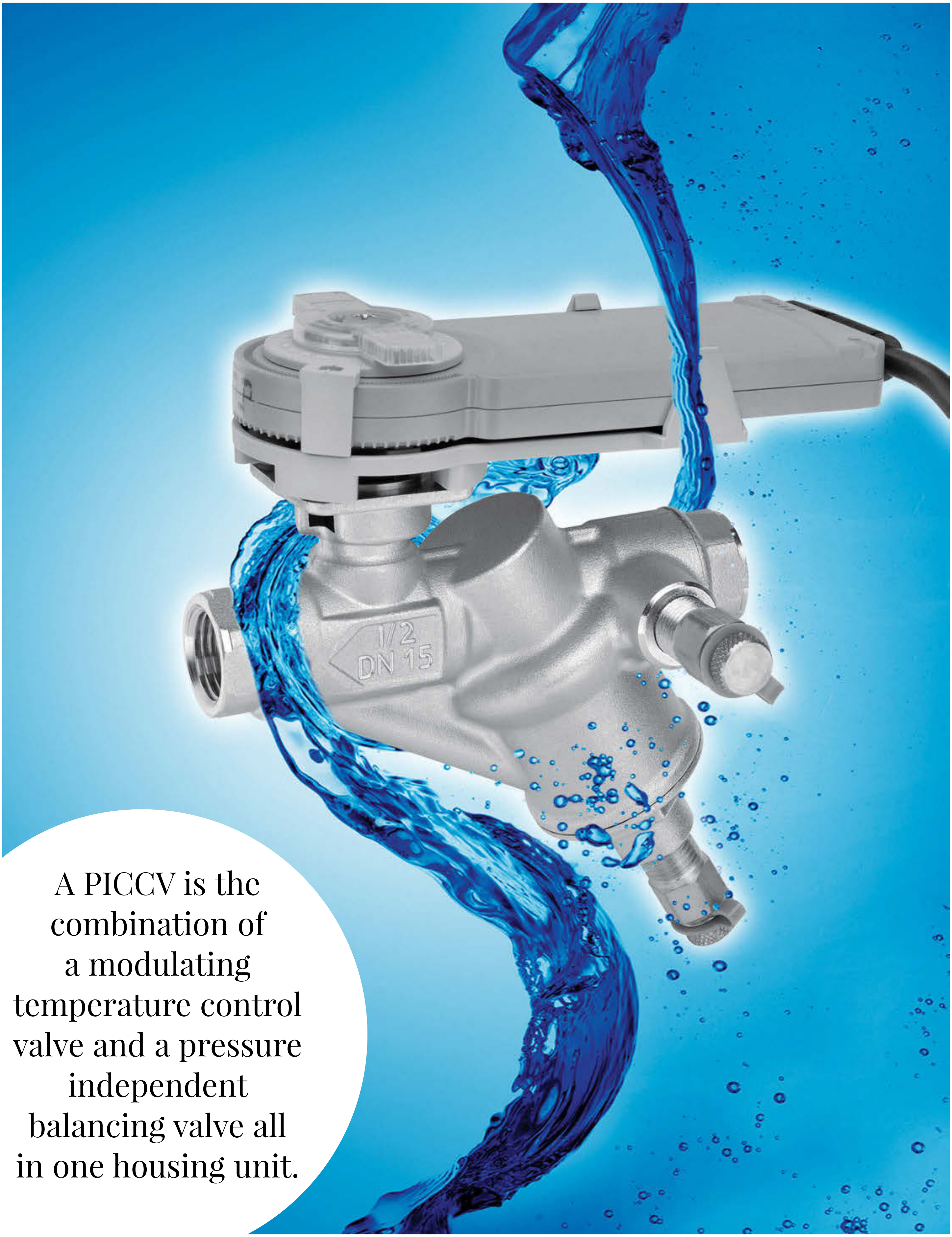
Also in this issue, Joe Helm, P.E., CxA, EMP, of Northwest Engineering Service, Inc., discusses the long term performance challenges that can arise in packaged rooftop units in spite of first cost considerations.

Henry Long, P.E., TBE, CxA, of Building Environmental Systems Testing, Inc., goes over Electronically Commutated Motor (ECM) technology and considerations to keep in mind when balancing systems with ECMs.

Christopher White, P.E., TBE, of Systems Analysis, Inc., looks at the benefits of fan wall systems compared to traditional single motor and fan combinations in air handling units.

And finally, Jonathan Young, TBE, of Southern Balance Company discusses how architectural decisions can affect HVAC performance, and the importance of Total System Balance.

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



A PICCV is the combination of a modulating temperature control valve and a pressure independent balancing valve all in one housing unit.

PICCVs
In The Real World

by David E. Gilbert, TBE, *Fluid Dynamics, Inc.*

When it comes to preparing for the hydronic system test and balance, the approved design documents (mechanical plans, project specifications, and approved submittals) are reviewed and analyzed prior to preparing a field book.

Pump capacities versus connected loads are determined. Temperature control sequences are reviewed and field test instrumentation is evaluated. Typically, the majority of projects are equipped with either pressure gauges or P/T ports at the pump flanges, chiller bundles, air handling coils and terminal reheat coils for use during the balancing procedures. Also, most projects have calibrated flow measurement devices installed at the terminals.

The flow devices that are normally installed are manually field adjustable or a self-limiting flow device usually referred to as "automatic flow control valve."

In the fall of 2009, when preparing to start TAB procedures at a university biology research center, it was discovered that the project was equipped with a different type of balancing valve that was similar to an auto-flow control valve.

The valve was a pressure independent characterized control valve, or PICCV. Per the manufacturer's instructions, if it had a pressure differential of 5-50 psi across the valve body, then it would control at the rated flow. Since these PICCVs were to regulate the flow at design, there were no regular circuit setters or balancing valves installed external to the PICCV.

A PICCV is the combination of a modulating temperature control valve and a pressure independent balancing valve all in one housing unit. Per the manufacturer, in addition to reducing installation costs, the PICCV allows the temperature control valve portion to have the proper authority across its entire range of travel. This is accomplished as the pressure independent portion allows for a constant pressure differential across the control valve at any given position of the flow limiting actuated ball. This is designed to ultimately reduce total pumping, heating, and cooling costs.

The particular manufacturer of PICCV used at this facility will be referred to as Brand X. During preliminary balancing procedures, it was found that many of the actuator valves were not fully open when commanded open via the building automation system (BAS). Consulting with the Brand X representative led to the discovery that these types of actuator valves are limited in their travel based upon the flow rating of the valve. The valve comes either factory set for the design flow rate, or the temperature control contractor programs it for the correct flow rate. In some cases a manual stop is set to a gpm tic mark. Per the manufacturer, if the predetermined valve position is not set accurately, the flow rate will be incorrect. The valve position can be affected by having the manual limit set incorrectly, or having the wrong type of control signal to the actuator motor.

In addition to reducing installation costs, the PICCV allows the temperature control valve portion to have the proper authority across its entire range of travel.

This raised the issue: "If we are to properly verify flow, how do we know that the valve position is correct? Also, how do maintenance personnel know when troubleshooting a temperature problem that the valve stem is designed to only partially open?" Documentation was requested on valve position vs. flow included as part of the submittals. This information could only be obtained by calling in with the date code that is stamped on the side of the particular valve actuator in question. The inclusion of this data within the normal valve schedules was unavailable per corporate decision. The decision was made to trust the valve position setting, similar to trusting the flow ratings stamped on an auto flow control device.

Normally it can be assumed flow is less than design on an auto-flow device if the pressure differential is below the rated pressure range; in this case, below the 5-50 psi range. However, on several devices below the rated pressure differential range, pressure was measured across the terminal unit coils to compare the rated pressure drop for the coil as stated in the submittal data. In many cases the coil pressure drop readings were above the design flow rate. It was explained that a low differential pressure reading across the PICCV does not necessarily mean it has low flow, only that the system pressure at that point is not adequate to achieve the correct operating range for the device.

A meeting was called between representatives from Brand X, several university personnel, and TAB technicians to discuss these concerns. After questions and concerns about trying to establish proper balancing procedures using PICCVs, the Brand X representative stated that these were actually not balancing devices, and should not be viewed as such. When asked how to verify flow throughout the system, the representative indicated four methods for flow verification as listed in their PICCV bulletin.

Those four methods are :

1. total flow verification at the pumps;
2. air temperature differential at the terminal units;
3. water temperature differential at the terminal units; or
4. pressure differential across the coils.

Although such methods are valid for flow verification, they are normally used as alternative methods when flow metering devices are not available, or to substantiate flow device test data. These methods, however, are subject to error and difficult to repeat in today's variable pumping pressure-controlled systems. When bringing up the question of practicality for balancers, the Brand X representative stated that in reality, balancers are not really needed on systems equipped with PICCVs.

In the spring of 2011, at another university project at an agricultural laboratory, it was found the flow regulating device used was the Brand X PICCV. In addition to these, the design engineer specified for the installation of manual-set balancing valves to be used as flow metering devices. Once again, flow problems were encountered. The flow measured using the manual-set balancing valve did not match the rated flow as tagged on the PICCV. In addition, results were inconsistent from device to device, providing a pattern to aid in pinpointing the cause of the discrepancies.

Flow issues were categorized into four basic groups: no flow, low flow, high flow, and normal flow. A letter was written to the project commissioning agent detailing these issues, and giving an example of the measurements for each. Following is an excerpt of that letter:

Example #1: No Flow

The chilled water for FCU-G353 was tested, and we measured little or no flow. A pressure profile was taken at each available p/t port as follows: 72 psi entering strainer, 72 psi leaving strainer and entering coil, 72 psi leaving coil and entering Brand X actuator valve, 55 psi leaving Brand X actuator valve and entering manual-set balancing valve (BV), and 55 psi leaving the BV. This is a line differential pressure of 17 psi, all of which is pressure drop across the Brand X actuator valve. The valve was commanded open/close, with nearly the same results. Of the 23 systems tested, seven devices fall into this category. All the CW mains/branches for these areas were checked to be sure all valves were open. We spot checked the air vents to assure there was no air trapped in the lines.

Example #2: Low Flow

The chilled water for FCU-G347 was tested, and the measured flow obtained from the BV was 76% of what the Brand X PICCV was rated. The PICCV had a pressure drop of 14 psi, which was within its normal operating range of 5-50 psi. The BV was 100% open, and the PICCV was at

a position of 90-100% open during this test (that was the programmed open position for the rated flow). A pressure profile was measured as follows: 76 psi entering strainer, 76 psi leaving strainer and entering coil, 74 psi leaving coil and entering Brand X actuator valve, 60 psi leaving Brand X actuator valve and entering balancing valve (BV), and 59 psi leaving the BV. Of the 23 devices tested, three fall into this category.

Example #3: High Flow

The chilled water for FCU-G463 was tested, and the measured flow using the BV valve was 40% higher than the rated flow of the PICCV. The pressure drop across the PICCV was 9.5 psi. The PICCV was open to a position of 90-100% for this test. A pressure profile was taken for this system as follows: 67 psi entering strainer, 66 psi leaving strainer and entering coil, 61 psi leaving coil and entering PICCV actuator valve, 54 psi leaving the actuator valve and entering the (BV), and 52 psi leaving the BV. Of the 23 devices tested, three devices fall into this category. At another project we just completed using PICCVs, all of the hot water devices tested there fell into this category.

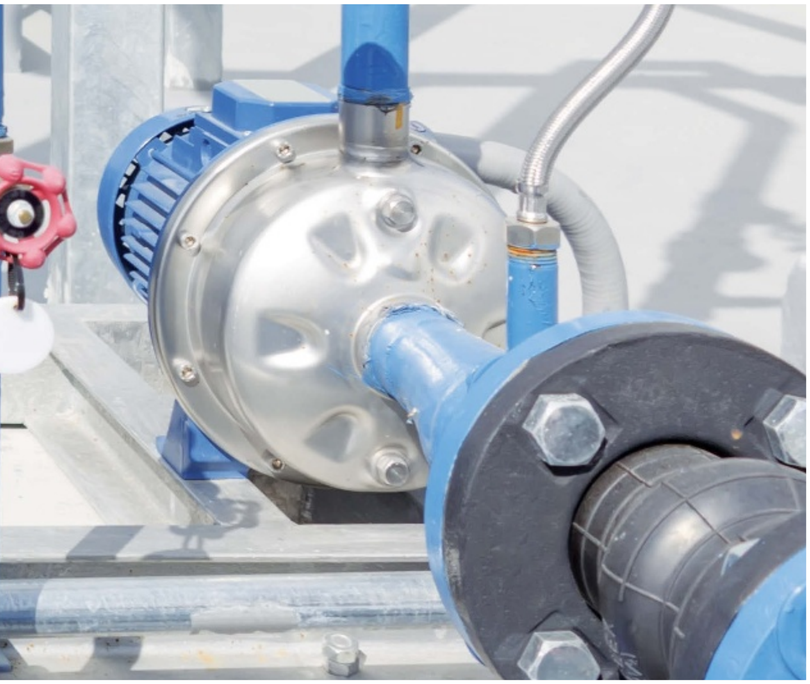
In order to set the flow to design, the BV was adjusted to 50% open to achieve its required pressure drop. This resulted in the pressure drop across the PICCV dropping to 2.7 psi, which is below its operating range of 5-50 psi.

Example #4: Normal Flow

The chilled water for FCU-G450 was tested, and the measured flow using the BV was within 10% of the rated flow of the PICCV. The PICCV was open to a position of 90% for this test (the programmed position for the rated flow). A pressure profile was taken as follows: 70 psi entering strainer, 70 psi leaving strainer and entering coil, 65 psi leaving coil and entering PICCV actuator valve, 56 psi leaving the PICCV and entering the (BV), and 54 psi leaving the BV. Of the 23 devices tested, about 10 fall within this category.

One concern we raised was the location of the BV immediately downstream of the PICCV. We consulted BV documentation, as well as made a call to the BV representative that supplied the equipment. The representative confirmed that the BV valves are typically installed at this location, and does not present a problem with turbulent flows at the p/t ports.

Also, we observed the rated input signals as coded within the model number tagged on the side of the Brand X PICCV. We consulted the temperature control representative onsite. The input that they wired to the control matched that of the control device (0-10V signal) for those devices in question.



On this project, the university chose to deal only with the no-flow items listed in the examples. The mechanical contractor removed a couple of the PICCVs and took them apart to inspect for blockages and debris. They found some of the valves were plugged up with debris. It was also observed that these PICCVs have very small orifices and intricate parts that are very vulnerable in a system that has any amount of fouling in the water. Also, on the items with high flows, the BV circuit setters that had been adjusted to design flow were instructed to be re-open. The univserity chose to leave the flows high in lieu of having external circuit setters that were partially closed.

In mid-winter of 2013, testing and balancing was conducted at a university laboratory that develops techniques for constructing energy efficient buildings. Initial hydronic testing with a new Brand Y PICCV revealed that the valve opened 100% for its maximum flow. This eliminated one of the troublesome variables, i.e. having to worry if the valve position was at the correct setting.

It was observed that these PICCVs have very small orifices and intricate parts that are very vulnerable in a system that has any amount of fouling in the water.

The Brand Y valves have a chart that lists nine (9) psi vs gpm differential pressure ranges required to achieve design flow, depending on the size of the valve and the flow gpm rating. The valve housing has a high and low p/t port with which to test this pressure differential to verify correct flow. The installed coil packs were equipped with an additional (combo pressure/temperature/air vent) port upstream of the housing. This was used in conjunction with the low port to obtain a pressure external to the valve housing.

The problem was that the external pressure differential was often significantly different from the pressure differential as measured using the provided p/t ports on the valve housing. The required differential pressure could generally be generated externally, but not across the provided high/low ports on the Brand Y PICCV. Following is an excerpt of a letter written to the commissioning authority on this project, which details the measurements and troubleshooting methods to demonstrate the problem encountered with these devices:

Initially there was some uncertainty as to whether the required pressure drop for the Brand Y PICCV was across the two p/t ports on the valve body, or across the entire valve, such as would be measured at external p/t ports upstream and downstream of the valve. A Brand Y representative indicated that the rated pressure drop should be across the entire valve, and that the two ports provided on the valve should represent this pressure drop. We informed him that we were, in many cases, getting a significantly different pressure drop when measuring across the entire valve with external p/t ports, and measuring the two p/t ports provided on the valve body. After checking with other Brand Y service personnel, the representative advised that the p/t ports on the valve body were the correct ports to use, and that the pressure drop across these ports should generally represent the pressure drop across the entire valve, although there may be some small difference between this reading and an external pressure drop reading.

What we are finding is that when measuring the external pressure drop across the entire valve, we are able to achieve the rated pressure drop, but we are having trouble obtaining the rated pressure drop across the two p/t ports provided on the valve body. We increased the system differential pressure (DP) from 15psi, to nearly 30psi. This took care of some of the valves in the penthouse, but several other valves showed little/no improvement.

ACP-6B is the worst case. Initially at 15psi system DP, the PICCV had a pressure drop of 6psi across the entire valve, and a pressure drop of 1psi across the provided p/t ports on the valve body. Upon increasing the system DP to 27psi, the pressure drop across the entire valve increased to 12psi, and the pressure drop across the valve body p/t ports was 5psi. This falls short of the rated pressure drop range of 6-36psi required to regulate flow through the valve.

We had the mechanical contractor take ACP-6B valve apart to inspect the valve for any possible obstructions in the valve components. They did find some debris in the spring housing, which we cleaned out. Upon reinstalling the valve and retesting the flow, we did not see any improvement to the pressure drops.

We checked the strainer pressure drop for ACP-6B, which was 1.5psi. We also measured the pressure drop across the coil. The coil pressure drop was 7.5psi, which would indicate a flow of 27.9 gpm comparing the coil gpm to pressure rating as found in the submittal. The design HW flow for this coil is 15gpm.

Following is a summary of four other coils we tested in the mechanical penthouse:

ACP-5B

Initial press. drop @ 15psi system DP (external ΔP/valve body p/t ΔP): 6/3psi

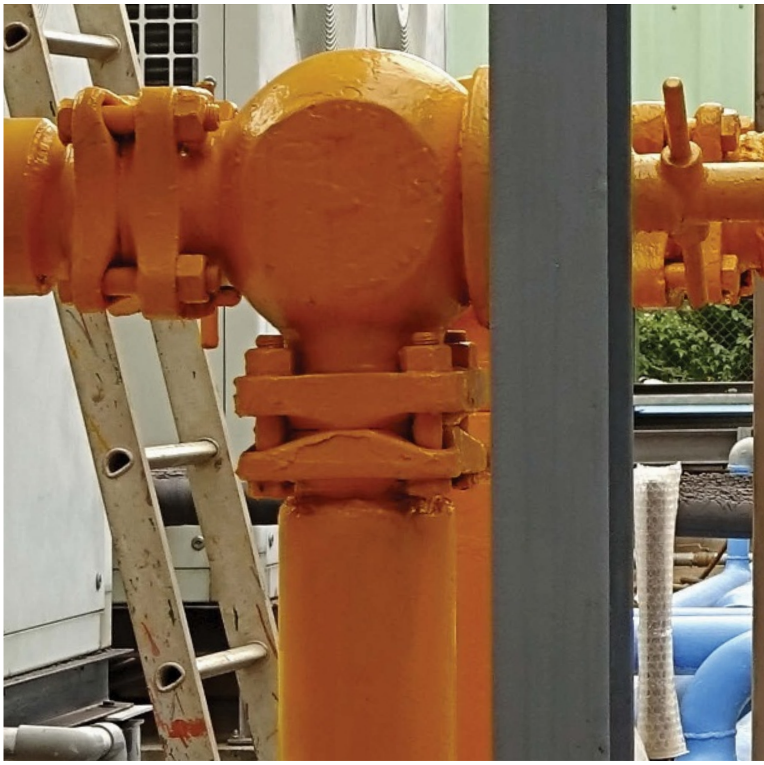
Press. drop @ 27psi system DP (external ΔP /valve body p/t ΔP): 10/4psi

Rated press. drop range & gpm: 6-36psi, 15gpm

Strainer press. drop: 1.5psi

Coil press. drop & calculated flow from coil submittal: 7.5psi, 27.9gpm

Note: after the system DP was nearly doubled, the flow as measured with coil pressure drop was nearly twice the design flow rate. Yet the pressure drop across the valve body p/t ports was still below the required rating of 6-36psi.



ACP-7A

Initial press. drop @ 15psi system DP (external ΔP /valve body p/t ΔP): 6/3psi

Press. drop @ 27psi system DP (external ΔP /valve body p/t ΔP): 11/8psi

Rated press. drop range & gpm: 6-36psi, 15gpm

Strainer press. drop: 0psi

Coil press. drop & calculated flow from coil submittal: 9psi, 30.4gpm

Note: the 8psi is within the rated pressure drop, so we should have the rated flow. The calculated flow from the coil pressure drop does not agree with the Brand Y rated flow.

ACP-7B

Initial press. drop @ 15psi system DP (external ΔP /valve body p/t ΔP): 6/3psi

Press. drop @ 27psi system DP (external ΔP /valve body p/t ΔP): 10/7psi

Rated press. drop & gpm: 4-50psi, 20gpm

Strainer press. drop: 1.5psi

Coil press. drop & calculated flow from coil submittal: 7psi, 27gpm

Note: the 7psi is within the rated pressure drop, so we should have the rated flow. The calculated flow from the coil pressure drop does not agree with the Brand Y rated flow.

ACP-8A

Initial press. drop @ 15psi system DP (external ΔP /valve body p/t ΔP): 6/3psi

Press. drop @ 27psi system DP (external ΔP /valve body p/t ΔP): 11/8psi

Rated press. drop & gpm: 6-36psi, 15gpm

Strainer press. drop: 1.0psi

Coil press. drop & calculated flow from coil submittal: 9psi, 30.4gpm

Note: the 8psi is within the rated pressure drop, so we should have the rated flow. The calculated flow from the coil pressure drop does not agree with the Brand Y rated flow.

All of the VAV's tested on the 3rd Floor, with the exception of two, had a pressure drop of 4 or 5psi across the entire PICCV body as measured with external p/t ports, and a pressure drop of 1 or 2psi using the p/t ports provided on the valve body. This was with a system DP of 15psi. Upon increasing the system DP to 27psi, we retested three of the valves and found the following results:

VAV-3057D

Pressure drop (external ΔP /valve body p/t ΔP): 7/1psi

Rated pressure drop & gpm: 3-35psi, 2.4gpm

Supply to return line pressure differential: 9.5psi

Coil pressure drop & calculated flow from coil submittal: 3.5psi, 5.6gpm*

VAV-3071D

Pressure drop (external ΔP /valve body p/t ΔP): 11.5/1psi

Rated pressure drop & gpm: 3-35psi, 2.4gpm

Supply to return line pressure differential: 16.5psi

Coil pressure drop & calculated flow from coil submittal: 5.0psi, 6.7gpm*

VAV-3055D

Pressure drop (external ΔP /valve body p/t ΔP): 7/0psi

Rated pressure drop & gpm: 3-35psi, 2.4gpm

Supply to return line pressure differential: 11psi

Coil pressure drop & calculated flow from coil submittal: 4.0psi, 6.0gpm*

**The VAV submittals do not specify that the coil pressure drops are in feet or psi. We have assumed feet in our calculations, as this is the standard unit.*

As seen above, we did not improve the pressure drop across the PICCV body p/t ports by nearly doubling the system DP. Also, the calculated flows from the submittals do not agree with the rated flows of the Brand Y PICCVs.

Two of these valves listed above were taken apart by the mechanical contractor to see if there were any obstructions within the valve bodies. VAV-3057D did have a slight amount of debris, but cleaning this out did not improve the pressure drops across the valve.

We need some direction as to why we are not getting the rated pressure drops across these valves, given a significantly high system DP. We also need some direction as to why the calculated coil flows do not match the ratings listed for the PICCVs.

To this day questions remain that were raised in the last paragraph. The Brand Y representative asked for piping line sizes, coil sizes, and valve sizes to be sure there were no problems caused by size mismatching. Everything checked out okay in terms of size. The representative also raised concerns about the location of the p/t ports and their nearness to the PICCV. The Brand Y documentation recommends allowing 2-3 pipe diameters upstream/downstream of the control valve for installation of any p/t ports. He claimed that the disparity between the external pressure readings and the valve housing p/t pressure readings were probably a result of this external p/t being too near the PICCV. However, at another job site where these same type of problems had been observed, the external p/t ports were located a recommended distance away from the PICCV control valve.

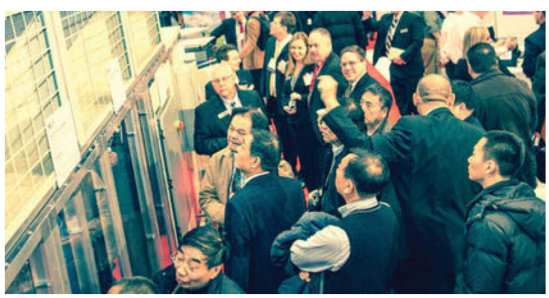
In conversation with the Brand Y representative, he affirmed like the Brand X representative that these devices really do not need any balancing. He stated,

The whole purpose is to reduce energy costs, but how can energy costs be reduced when a building system DP must be run at 30psi just to overcome the apparent resistance of the PICCVs?

"Balancers often claim to have a different flow based upon an external circuit setter reading, or based upon a calculated flow through the coils, and this is just not correct! The only way to verify flow through a PICCV is to measure the differential pressure across it, and if it is within range, you have the correct flow. We do not want circuit setters installed in series with our valves, because they only provide more resistance."

However, based upon the readings above, there is cause for concern about the use of PICCVs. The whole purpose is to reduce energy costs, but how can energy costs be reduced when a building system DP must be run at 30psi just to overcome the apparent resistance of the PICCVs? At least, there needs to be more testing of these devices to address the raised concerns. Design engineers and other personnel at this university have considered the findings, and hopefully will take action to assure that their facilities have the best balance possible, and that they are operating to the fullest efficiency the new equipment is capable of delivering.


GAS FURNACES



PACKAGED ROOFTOP UNITS

First Cost HVAC Solution / Long Term

Joe Helm, P.E., CxA, EMP

Northwest Engineering Service, Inc.

The innovative approaches various manufacturers have used in satisfying the HVAC needs of the building industry are truly remarkable. Expecting a single piece of equipment, like a Rooftop Unit (RTU), to supply air, ventilate when required, economize to save energy, heat and cool for space temperature control, reject waste heat, and effectively pressurize the building is asking a lot from one assembly. For many projects, installation and equipment costs drive the mechanical design and selection decisions. To minimize these costs, the industry demands solutions like the RTU.

This article will explore some of the unintended consequences of selecting this equipment as it impacts the TAB effort, system performance, and occupant comfort. These considerations carry potential hidden operational costs over the life of the building and questionable effects on the Testing, Adjusting and Balancing profession.

TAB and the RTU

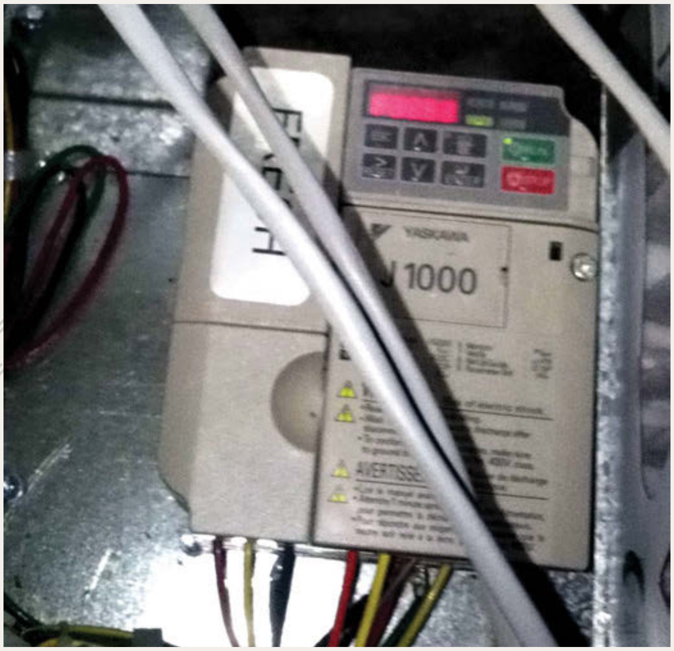
The TAB professional's life isn't necessarily made easier because RTUs were installed on the project. Quantifying how much air is actually handled by the fan and ultimately reaching the intended spaces isn't a trivial exercise. What is the correct

Performance Challenge

> It raises the question, how many owners really understand what they might be giving up in the long run while saving a few bucks on equipment serving a building with a 40 to 75 year life expectancy?

RPM, pressure and BHP required to get the job done efficiently? How do these parameters vary as conditions change in the building below? Whose idea was it to mount the variable frequency drive (VFD) inside the cabinet on the suction side of the fan? Is a drive change really warranted? Has that possibility been accounted for in the estimate?

- Often fan performance itself is impaired by “system effects” resulting from component configuration and orientation inside the unit. The size of the box dictates what can be done to make it all fit.
- Although duct runs that allow for developed airflow are desirable for accurate traverse measurements, these conditions rarely exist in the field, and are even harder to find for installations utilizing RTU equipment.
- Effective sealing between the unit and the ductwork cannot be assumed. Often leakage results in the short circuiting of supply and return airflow. Verifying this condition may require traverses at multiple locations often in areas with difficult access or compromised accuracy due to poorly developed flow. Standard visual site inspections cannot be relied upon verify what is happening between the curb and the unit. A bore scope examination may even be required to help confirm the integrity of the seal.



- Establishing problem damper leakage and adjusting Minimum OSA quantities by direct measurement will be difficult at best. In addition, poor mixing conditions within the unit as well as weather conditions at the time of testing often prevent minimum outside air (MOSA) adjustments using temperature difference calculations.
- Setting up reliable building pressure control using barometric relief or powered exhaust strategies can be quite time consuming, especially when multiple RTUs serve common occupancies.
- Economizer operation and ventilation control may or may not work well together. This depends on the specified Sequences of Operation, the presence and integration of the Building Automation System with packaged controls, the programmer’s understanding of dependent vs. independent control variables, and how much time the TAB provider is willing to donate to troubleshooting the control system.
- Often exhaust air is reintroduced into the unit as fresh air through outside air intake. Various circumstances may cause this condition including, RTU equipment configuration, nearby exhaust fans, proximity to building features, and prevailing winds.

System Performance and Occupant Comfort

- Discharge sensors used for static pressure control might sound like a great idea, but in reality they are only effective as a high limit safety. High pressures are less likely to be an issue for the forward curve fan types commonly used in these devices. When static pressure control is needed, a better strategy locates the sensors nearer the demand downstream in the ductwork. Poor sensor location and ineffective SP control adds to the loss of energy efficiency and greater operating cost.
- Load control for heating and cooling is somewhat limited with off-the-shelf models. When tighter control over space temperature is required, on/off operation fails to provide the desired results. Lack of modulating control causes temperature swings, variation and stratification in the space.
- In cold weather climates MOSA and return air (RA) often complicate freeze protection (e.g. nuisance tripping with manual reset requirements) and discharge air temperature (DAT) control.

- Effects of vibration, weather and other environmental elements compounded by weight reduction for manufacturing and delivery costs help push the limits of equipment durability and reliability.
- Maintenance handled by outsourced service providers may miss the importance of system settings as it relates to energy usage and other performance issues.

First Cost Not the Only Consideration

First cost considerations may make RTU selection a "No Brainer" for the Design and Construction audience. Certainly when budget and performance needs are taken into account, not every building warrants enclosed mechanical rooms with central station air handling equipment and hydronic heating and cooling capability. It raises the question, how many owners really understand what they might be giving up in the long run while saving a few bucks on equipment serving a building with a 40 to 75 year life expectancy?

What are reasonable expectations for the TAB industry with trends like the RTU? How will TAB providers justify spending the extra time and expense needed to assure this equipment is operating as intended? Does the expectation change if the project is to be commissioned? Will that increase or decrease the amount of time spent troubleshooting by the balancer. Certainly this calculation varies by project, especially when the low bid is more important than qualifications, credentials or delivery methods.

For TAB providers, perhaps there are no expectations. It is assumed that this equipment requires no additional controls and configurations are universal with flexible applications implemented by the installing contractor. First cost pressures and the unintended consequences of manufactured solutions such as the Rooftop Unit run the risk of marginalizing the effectiveness of TAB services and reducing the perceived value of Total System Balancing.

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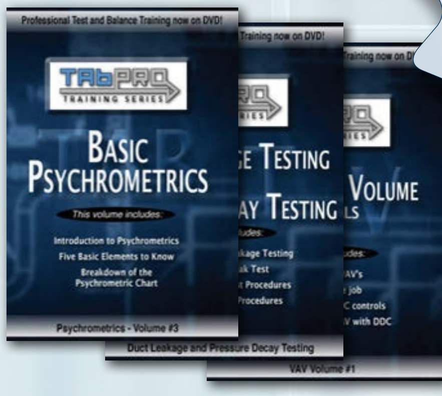

Basic Psychrometrics

DVD format
Run time: 19 minutes
List price: $120.00
Member price: $90.00

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

Duct Leakage and Pressure Decay Testing

DVD format
Run time: 42 minutes
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Balancing Fans with Electronically Commutated Motors (ECM)

Henry Long, P.E., TBE, CxA
Building Environmental Systems Testing, Inc.

Electronically Commutated Motors (ECM) are becoming more prevalent in use in the HVAC industry. Currently ECMs have been used with air handling units, furnaces, heat pumps, exhaust fans, and even fan powered VAV boxes. ECM technology is based on a brushless DC permanent magnet design that is inherently more efficient than the shaded-pole or permanently-split-capacitor (PSC) motor.

By combining electronic controls with brushless DC motors, ECMs can maintain efficiency across a wide range of operating speeds, plus the electronic controls make the ECM programmable.

ECM Technology

The GE ECMs as manufactured by Regal-Beloit are one of the most common ECMs used in the HVAC industry. This motor is a brushless DC, three-phase motor with a permanent magnet rotor. ECMs also incorporate ball bearings in lieu of sleeve bearings, thus the ECM does not have a minimum RPM requirement for bearing lubrication. This allows the motor to operate over a wider speed range, and the motor can handle the capacity previously required by two motors.

ECM motors are made of two components, a control module and motor module. The control module converts AC power to DC power to operate the internal electronics, thus the name DC motor. The micro-processor in the control module is programmed to convert the DC power to a three phase signal to drive the motor. It also has the ability to adjust the frequency (which controls the speed) and the level of current (power/torque) it delivers to the motor.

The motor module is basically a three phase motor with a permanent magnet rotor. The permanent magnet rotor contributes to the electrical efficiency and the ability to control the RPM.

The benefit of all this technology is increased electrical efficiency and the ability to program more precise operation of the motor over a wide range of HVAC system performance needs. Thus the motor can be programmed with software by the original equipment manufacturer. This program has complete control over the RPM, torque, and amp draw of the motor. Programming can be as simple as a 3-speed fan setting or as complex as multi-stage room thermostat that ramps the fan motor up or down for heating or cooling depending on the set point.

Balancing Issues

When balancing the air distribution for a fan system, to reduce the airflow at one inlet/outlet, the balancing damper is usually adjusted. Thus, this makes it harder for the fan to deliver desired CFM. Airflow is then expected to continue down the duct system, go back to the fan or be lost due to an increase in static pressure. This does not happen with the ECM. When an ECM program senses additional static pressure added to the system, it increases the fan RPM (CFM) to compensate for the additional pressure. In PSC motor driven systems, when dampers are adjusted, systems increase in static pressure and lose airflow. ECM driven systems tend to keep their airflow with an increase in static pressure.

When placing a flow hood on an air device, the hood restricts the flow and compensation needs to be made for the back pressure. A reading is typically made with the hood flaps both open and closed to compensate for this. With an ECM, seconds after the hood is placed, the motor detects the rise in static pressure and adjusts. Then, when the hood flap is closed for the second reading, the ECM again sees the restriction and adjusts. Typically, the ECM fan CFM will only vary -7% to +2% with a static pressure range of .50” wg to 1.50” wg.

Another unique ECM balance issue is trying to balance air systems where low speed settings exceed the design airflow. We have noted fan setting on low speed that were 120% to 130% over design airflows. This ECM fan program makes it impossible to balance these systems to design airflows.

Additionally, we also noted the thermostats and ECM should be provided by the same equipment manufacturer. It is very important that the thermostat be wired correctly to the control board as noted in the installation instruction. It is also important that the thermostat/control board are properly grounded or the ECM will not work correctly.

Specific Applications

Because of the constant CFM over varying static pressure characteristics of ECMs, it is very important when field balancing to select the fan application (fan speed) that is as close as possible to the design requirements. The fan applications are programmed by the equipment supplier which may or may not meet the design requirements. The application data is typically not found in your normal equipment performance submittal, but rather in the Installation, Operation and Maintenance product data. This data is usually available at the jobsite or from the equipment manufacturer. Most HVAC manufacturers using ECM technology have several different equipment models depending on the energy savings desired. A few examples are as follows:

Trane Gam5 & Tam7 Series

The Trane Gam5 series is a medium efficiency unit with fan speed taps using dip switches to change the fan speed. The heating and cooling speeds are the same and are factory set as Speed Tap #4. This may or may not match a project’s design CFM. If adjustments are made by changing the dip switches, electrical power must be removed from the unit. The correct voltage (208v or 230v) must also be tapped.

The Tam7 series is a higher efficiency model than the Gam5 and is more complicated to set-up and balance. Initially, a dip switch is provided for an airflow mode of either constant torque or constant CFM. These units have an airflow control module where the airflow for your application is provided through a series of four dip switches. Switches set the outdoor unit capacity and indoor CFM/ton. Additionally, there is an enhanced mode that ramps the blower speed up and down to provide additional energy savings when initially starting and shutting the fan down.

Florida Heat Pump ES Series

The ES series unit by Florida Heat Pump has an ECM Interface Board. Located on the board is a red LED labeled CFM that will blink intermittently. Each blink

ECM technology is based on a brushless DC permanent magnet design that is inherently more efficient.

> HVAC units with ECMs are becoming more widely used due to their higher efficiency, reduced sound levels, and lower maintenance cost.

of the LED represents approximately 100 CFM of air delivery. Also on the Interface Board are a set of Motor Program Jumper Pins. Airflow can be increased (+) or decreased (-) by 15% from the pre-programmed setting by relocating the jumper pins.

Water-Furnace Envision Heat Pump

The Envision Heat Pump can be provided with either a PSC motor, 5-speed ECM or Variable Speed ECM. The 5-speed ECM is a constant torque motor with a specific "torque" value programmed into the motor for each speed selection. As static pressure increases, airflow decreases resulting in less torque. The fan motor adjusts the speed accordingly. The blower performance data is also based on a dry coil.

The variable speed ECM can be programmed either through the unit mounted control board or through a field tool provided by the manufacturer. There are 12 possible dip switch settings to evaluate. A yellow LED in the control board flashes the current variable speed blower speed selection. Programming through the field tool is much easier than at the control board.

Price Fan Powered Constant Volume Terminal Units

Price model FPC and FDC terminal units can be supplied with either PSC or ECM motors. ECM speed controllers are offered as a Standard Speed Controller or Deluxe Speed Controller. The Standard ECM Speed Controller allows manual adjustment of the fan flow by using an adjustment dial in the control board and volt meter. An equation for each box size is provided to calculate the air flow based on a DC voltage measurement. The Deluxe ECM Speed Controller has a red, three-digit display board for reading fan speed, motor RPM, and input voltage. Adjustments can be made to the speed/RPM/input voltage by moving the display push buttons either up or down.

Carrier Model 42

Carrier Model 42 equipped with ECM blowers are controlled by one of three control boards, the Control Board Type, Rheostat Speed Board, and Proportional Speed Board. On the Control Board Type, the unit has been configured at the factory for high, medium (80% of high speed) and low speed (60% of high speed). To adjust airflow, board mounted jumpers have to be relocated as indicated on a configuration chart located on the control box cover.

For the Rheostat Speed Board, Carrier has configured the unit for a high, medium, and low speed. Each output can be adjusted from 0% to 100% for the factory programmed operating range. Airflow is set using a DC voltmeter. The chart on the control box associates airflow rates with the measured voltage for each speed.

The airflow rate for the Proportional Speed Board is similar to the Rheostat Speed Board where speed can be increased or decreased based on DC voltage.

Summary

HVAC units with ECMs are becoming more widely used due to their higher efficiency, reduced sound levels, and lower maintenance cost. Due to their manufactured characteristics, ECMs are programmed by the original equipment provider and will provide either constant CFM, constant RPM, or constant torque.

ECMs present a special challenge to the air balancer. ECMs operate independently of static pressure, thus closing balancing dampers often does not reduce the total airflow significantly. Additionally, when placing an airflow hood on an air device, the ECM program senses the additional static pressure and increases fan speed (CFM) to compensate. Each equipment manufacturer is unique as how to select and adjust fan speeds.

To balance fans systems with ECMs, we suggest the following sequence:

Identifying as early as possible through mechanical submittal review or pre-TAB checkout if an air system uses an ECM.

Once an ECM is identified, obtain the Installation, Start-up, and Operation information for that unit. This information is usually not in the submittal data and has to be obtained directly from the supplier, on the internet, or at the job site (usually in the equipment cabinet).

Review the start-up information and determine method of speed selection and which speed/tap closely matches the design CFM. Fan performance data is usually included in the HVAC submittal data. Note that fan performance from some manufacturers is presented without unit filters and/or a dry coil.

Once the speed has been selected, traverse the main duct, and make a read-only pass of all the outlets. Proportionally balance the system with the flaps open on the flow hood.

If the balancer is unable to reach the design airflow limitations (+/- 10%), the procedure used, the fan speed selected, and that the outlets were balanced proportionately should be noted in the report.

Delivering Air Performance with Fan Wall Systems

Christopher White, P.E., TBE, *Systems Analysis, Inc.*

Over the last several years, many manufactures have increased the usage of multiple supply fan array or "fan wall" in place of traditional single motor and fan application in air handling units. The concept is to replace the single larger, heavier fan and motor combination with a bank of smaller purpose-built direct-drive fans that can deliver the same air performance. These fan wall systems can provide many benefits over the conventional single fan systems.

Fan wall units can be smaller. The physical length of fan wall units is shorter than with conventional systems. The multi-fan wall arrays are typically no more than 48 inches deep. This is accomplished through the use of smaller fans, smaller motors, and direct-drive arrangement. When compared to larger single-motor, single fan units with belts, the overall unit length has been reduced by as much as 30% or more.

Fan wall systems also increase efficiency and reduce equipment sound and vibration. The smaller direct-drive fans can eliminate the need of heavy inertia bases and spring isolation systems. Sound and vibration levels are both reduced. Coil and energy performance can both be reduced by the multiple fan air sources providing a more even distribution of air across the filters and coils versus single fans with a higher concentrated discharge velocity.

Fan wall systems provide the ability of true N+1 redundancy. If a single motor failure occurs, the remaining fans will increase speed to cover for the broken fan until it can be repaired without a loss of unit operation. Single-fan systems and motor configurations have to be shut down completely until repairs are completed. This can be detrimental in critical applications for hospitals and data centers. Fan walls are being increasingly used in schools, hotels, government projects, and more.

From a TAB perspective, the following differences need to be looked at during the balance of these systems:

- Fans may operate by design above 60 Hz; controls and VFDs need to be properly set during startup per manufacturer
- Each fan needs to be checked for proper rotation, amps and rpm
- Check proper operation of backdraft damper on each fan; the damper must close when fan is not running to prevent recirculation inside unit
- Test system in one fan failure mode

Architectural decisions often override mechanical decisions in these buildings and can result in poor HVAC performance relative to design.

BUILDING DESIGN:

Curb Appeal, Efficient HVAC, and the Importance of Total System Balance

Jonathan Young, TBE
Southern Balance Company

Commonly, projects are awarded to test and balance firms at a point well beyond the design phase during the construction phase. Occasionally, a facility's modern or innovative architecture results in odd-shaped buildings with cramped HVAC mechanical spaces. Architectural decisions often override mechanical decisions in these buildings and can result in poor HVAC performance relative to design. Incorporating the Total System Balance process, specifically the initial contract document review and analysis, can alleviate some potential roadblocks to ideal system performance, or at least notify the design team of potential shortcomings.

One such circumstance was a project situated on an odd-shaped plot of land. This multi-story project had both circular and rectangular rooms with very few parallel walls and limited ceiling plenum space for electrical and mechanical equipment installation. Additionally, the design incorporated one small basement mechanical room which housed all air-handling units, and pumps for the building, including the exhaust fan system. The architectural preference was to ensure only one rear/lower section of the building contained visible louvers or penetrations pertaining to the HVAC system.

Although a design review was performed on receiving the project, the construction and system installation was already significantly underway. The mechanical room space had decreased from the original design due to architectural limitations; however, the selection of the air-handling units and exhaust fans was unchanged.

One major example centers on the exhaust system, which was handled by one basement fan rated for 1.0" w.c. external static pressure. The fan system handled 21 separate exhaust inlets/registers spread out over three floors. The major issue was that one important exhaust register on the 1st floor was rated for 100 CFM, but the 10" x 10" ductwork serving it had to travel over 100 feet up and away from the mechanical room to the other end of the building, and then zig-zag around a large rotunda and back to the front of the building. This airflow path consisted of 10 elbows and 4 elevation changes in order to reach the fan. This small exhaust grille could have had its own small exhaust fan connected to an outside wall louver less than 20 feet away (however, this would have been located on the front / visible side of the building). To reach proportional airflows of only 70% of design CFM, the entire system required substantial throttling of all other branch dampers just to accommodate this problem area. The overall effect was very high fan static and low airflow.

> This entire project validates the Total System Balance approach, and has renewed the emphasis to encourage owners and contractors to get AABC-Certified firms involved on their projects as early as possible.

The next example on this system involves the excess pressure loss in the supply ductwork from one AHU serving the top floor. The supply duct has 4 90-degree elbows within the mechanical room, travels up 2 floors, and turns 90 degrees into a bullhead tee. The variable-air volume system AHU was sized for no diversity, thankfully, and a supply air duct static set point was determined to be 1.5" to reach design CFM at certain terminal units. After accurate calibration of all terminal unit air valves. It was no surprise to see that there was a 1.1" w.c. loss of static pressure in the main duct between the fan and the first terminal unit take-off on the top floor, which resulted in low system airflow and starved terminals during maximum load testing. Although the original design of the units incorporated high static pressure ratings, the tightened mechanical spaces required additional duct modifications, transitions, and elbows to reach the desired floor penetrations. As a result, additional sheave changes to increase fan speed provided only very marginal airflow increases, on top of the new higher amperage and brake horsepower required.

This condition is not unusual or unique to test and balance firms. It does, however, again illustrate that design-phase analysis of the mechanical system with the design team should be emphasized whenever possible. In this case, the design team and owner were faced with decisions on accepting less than ideal HVAC performance at the expense of maintaining the architectural design concept. Much expense was wasted in analyzing static pressure drops and ductwork installation, when a thorough pre-construction review would have highlighted these two potential issues. This entire project validates the Total System Balance approach, and has renewed the emphasis to encourage owners and contractors to get AABC-Certified firms involved on their projects as early as possible.

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(204) 224-1416

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

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

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

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Controlled Air Management Ltd.
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(506) 852-3529

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

Source Management Limited
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(506) 443-9803

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

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Air & Water Precision Balancing, Inc.
Toronto, Ontario
(647) 896-5353

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

Designtest & Balance Co. Ltd.
Richmond Hill, Ontario
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Dynamic Flow Balancing Ltd.
Oakville, Ontario
(905) 338-0808

Kanata Air Balancing & Engineering Services
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Pro-Air Testing Inc.
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(416) 252-3232

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

VPG Associates Limited
King City, Ontario
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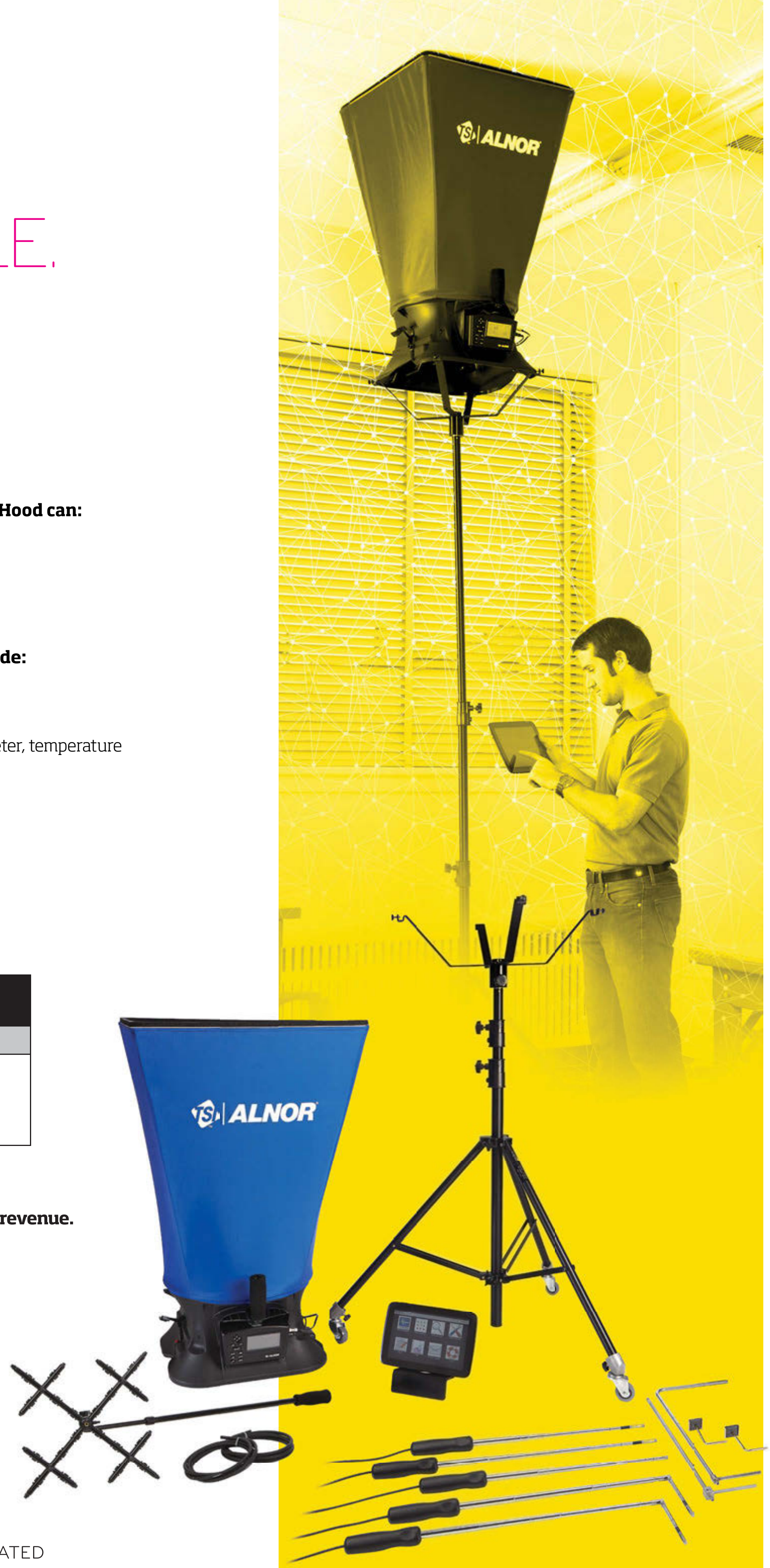

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