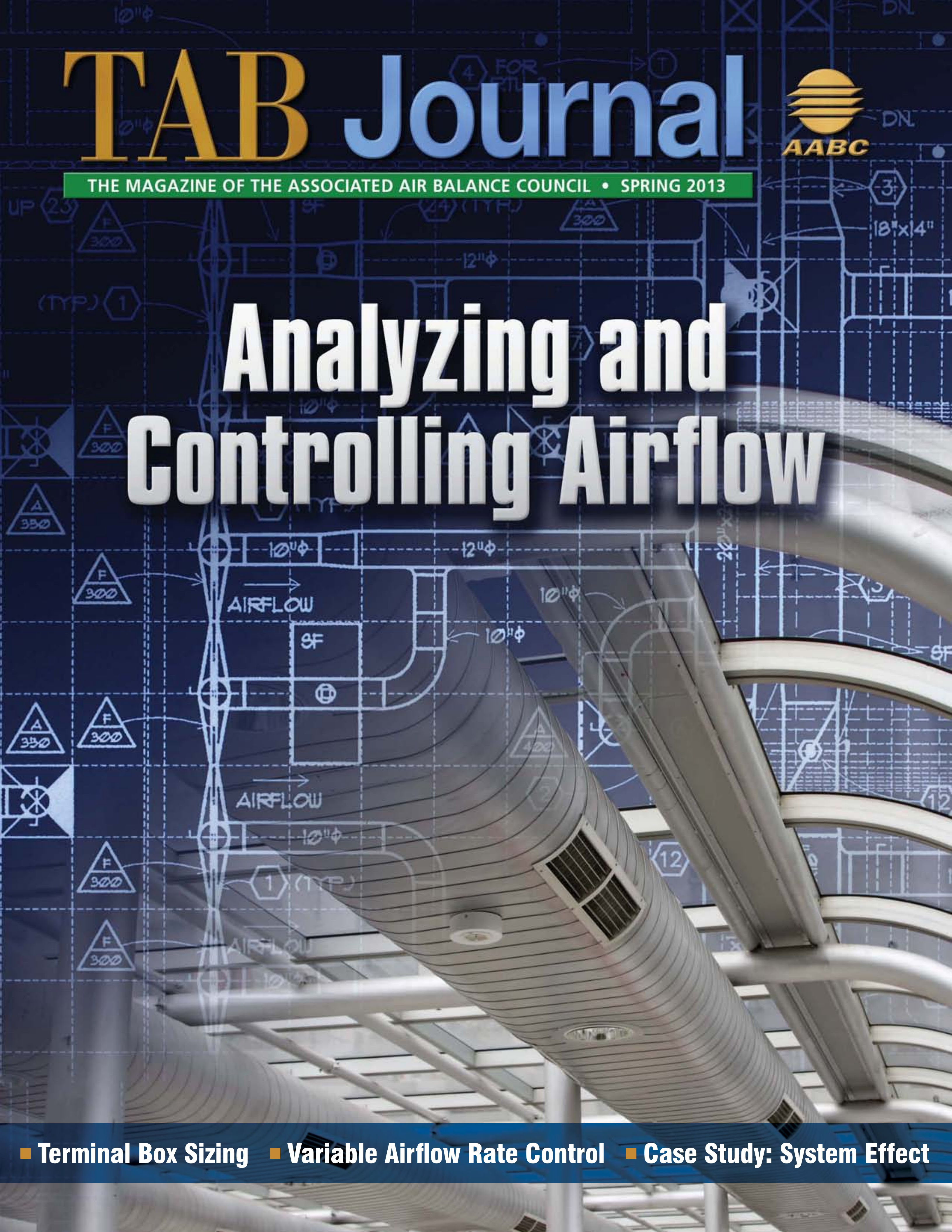
TAB Journal

AABC

THE MAGAZINE OF THE ASSOCIATED AIR BALANCE COUNCIL • SPRING 2013



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- Case Study: System Effect

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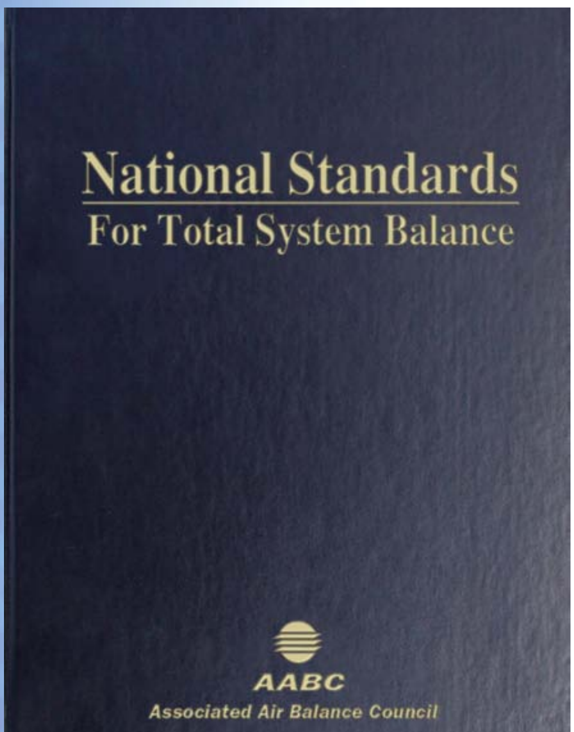

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From the Publisher

The Spring 2013 issue of *TAB Journal* focuses on topics concerning airflow analysis and control. First, Alan Little, TBE, of Engineered Air Balance Co., Inc., explains the important correlation between minimum CFM and choosing the correct terminal box size. Next, Surrinder Sahota, TBS, of Designtest & Balance Co., Inc., provides a comprehensive look at how best to control variable airflow rates. Finally, Kevin Underwood, TBE, of Engineered Air Balance Co., Inc., presents a case study on system effect.

Also in this issue…

Frederick Seed, TBE, of Arizona Air Balance Company, explains why duct leakage testing is particularly important in hospital and laboratory environments.

Timothy Demchuk, TBE, of Precision Air Balance Company, Inc., takes a close look at why not all variable frequency drives (VFDs) are created equal.

In "Effective Fire System Testing from Door to Door," Ray Armstrong of Mechanical Test & Balance takes readers on a step-by-step testing of fire life safety systems.

Mechanical Data Corporation's Brandon Johnson, TBE, outlines the differences between belt-driven and direct drive fans, and also explores a new technology for air handling systems called "fan banding."

And finally, this issue's Tech Talk answers questions regarding the best instruments to use when testing exhaust and return systems and where and when to use volume dampers.

"There is more to consider than just the maximum CFM when selecting the terminal box size. The minimum CFM is just as important and possibly even more important to the selection process."

Minimum CFM and Terminal Box Sizing:

What a TAB agent should know

Alan Little, TBE
Engineered Air Balance Co., Inc.

Participants at a recent ASHRAE meeting discussed terminal boxes with an emphasis on design requirements and box sizing for maximum and minimum CFM, medium and low pressure static pressures and noise criteria.

During the maximum and minimum CFM discussion, an actual project terminal box plan schedule was cited as an example. The speaker noted that terminal boxes need to be sized at maximum airflows with an equivalent of about 2000 FPM at the box inlet. There was also a note on the schedule, indicating that the minimum CFM was to be set at 20% of the design maximum CFM. The presenter did note that the minimum CFM of a terminal box is limited by the velocity controller's minimum operating pressure.

While the presenter's information seemed sound in this case, it could mislead a design team into thinking that a design minimum CFM of 20-25% of the maximum CFM was attainable under any conditions.

The following table indicates the velocities through the terminal boxes with the maximum CFM selected at about 2000 FPM with a 20% design maximum that was cited in the presentation. Included for comparison are airflow minimums at the equivalent of 635 FPM (.025" w.g.), 565 FPM (.020" w.g.) and 490 FPM (.015" w.g.).

Max CFM (2000 FPM)	Min CFM (20% Max.)
395	80
700	140
1090	220
1570	315
2140	430
2800	560

Box Inlet Size	Min. CFM at 635 FPM (.025" w.g.)	Min. CFM at 565 FPM (.020" w.g.)	Min. CFM at 490 FPM (.015" w.g.)
6"Ø	125	110	95
8"Ø	220	195	170
10"Ø	345	305	265
12"Ø	390	350	300
14"Ø	680	605	525
16"Ø	885	790	685

Note: This data is based on a 100% free area of inlet.

So how does this relate to test and balance? TAB services include reviewing the project documents (plans, specifications and submittal) for any balance issues. During the document review process, these issues should be identified and, if necessary, requests for information should be sent through the proper channels to clarify minimum CFM requirements and other concerns.

When testing begins, the minimum CFM the velocity controllers can maintain for the various box sizes needs to be determined. If the design team mandates a minimum CFM that cannot be achieved, the test data needs to be provided and more discussions should be held, or additional information obtained in order to resolve the problem.

Minimum airflows are an important part of providing the required minimum outside air for the air handling unit system. The total of the minimum airflows for all terminal boxes in a system must be at or above the design minimum outside air CFM of the air handling unit.

With a greater focus now on the indoor environment, energy resources and LEED, the terminal box maximum and minimum CFM designs are connected more than they were in the past. Keeping the velocity through terminal boxes low at maximum CFM reduces the required static pressure and reduces energy consumption. However, there is more to consider than just the maximum CFM when selecting the terminal box size. The minimum CFM is just as important and possibly even more important to the selection process. The minimum CFM (velocity) is a physical limit that the controller needs to be able to function.

The minimum CFM of a terminal box is limited by the accuracy of the velocity controller when operating at the required minimum differential pressure. The velocity controller must be able to maintain a consistent and repeatable minimum velocity (airflow). In many cases, when a velocity controller is operating at or slightly below its minimum pressure limit, the controller has a tendency to lock-up and/or provide an inaccurate airflow. When this occurs, the velocity controller cannot differentiate between the minimum CFM value and no airflow (zero CFM).

There are also instances when a set point, which is below the minimum limit of the controller, is input, and the DDC system may indicate the CFM is at the minimum CFM set point, but the actual CFM could be either significantly higher or lower.

The box manufacturers' flow charts I reviewed vary from minimums of 0.03" w.g. (695 FPM) down to 0.010" w.g. (400 FPM). It is important to note that even though a box manufacturer's flow chart indicates a minimum of 0.010" w.g., that doesn't mean 0.010" w.g. is attainable. The minimum CFM is a function of the velocity controller manufacturer, not the box manufacturer. The actual minimum CFM of a terminal box is limited by the accuracy of the velocity controller at its specified minimum differential pressure.

A typical velocity controller is able to operate consistently down to a minimum differential pressure of about to 0.025" w.g. (635 FPM). Some of the better controllers are able to operate as low as 0.020" w.g. (565 FPM) and 0.015" w.g. (490 FPM). The flow sensors in some terminal boxes do provide amplification, which does allow for lower minimum airflows.

Review of mechanical plan terminal box schedules of several local mechanical engineers showed the average maximum CFM to be equivalent to an airflow inlet velocity of 2100 FPM for each box size.

The table below indicates a typical range for the most common box sizes when selected at a maximum of 2100 FPM and corresponding minimums of 20% and 30%.

BOX INLET SIZE	MAXIMUM			MINIMUM (20% MAX.)			MIN. (30% MAX.)		
	CFM	FPM (INLET)	DP "w.g.	CFM	FPM (INLET)	DP "w.g.	CFM	FPM (INLET)	DP "w.g.
6	250-410	1275-2100	.10-.28	50-82	255-420	.004-.011	75-125	380-640	.009-.026
8	411-725	1190-2100	.09-.28	82-145	240-420	.0036-.011	125-220	365-640	.008-.026
10	726-1150	1330-2100	.11-.28	145-230	265-420	.004-.011	220-345	405-640	.010-.026
12	1151-1650	1465-2100	.135-.28	230-330	295-420	.005-.011	345-495	440-640	.012-.026
14	1651-2250	1545-2100	.15-.28	330-450	310-420	.006-.011	495-675	465-640	.013-.026
16	2251-3000	1610-2140	.16-.285	450-600	315-430	.006-.012	675-900	480-645	.014-.026

Note: This data is based on a 100% free area of inlet.

Based on the data above, none of the design maximum airflows for boxes of various sizes could provide 20% minimums. Any minimum CFM with an equivalent velocity of 565 FPM (0.02" w.g.) or less may not be able to be maintained. Keep in mind that poor box inlet conditions (radical turns or offsets in flex and/or ductwork) may limit the ability to maintain consistent design minimum airflows, even with the better lower range velocity controllers and/or flow sensors.

Just increasing the minimum CFM may not be the solution and could cause other problems. In some cases, areas where the terminal boxes record minimums of 25-30% or higher might be subject to excessive cooling. With energy conservation being such a priority, heating may not be available all year.

Many facilities, such as offices or schools, might shut down their heating sometime in the spring and will not restart the units until fall. In these areas, room temperatures may be tolerable in winter when heat is available or during summer when there is a significant cooling load, but under light load conditions some areas may experience over-cooling. Under light load conditions, exterior areas that are served by terminal boxes with minimum airflows may be susceptible to over-cooling when heating is disabled.

While performing a document review of terminal boxes, there are multiple issues to consider and each one can have an effect on another. The only real way to resolve issues related to the design minimum CFM on a project is to perform testing to determine the actual minimum CFM that can be attained for each typical size terminal box on a project. Once the actual minimum CFM is known for each box size, we can either verify that the design minimum airflows can be achieved or provide information about the actual minimum airflows that can be attained.

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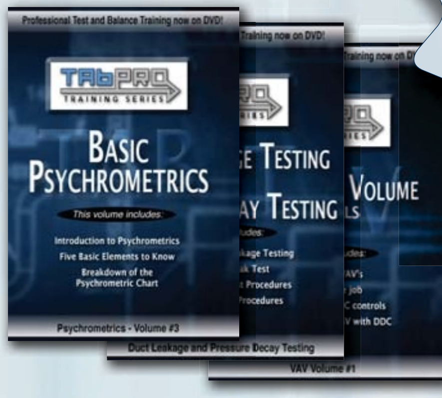

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Controlling Variable Air flow Rates

Surrinder Sahota, TBS
Designtest & Balance Co., Ltd.

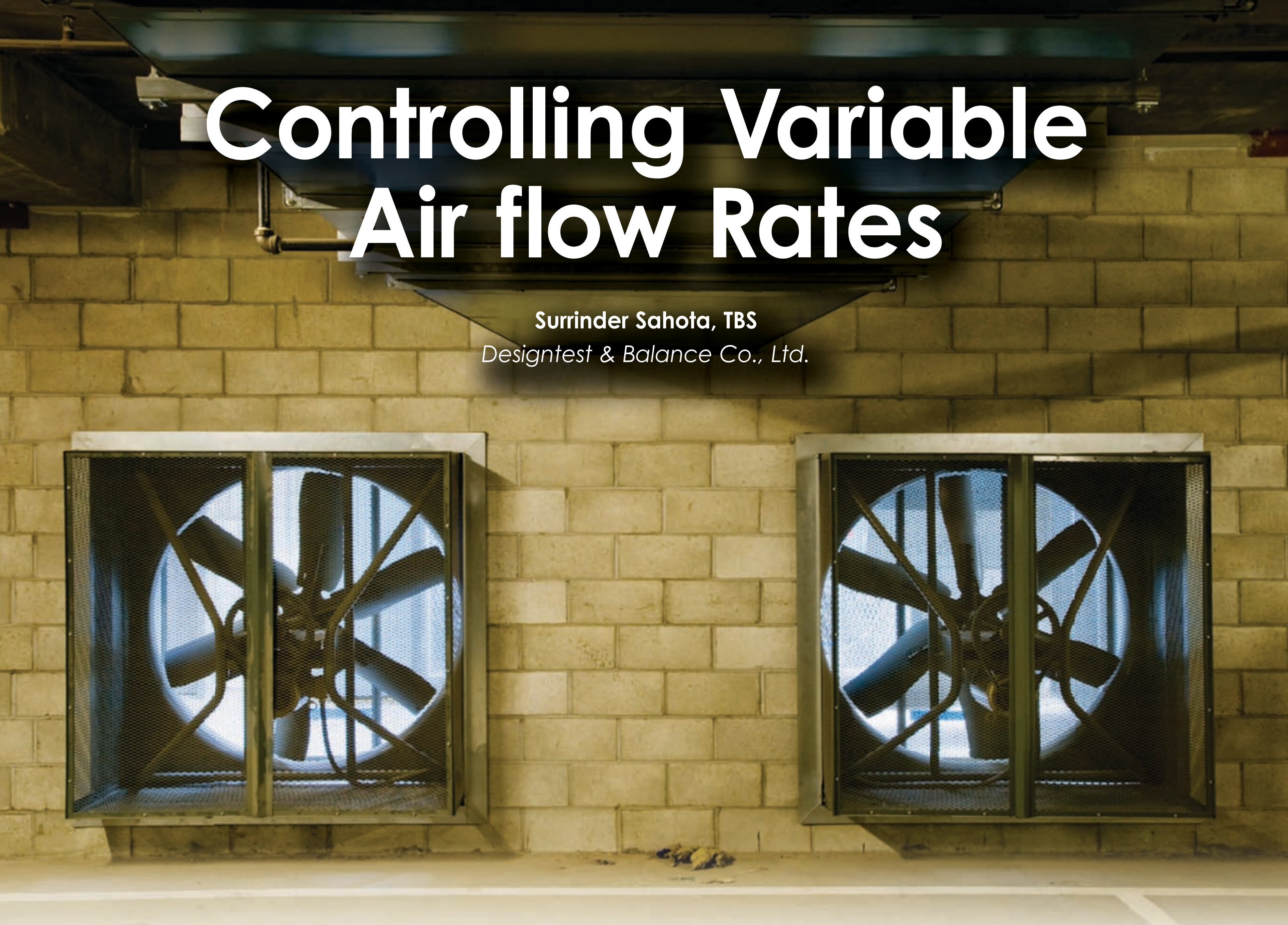
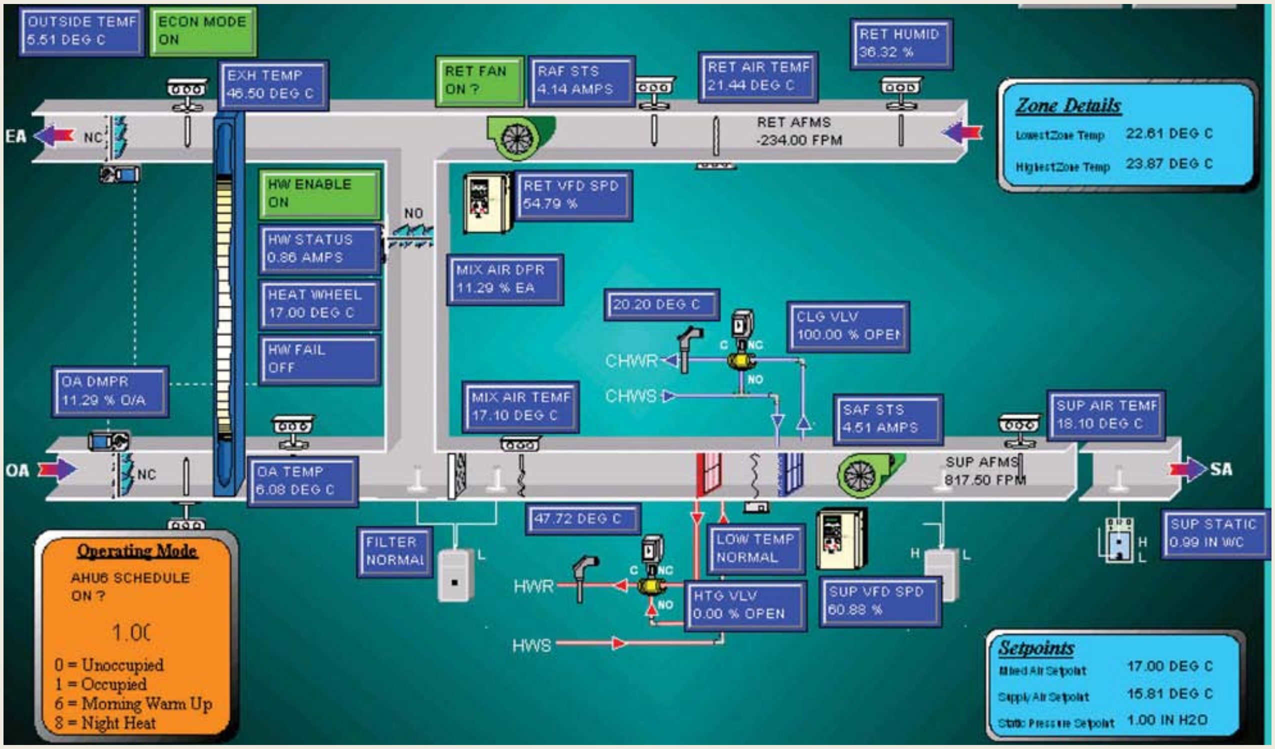


Figure 1.



In this building automation system (BAS) graphic, (Figure 1) there are two unique variable flow rate systems, and each one has a fixed value to consider for BAS control.

One constant value is the airflow rate required to control building pressurization and another constant value is the minimum airflow rate to satisfy ventilation requirements.

Control of Building Pressurization

The first variable flow rate system with a fixed value is the supply air fan and return air fan relationship. The supply air fan was designed to deliver an airflow rate of 10,384 l/s while the return air fan was designed to extract an airflow rate of 8,260 l/s. The difference—2,124 l/s—makes up for air extracted by local exhaust fans and for building pressurization. If this were a constant airflow rate system, the air balancer would set the speed of each fan to deliver the design flow rate and the flow rate difference of 2,124 l/s would be maintained.

Variable flow rate systems are substantially different. The supply air fan operates at a variable speed thereby delivering a variable airflow rate. The supply airflow rate is not controlled by the BAS. The pressure in the supply air duct or the VAV damper position is the control reference for the BAS. There is no way to derive the delivered flow rate from the measured fan speed. The only BAS method to calculate the flow rate delivered by the supply air fan is to measure with an airflow station, or measure the instantaneous flow rates of all VAV terminals served by that supply air fan.

The measured supply airflow rate becomes a BAS data point. The fixed difference of 2,124 l/s is subtracted from that data point to yield a control point, called the return airflow rate set point.

The BAS then compares measured flow rate from the return airflow station with the calculated return airflow rate set point and proportions the signal to the VFD to maintain the return air set point.

Σ of Supply Air (Variable Input) – Difference (Fixed Input) = Return Fan Flow Rate (Variable Set Point)

Figure 2.

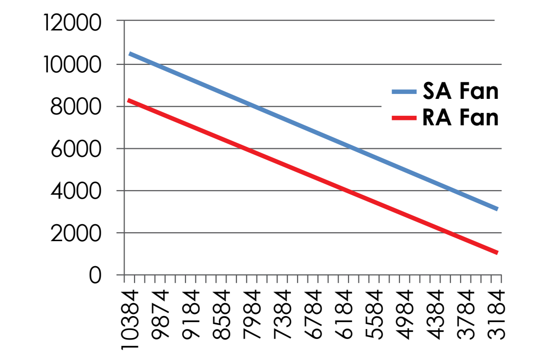


Figure 3.

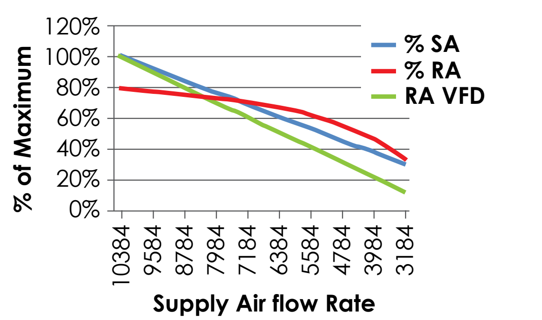


Figure 2 indicates the design, maintaining a fixed difference in airflow rates of 2,124 l/s. Figure 3 tracks the return airflow rate as a percentage of the supply airflow rate and tracks the signal to the return air fan VFD. As the measured supply airflow rate declines from 100% to 36.4%, the BAS command to the RA VFD declines from 100% to 20%.

As the measured supply airflow rate changes from 10,384 l/s to 3,784 l/s, the return air set point will change from 8,260 l/s to 1,660 l/s to maintain the constant difference of 2,124 l/s. The supply air fan VFD will decrease from 100% to 50%, 30 Hz, and the return air fan VFD will decline from 100% to 20%, 12Hz.

	VAV @ Maximum	VAV @ Minimum	Min as % of MAX
Supply Airflow	10,384 l/s	3,784 l/s	36.4%
Return Airflow	8,260 l/s	1,660 l/s	20.1%
Difference	2,124 l/s	2,124 l/s	-
RA as % MAX SA	79.5%	16%	-
RA Fan VFD	100%	20%	-

Control of Ventilation Air

The flow rate of ventilation air into a system is a function of the pressurization airflow rate and a function of the relative control of the outdoor air damper, the return air damper and the exhaust air damper. There will be a constant value, specifically a minimum ventilation airflow for each air handling system.

Possible scenarios:

- Pressurization airflow exceeds the minimum ventilation airflow rate.
- Pressurization airflow equals the minimum ventilation airflow rate.
- Minimum ventilation airflow exceeds the pressurization airflow rate.

If either scenario 1 or scenario 2 is true, then the only BAS command necessary to maintain minimum ventilation is to command the outdoor air damper open to its minimum position as determined by the air balancer. The return air damper will remain 100% open and the exhaust air damper will not be open. The outdoor air damper and exhaust air damper may be opened in proportion and the return air damper proportioned towards closed for economizer operation. This is applicable to fixed supply airflow rate systems and variable supply air systems.

Unfortunately, scenario 3 is common for school applications and quite complicated to control. The system under consideration was designed to deliver a minimum ventilation rate of 60%, 6,230 l/s. Operating the outdoor air damper according to the second scenario would introduce only 2,124 l/s of ventilation air, only 34% of the design flow rate of ventilation air.

BAS Command for a Constant Flow Rate System

The outdoor air damper is set to allow a flow rate of 6,230 l/s while the return air damper is set to restrict the flow rate of return air to 4,154 l/s. (10,384 l/s - 6,230 l/s) (60% outdoor air and 40% return air).

The mixing dampers must be set so the return air fan rejects 4,106 l/s via the exhaust air damper while 4,154 l/s is allowed to flow through the return air damper (8,260 l/s -4,154 l/s) (49.7% exhaust air and 50.3% return air). These will be the damper positions any time mechanical heating or mechanical cooling are active.

BAS Command for a Variable Flow Rate System

Refer to the table and the graph to the right and observe how the amount of return air used decreases from 50% of the return air fan flow rate at the supply air fan maximum to 0% at about 60% of the supply airflow rate. The return air fan must continue to operate even though none of the air is used as return air.

The outdoor airflow rate and the exhaust airflow rate are constant until the measured supply airflow rate is less than the minimum outdoor airflow rate. Then both flow rates decline.

- The first step is to verify that the design building pressurization flow rates were calibrated.
- Set the sum of the flow rates of the VAV terminals to their design flow rate of 10,384 l/s. This would include all the VAV terminals if there is no diversity in the system.
- Open the exhaust air damper 100% and the outdoor air damper 100%.
- Manipulate the return air damper towards closed until a flow rate of 4,154 l/s is measured either directly or indirectly. Record the damper position as “maximum return air position.”
- Increment the exhaust air damper towards closed until the threshold of control at 4,106 l/s is achieved. The threshold of control is the position where additional closing would force more air through the return air damper. Record the damper position as the “maximum exhaust air position.”
- Increment the outdoor air damper towards closed until the threshold of control at 6,230 l/s is achieved. The threshold of control is the position where, if it were closed further, it would draw more air through the return air damper. Record the damper position as the “minimum outdoor air position.”
- Set a proportional control: return air damper varies from maximum return air position to 0% open as return air fan varies from 8260 l/s to 4106 l/s.
- As the return air damper approaches 0% open, the outdoor air damper and the exhaust air damper could be allowed to open or be maintained at their “minimum outdoor air position” and “maximum exhaust air position” respectively.

Note the importance of a unique BAS address for each of the three dampers.

System Performance as Measured Supply Airflow Rate Decreases

SA Fan	RA Fan	OA	RA Used	EA= RA Reject
10384	8260	6230	4154	4106
10184	8060	6230	3954	4106
9984	7860	6230	3754	4106
9784	7660	6230	3554	4106
9584	7460	6230	3354	4106
9384	7260	6230	3154	4106
9184	7060	6230	2954	4106
8984	6860	6230	2754	4106
8784	6660	6230	2554	4106
8584	6460	6230	2354	4106
8384	6260	6230	2154	4106
8184	6060	6230	1954	4106
7984	5860	6230	1754	4106
7784	5660	6230	1554	4106
7584	5460	6230	1354	4106
7384	5260	6230	1154	4106
7184	5060	6230	954	4106
6984	4860	6230	754	4106
6784	4660	6230	554	4106
6584	4460	6230	354	4106
6384	4260	6230	154	4106
6184	4060	6184	-46	4060
5984	3860	5984	-246	3860
5784	3660	5784	-446	3660

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System Effect: CASE STUDY

Kevin Underwood, TBE
Engineered Air Balance Co., Inc.

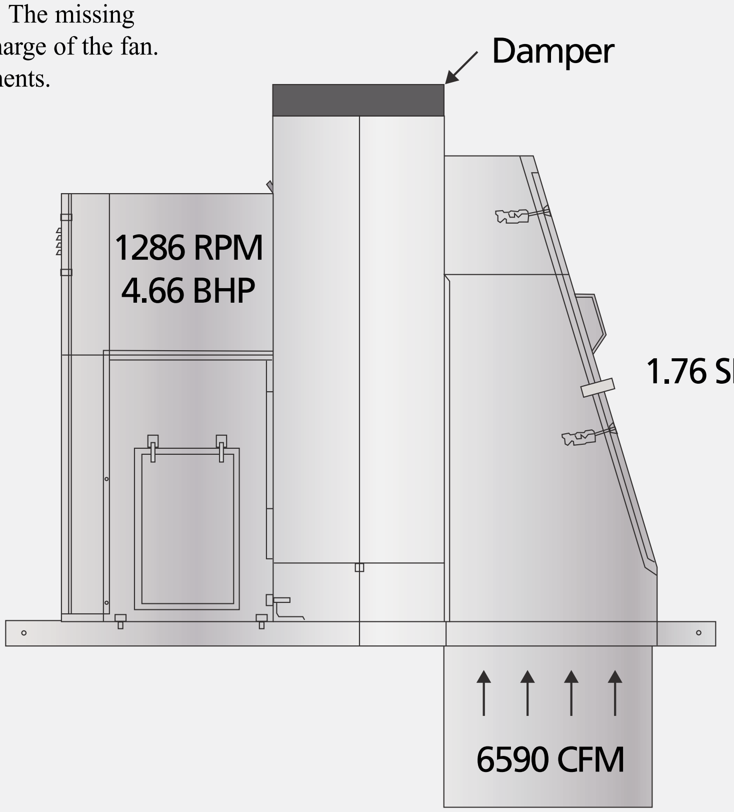
A common term used to describe some inlet and outlet conditions that affect the performance of fans is "system effect." The term refers to the results obtained after the fan was tested in a laboratory compared with how the fan was installed. A case study was performed on an exhaust fan that was not performing as submitted on recent project. The following was the original submittal data for the subject fan:

- Flat Blade Centrifugal Blower
- Model 245 CPS
- Design SP = 1.00" w.c.
- Design Motor HP = 5.0
- Counterclockwise Up-Blast
- Design CFM = 8195
- Design Fan RPM = 1080

After the exhaust was proportioned within the system, the fan was operating at 80% of the design CFM, and the motor was operating at nameplate amps.

STEP 1: Evaluate the air balance measurements and compare them with the fan curve data. The air balance measurements are correct. The missing component is the effect of the back draft damper on the discharge of the fan. This data has to be added back into the air balance measurements.

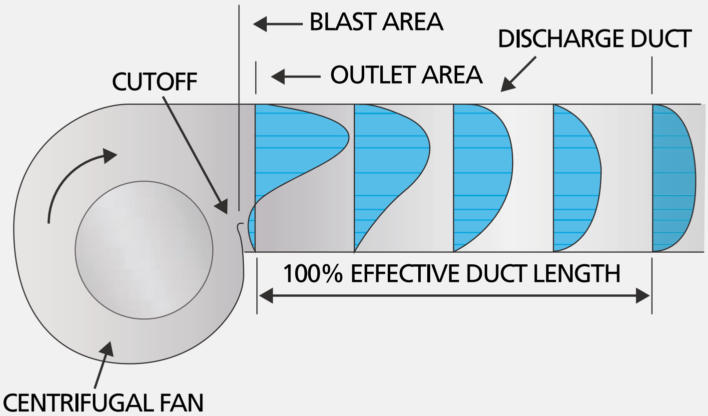
	Initial Conditions	New Conditions
CFM=	6590	8195
SP=	1.76	2.72
RPM=	1285	1598
HP=	4.66	8.96
AMPS=	6.1	11.73
Motor FLA=	6.55	
Motor HP=	5	



Step 2: Calculate the effect of a free outlet discharge.

The CP blowers are tested as free inlet ducted outlet fans consistent with AMCA standards. The amount of duct on the discharge of the fan is equal to 2.5 discharge diameters less the effect of any discharge damper.

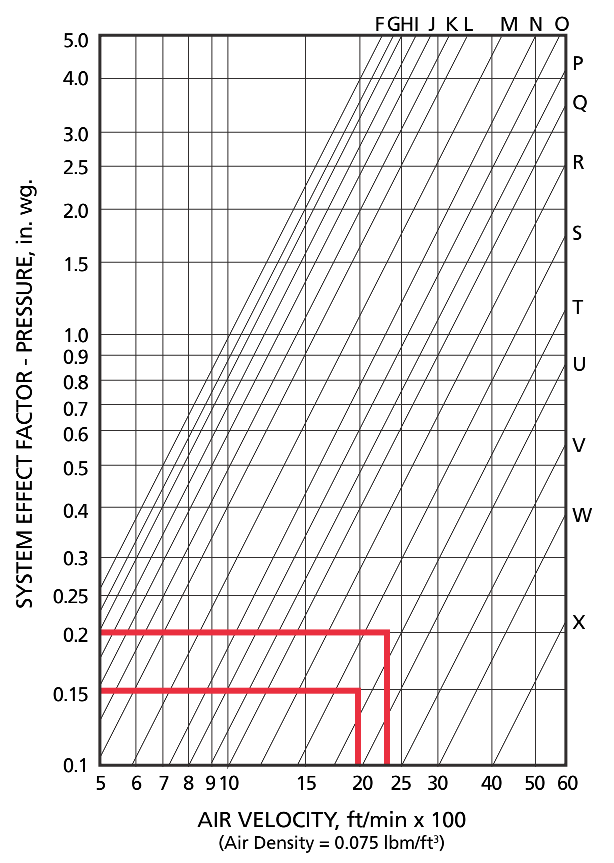
Calculate the system effect factor without the discharge duct. Using the chart below, add 0.15” based on 6590 CFM (1890 FPM) or based on 0.20" based on 8,195 CFM (2350 FPM).



To calculate 100% duct length, assume a minimum of 2 1/2 duct diameters for 2500 fpm or less. Add 1 duct diameter for each additional 1000 fpm.

EXAMPLE: 5000 fpm = 5 equivalent duct diameters. If the duct is rectangular with side dimensions a and b the equivalent duct diameter is equal to (4ablϖ)$^{0.5}$.

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Pressure Recovery	0%	50%	80%	90%	100%
Blast Area / Outlet Area	System Effect Curve				
0.4	P	R-S	U	W	-
0.5	P	R-S	U	W	-
0.6	R-S	S-T	U-V	W-X	-
0.7	S	U	W-X	-	-
0.8	T-U	V-W	X	-	-
0.9	V-W	W-X	-	-	-
1.0	-	-	-	-	-



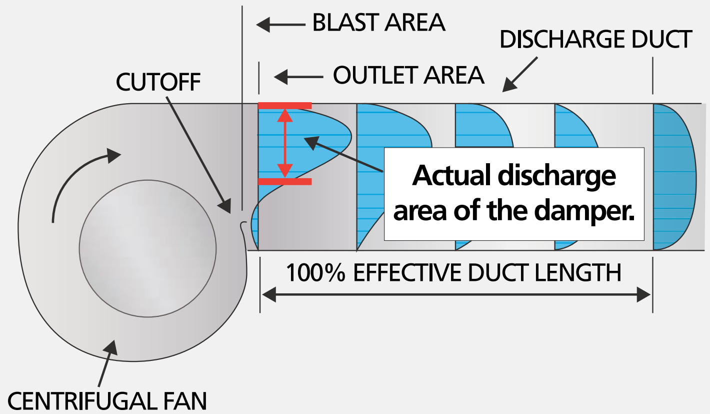
Step 3: Calculate the effect at the discharge damper.

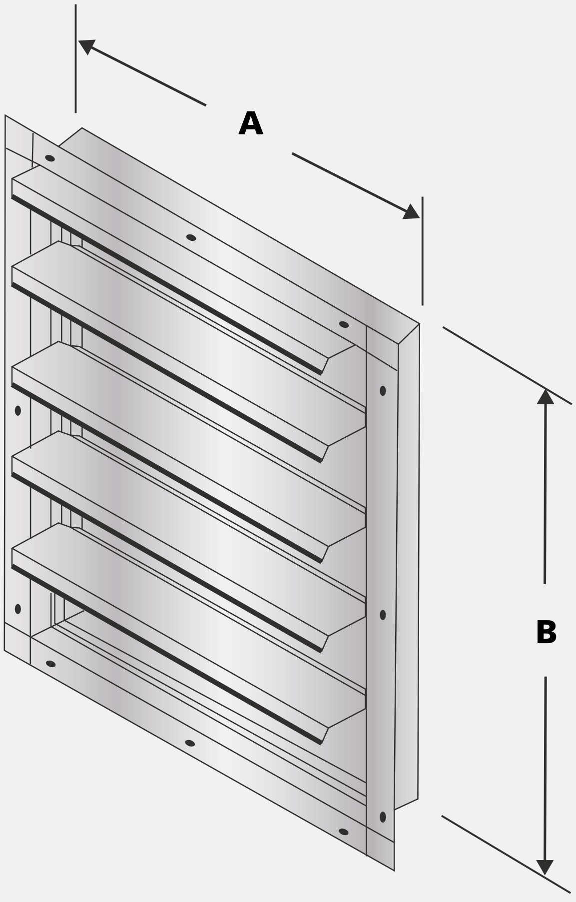
The CP blowers are tested as free inlet ducted outlet fans as previously noted, with no accessories. When the dampers are added to the discharge, the effects are multiplied several times beyond simply looking at the pressure drop of the damper.

The damper data is based on testing with the air flow laminar across the entire damper surface area. By locating the damper immediately at the discharge of the blower, the effective damper surface is reduced by one-half due to the velocity profile shown below.

On subject fan 245 CPS, the outlet area is 26 13/16” (27”) x 18 ¾” (19”). Reduce the surface area by half for the calculation and the actual damper area is 13.5” x 19”. The velocity is 3700 FPM based on 6590 CFM and 4600 based on 8195 CFM.

The highest published pressure drop on the damper is 0.39” based on 3000 FPM shown below. As a result, the calculated effect is probably between 0.5” and 0.75”.





Air Velocity (FPM) Through Damper Area	Static Pressure Drop (Inches W.G.)
600	.110
800	.110
1000	.075
1200	.070
1500	.100
1800	.140
2000	.180
2500	.290
3000	.390

S3G PRESSURE DROP

Based on 36" x 36" (914 x 914) unit.

Step 3: *Continued*

The chart below reveals the multiplied effect. The problem is that the data is based on the ventricle mounted opposite to the blade damper. We are using a horizontal back draft damper so the multiples could be as high as 2.5 to 3.0 times greater than the catalog pressure. The actual factors could be:

3700 FPM based on 6590 CFM (0.5 x 2.5 = 1.25″)
3700 FPM based on 6590 CFM (0.5 x 3.0 = 1.50″)
4600 FPM based on 8195 CFM (0.75 x 2.5 = 1.875″)
4600 FPM based on 8195 CFM (0.75 x 3.0 = 2.25″)

Blast Area Outlet Area	Pressure Drop Multiplier
0.4	7.5
0.5	4.8
0.6	3.3
0.7	2.4
0.8	1.9
0.9	1.5
1.0	1.2

Step 4: Add the potential calculated system effect loss resulting in the lack of discharge duct and the effect of the discharge damper.

SP from inlet of the fan	1.75″
SEF based on no duct	0.15″
SEF based on the discharge damper	1.50″
Total	3.40″

Based upon data from this case study, the fan manufacturer determined that the motor previously installed with this fan (5 hp) needed to be increased to a 7.5 hp motor. The following is the new submitted fan data:

Design CFM = 8195
Design SP = 2.72″ w.c.
Design Fan RPM = 1598
Design Motor HP = 7.5

"Hospitals are especially sensitive to duct leakage because of large airflow rates, room pressurization requirements, and ceiling spaces not normally utilized as return air plenums."

The Importance of Duct Leakage Testing in Hospitals

Frederick A. Seed, TBE
Arizona Air Balance Company

The necessity of duct leakage testing is a constant topic of debate, as perspectives on permissible levels of leakage vary. From a test and balance perspective, the hope is that all duct systems are tested for potential leaks. Laboratories and hospitals are especially sensitive to duct leakage because of large airflow rates, room pressurization requirements, and ceiling spaces that are not normally utilized as return air plenums.

Hospital specifications usually state that a medium pressure supply air duct will be tested for leaks. An AABC or SMACNA test standard may not be specified, but there may be at least a short paragraph defining some of the testing parameters. While the specifications might imply that all duct systems will be tested, they are most likely referring only to medium pressure ducts since testing of all duct systems is not typical. The installing contractor has probably not included the cost of sealing all duct systems to AABC or SMACNA leakage standards, which would include the cost of hundreds of duct leakage tests, corrective sealing, and retests.

As important as it is for a medium pressure supply air duct to be reasonably free of leaks, it is absolutely essential that exhaust air systems be as leak-free as possible. If value engineering must cut costs, the low pressure supply air duct downstream of the zone volume control unit could be sealed, as could return air duct run-outs, and neither would need to be leak tested. However, the return air mains and all duct work in a shaft or chase should be independently tested and certified.

Some justifications for "airtight" exhaust duct systems include the following:

- Energy: Unlike supply and return air leakage, which remains within the building envelope, all exhaust air leakage is wasted to the outside, so additional make up air is required to compensate, and it needs to be conditioned if outside air is not at a favorable temperature. The increased motor energy needed for the exhaust fans to transfer the wasted air is continuous for the life of the building.
- The exhaust fans are usually limited in additional capacity.
- The duct may be extensive in distribution and have a long route to the roof, resulting in a high ratio of duct material and joints to airflow rate.
- A duct run-out may extend 40 feet to pick up 50 CFM from a janitor's closet or a remote restroom. Any leakage will bite into that 50 CFM requirement very quickly, making even 45 CFM unobtainable.
- Exhaust systems serve nuclear medicine rooms, patient isolation rooms, and small inlets such as patient room toilets. Many of those rooms are annually re-certified for room air change rates and negative pressurization, all based on adequate exhaust airflow.
- Much of the duct will be over hard-lid ceilings and not accessible for re-sealing at the time of TAB, especially during occupancy.
- The design engineering firm, hospital owner, or state health department might refuse to accept a TAB report showing deficient exhaust airflow rates.

The end result of not testing duct systems could be exhaust fans operating at maximum capability, providing 120% of the design intent, with exhaust inlet terminals proportionately balanced at 80% of the design intent. This is all too common in hospitals.

HVAC air conveyance systems are generally not meant to be absolutely air tight, unlike water pipes. However, sealed ductwork will not be sealed tight enough unless it is leak tested, the leaks are found and re-sealed, and then re-tested.

A long horizontal duct run serving patient room toilets is normally tested before the individual drops to the exhaust registers are installed. The contractor should seal all drops down to and including the register can (from the inside if a hard-lid ceiling is in place) before final trim installation. The TAB agency should confirm that this has been done. It does not require significant effort to compare a summary of capture hood readings with a traverse of the branch duct in order to verify that the drops have been sealed appropriately.

Let's consider a worse, but common case of high ratio of duct material and joints to airflow rate. If a six-inch by four-inch duct extends 40 feet to pick up 50 CFM, the AABC recommendation of 1% maximum rate will allow only 0.5 CFM leakage or 1.0 CFM at 2% maximum. This is not easily obtained, except if a six-inch diameter sheet metal pipe is used.

The SMACNA leakage standard in this example is at the other extreme: a 40-foot long, six-inch by four-inch Class 2 (inches WC) duct is 66.7 square feet of material, which results in 25 CFM allowable leakage for Seal Class C, which will kill our 50 CFM at the inlet.

If a six-inch diameter pipe is installed, the SMACNA allowance is 11.8 CFM, which is better but still registers almost 24% of the desired airflow at the inlet. Actual field leak tests will be on larger duct sections where the ratio of duct material relative to the airflow rate decreases, making the SMACNA allowance more acceptable. This example illustrates how small leakage rates will have adverse affects on exhaust air systems and why it is difficult to obtain the elusive 50 CFM requirement on that remote exhaust inlet.

Success was recorded in balancing hospital exhaust systems when all exhaust ductwork was tested according to the SMACNA standard. The preference is a flat 2% leakage maximum based on total scheduled airflow of the fan, across the entire exhaust duct system at 2" WC test pressure. Admittedly, this is difficult and expensive to achieve with rectangular/flanged ductwork. Whichever standard the specifying engineer prefers, the expectation is that all exhaust ductwork be tested independently.

During the initial project meeting, the TAB agency should insist at a minimum that all exhaust ductwork have certified leakage tests along with medium pressure supply air duct and duct concealed in chases. It is better to make duct leakage a priority at the very beginning of the project, not during actual TAB when the owner is anxious to move into a new hospital and there is a financial penalty if the facility is not opened on the scheduled date.

A commissioning agent might not address this issue up front. The contractor might obtain a change order if the specification is not clear. The contractor should not have to provide something for free, and perhaps none of the other bidding mechanical contractors included extended duct testing.

Clearly, there is a need for certified independent duct leakage testing of hospital exhaust systems. The extensive ducted distribution system, which conveys a relatively small total airflow rate to the fan, is especially susceptible to adverse affects of duct leakage. The fan might be capable of compensating for leakage, but it will lead to additional energy costs for the motor load and increased volume of conditioned make up air every day for the life of the system. Any other fix will be costly in additional construction or duct sealing within a finished building and have a high risk of delayed occupancy.

"The extensive ducted distribution system ... is especially susceptible to adverse affects of duct leakage."

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

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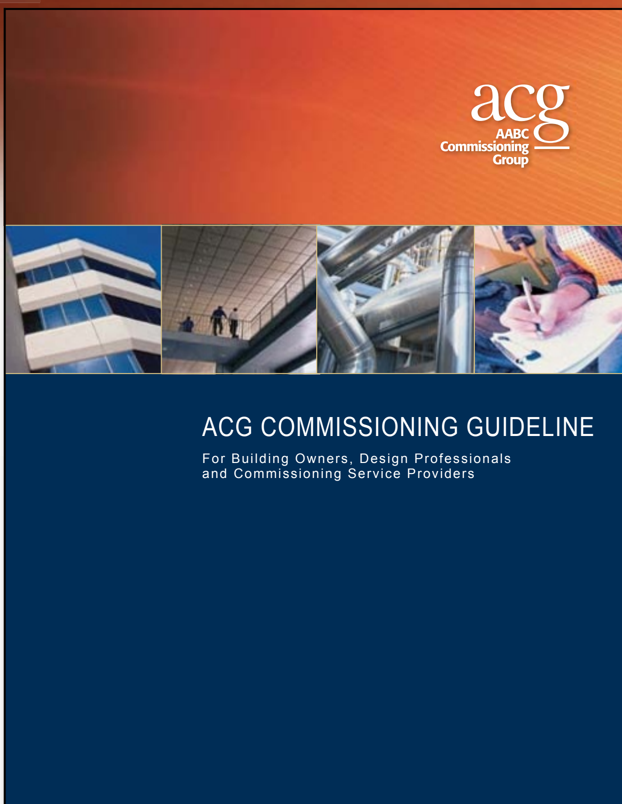

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Direct Drive Fans & Fan Banding:

"If properly designed, installed and balanced, a direct drive fan can operate for long periods of time with minimum maintenance such as filter changes. "

Branden Johnson, TBE
Mechanical Data Corporation

As mechanical engineers strive to conserve energy and increase efficiency, equipment designed for air handling systems has become complicated to use and maintain. There are motors that require grease, belts that need to be changed, and sheaves that can be adjusted to deliver desired airflow.

Recently there has been a change in direction towards lower maintenance direct drive fans. According to the engineers at AAON, the company that holds the patent to “fan banding,” a new technology in direct drive fan maintenance, the added cost of a belt-driven fan can be almost $20,000 over the typical 17-year lifespan of a 10-ton unit with a three horsepower motor. These costs include belt drive loss, bearing maintenance, and belt and bearing replacement.

If the end user does not have the personnel to service the belts and motors, installing direct drive equipment is an alternative. If properly designed, installed and balanced, a direct drive fan can operate for long periods of time with minimum maintenance such as filter changes.

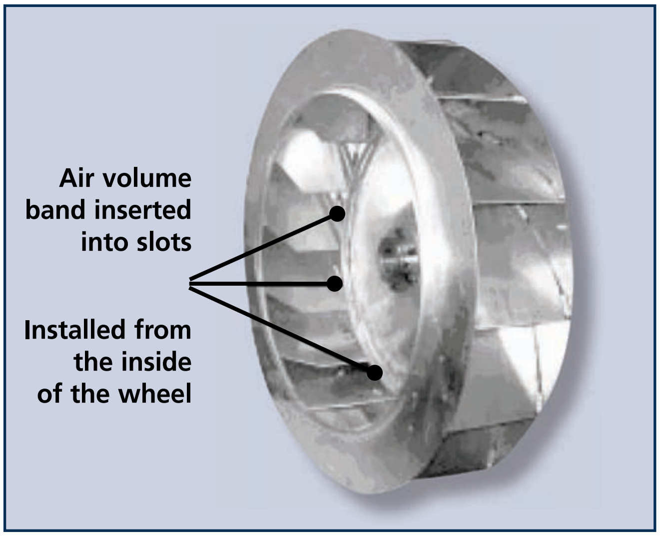
Direct drive motors are being installed on exhaust fans and supply fans and have the ability to create a variety of issues for the test and balance professional. If there is an error in sizing the motor or calculating system static pressure causing the system to operate below design airflow, a motor change or complete fan replacement will be required.

If the system is operating above the design airflow, there are a number of fairly easy solutions. Using balancing dampers is an example. This method adds static to the system, which will cause the motor to work harder, taxing it unnecessarily over time. If there is a frequency drive, a maximum limit can be programmed to operate the speed of the motor.

A New Direction for Air Handling Systems

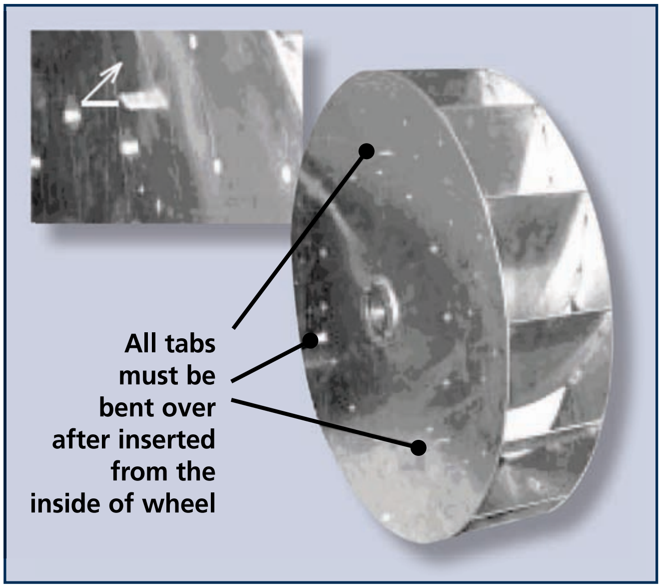
Plenum fan banding, or simply fan banding, is a third method of reducing airflow for direct drive plenum fans. Fan banding is the installation of a thin strip of sheet metal inside the fan wheel, which decreases the usable width of the fan blades, resulting in reduced airflow. *(See fig. 1)*

FIGURE 1.



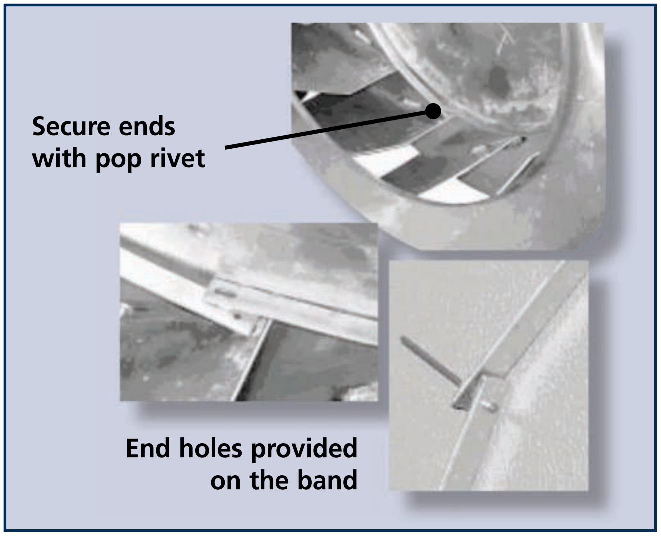
The AAON bands have tabs that are inserted through the back of the fan wheel and then bent over to secure the band to the wheel. *(See fig. 2)*

FIGURE 2.



The last step is to secure the ends of the band together with a pop rivet. *(See fig. 3)*

FIGURE 3.



The fan bands are typically installed by factory-authorized service contractors. There are, however, mechanical engineers who view banding as similar to changing belts and sheaves. The obvious difference is the inability to calculate the size of the equipment being installed because it is provided by the manufacturer. At any rate, knowing the latest methods of controlling fan speed can only benefit test and balance professionals.

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U.S. Patent No. US6,929,452 B1

Effective Fire System Testing

Ray Armstrong
Mechanical Test & Balance

Testing fire life safety systems is a challenging task in any setting. Whether you are working on a downtown high-rise building or inside a shopping mall, the goal is the same.

For an air balancer, testing duct detectors and smoke fire dampers is routine. When conducting a test in a larger space, active and passive areas may also require testing. This can include escape corridor and refuge area testing as well as stairwell pressure testing.

Smoke Detectors

Smoke detectors sense smoke particulates through ionization or optical sensitivity or a combination of the two. With the ionization method, a minute amount of a radioactive element is used. Two electrodes in an ionization chamber make a small, steady current. When smoke enters the chamber, alpha particles are absorbed, which interrupt the electrical current and trigger the alarm.

Optical detectors are based on the use of light sources and light sensors. In the absence of smoke, the light beam passes over the sensor undisturbed. When smoke is present, some of the light is scattered, which triggers the alarm. When set off, the smoke detector will shut down the associated mechanical system and/or activate the smoke control system.

Area detectors are commonly found throughout commercial and residential buildings. They sometimes are used in connection with the mechanical system like duct detectors are. Usually they have a sensing tube and a static tube. The sensing tube contains a series of holes on one side of the tube that is aimed in the direction of the airflow.

The duct detector should be installed in a system with a capacity of 2000 CFM or greater. When measuring the differential pressure, it is good practice to measure each tube individually. By doing this, the readings will reveal whether the tubes are in an appropriate location and whether they are obstructed. Some duct detectors do not use sensing tubes, but use an open head instead. Open head detectors can be prone to debris buildup, false alarms or loss of sensibility.

Measuring the airflow velocity across this device can be done with a pitot tube. It is important to verify that the device is installed in the proper location. Live smoke, aerosol can smoke, and supplied magnets can be used to activate the detector head. Some municipalities may require that live smoke be used to simulate an alarm. After the device detects smoke, the air handler should shut down within 15 seconds and the associated smoke fire dampers should be activated.

Smoke Fire Dampers

Smoke fire dampers are typically installed in a fire-rated wall. These dampers vary from having a motorized actuated damper or fusible link. The fusible links need to be clean, unpainted and free of grease. Chain or cable, s-hooks, and eyes must be in good condition. Kinked, pinched, twisted or inflexible chains or cables are not acceptable.

Poor or improper installation is common on fire dampers. Typical firewall penetration installations require flanges to be placed around the perimeter of the device with fasteners in the damper housing not more than six inches apart and not fastened to the wall. A gap needs to be maintained between the damper housing and the adjoining wall, normally

from Door to Door

measuring one-quarter inch per foot to allow expansion from heat.

If attached to ducting, the duct needs be able to break off the damper housing so the fire-rated wall can maintain the barrier. Access must be available for visual inspection of the damper's full range of movement and in order to record the unit tag information and the UL number. Access should be wide enough to permit inspection and maintenance of the unit's moving parts.

When the damper is placed in alarm mode, the device needs to fully open or close within 15 seconds. Another option may require that the damper have an end switch installed that sends a signal to the fire control panel. It is important to read the manufacturer's manual in order to ensure the installation and functional test comply with the instructions.

Smoke Control

Smoke control is an important consideration, so some basic aspects that relate to air balancers should be reviewed. Pressure fans are used to induce smoke-free air into an egress area. Stairwells, elevator shafts and exit corridors are pressurized. The stairwell should maintain a minimum positive pressure to meet all building codes, and all vestibule doors should be closed. Exit doors must open when force is applied.

Smoke relief fans are designed to remove smoke mechanically. When an alarm is activated and the smoke control fans are engaged, they usually work in conjunction with the rest of the fire life safety system. A typical test is to fill an area with theatrical smoke and then activate the system. One element of the test should measure the time required for the smoke to clear out. Pressurization, airflow or exhaust methods of mechanical smoke control systems need to be under automatic controls.

Carbon Monoxide Detectors

Carbon monoxide (CO) detectors are expected to go into alarm mode based upon designated carbon monoxide concentration levels versus time. For example, concentration amounts in 70 ±5 parts per million (ppm), the alarm should trigger between 60 – 240 minutes. At 150 ± 5 ppm, 10 –50 minutes and at 400 ±10 ppm, the alarm should sound from four to 15 minutes.

If the CO detector is connected to the fire life safety control system, the system alarm has 200 seconds to sound after a signal has been sent from the sensor. A qualified technician can conduct functional testing by following the detector's installation and maintenance instructions. Functional testing will apply to detectors installed after January 1, 2012. Sensitivity testing will begin after January 1, 2015.

Bear in mind that children, pregnant women, the elderly and individuals with certain health conditions can be affected by carbon monoxide gas sooner and at lower concentrations. At 200 ppm, the National Institute of Occupational Safety and Health recommends immediate evacuation. Standards for carbon monoxide safety are still in development.

Headache and dizziness can occur at only 35 ppm within six to eight hours of exposure to the gas. The short term effects associated with exposure at higher concentrations (1600 ppm or .16%) can lead to unconsciousness and possible death within two hours. The human body cannot use or dispose of this gas as easy as it can oxygen or carbon dioxide because carbon monoxide's affinity with hemoglobin is approximately 230 times stronger than it is with oxygen.

Not All VFDs Are Created Equal

Timothy J. Demchuk, TBE
Precision Air Balance Company, Inc.

While the normal operating range for most VFDs is from 0 to 60 Hz, some are capable of operating beyond 60 Hz.

Variable frequency drives (VFD) are an effective way to save energy every day in contemporary buildings because they limit excessive energy waste inside the facility. VFDs are a vital component in the movement toward greener buildings, and are consequently becoming more common in the industry.

A VFD regulates the speed of a three-phase AC electric motor by controlling the frequency and voltage of power delivered to the motor. When a VFD starts a motor, it initially applies a low frequency and voltage. The starting frequency is typically two hertz (Hz) or less, preventing the high jolt of a current that occurs when a normal motor is started. Depending on how the unit is controlled, the VFD ramps up the motor incrementally until either the desired RPM or the desired static pressure is met.

The normal operating range for these devices is from 0 to 60 Hz. There is a linear relationship between the frequency and voltage that VFDs control. For example, at 30 Hz, 230 volts are being delivered and at 60 Hz, 460 volts are being delivered. Technicians in the test and balance field typically work between this 0 to 60 Hz range every day. While there are a few different types of VFDs, most operate within this range.

A project at a facility called the Cleveland Clinic required testing and balancing an air handling unit in a penthouse that was designed to supply 20,000 CFM to several operating rooms below. With all the VAV boxes calibrated and set to their maximum positions, the unit was found deficient in airflow and unable to meet the necessary designed static pressure when it was at 60 Hz.

After a thorough investigation, including multiple phone calls to the mechanical contractor, the controls contractor, and the supplier, it was discovered that VFDs made by this particular manufacturer were capable of operating beyond 60Hz.

One can refer to the VFD manual for information on how to change the maximum Hz, but it is typically changed in the parameters section of the VFD's on-board interface. Before adjusting the maximum Hz, it is necessary to check the motor's nameplate amperage to make sure the installed motor is capable of handling an increased workload.

In conclusion, when a unit is underperforming according to design specifications, consult with the manufacturer's representative to determine the device's full potential. A VFD could be capable of exceeding an estimated normal range of operation.

Image: Wikipedia User C J Cowie

Tech Talk

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

Have a Question?

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

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

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

Flow Hood vs. Anemometer for Exhaust/Return Measurements

Q

QUESTION: ***I have been reading through an engineering spec. that states a rotating vane anemometer be used on return and exhaust system (grilles). The company I work for has been using a Shortridge flow hood with a digital multimeter for the past 15 years. In speaking with my boss, we have come to the conclusion that regardless of what device you use, you will need to come up with a correction factor of some sort (by doing a traverse). I cannot find an argument for use of one over the other for exhaust/return.***

A

AABC: We use both instruments. Each has to be verified by traverse and a correction applied as necessary. We use the flow hood whenever we can. In cases where it is physically impossible to use a hood, we use the anemometer. Additionally, in low flow cases of less than 100 CFM, definitely below 50 CFM, we find that the anemometer provides a more accurate measurement.

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

When and Where to Use Volume Dampers

Q

QUESTION: ***Does AABC have any written information regarding splitter dampers and/or balancing dampers installed between the air handlers and the inlet of VAV boxes?***

A

AABC: Chapter 6 of the Current Standards states that splitter dampers offer little volume control. They should not be used anywhere, in my opinion. Chapter 8 states if VAVs are pressure dependent, volume dampers are required at the inlet of each VAV to balance.

If you are referring to pressure independent VAVs, while the standards do not state it, volume dampers should not be installed at the VAV inlets or in the branch ducts 99.9% of the time. The only time a volume damper would be warranted would be if the inlet pressure at a VAV could exceed its static pressure rating. In that case, the volume damper would be used to reduce the inlet static to a point where the VAV could maintain air flow under its own control.

— *Joseph E. Baumgartner*

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(952) 807-6744

Mechanical Data Corporation
Bloomington, Minnesota
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Mechanical Test and Balance Corporation
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SMB of Minnesota
Blaine, Minnesota
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Coastal Air Balance Corp.
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Precisionaire of the Midwest, Inc.
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(816) 847-1380

Senco Services Corporation
St. Louis, Missouri
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Mechanical Test and Balance Corporation
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National Air Balance Company, Inc.
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Penn Air Control, Inc.
Las Vegas, Nevada
(702) 221-9877

Raglen System Balance, Inc.
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RSAnalysis, Inc.
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RSAnalysis, Inc.
Reno, Nevada
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Palmetto Air and Water Balance, Inc.
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Palmetto Air and Water Balance, Inc.
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Palmetto Air and Water Balance, Inc.
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Kahoe Air Balance Company
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Texas Test & Balance
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& Balancing Ltd.
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Kanata Air Balancing &
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