TAB Journal

AABC

THE MAGAZINE OF THE ASSOCIATED AIR BALANCE COUNCIL • WINTER 2015

Instrumentation & Equipment for Proper Balancing

IN THIS ISSUE:

- Sheave and Belt Changes
- Differences in Digital Multimeters
- Micron Filters

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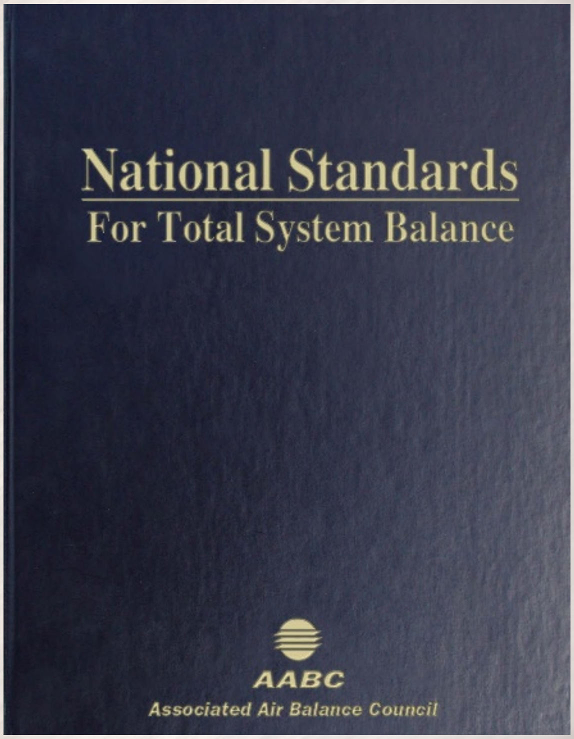

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From the Publisher

The winter 2015 issue of *TAB Journal* looks at some of the instrumentation and equipment encountered in testing and balancing projects. Mike Van Weichen, TBS of *Airwaso Canada Inc.* discusses the use of micron filters in filtration systems.

Benjiman J. Link, TBE of *United Testing & Balancing, Inc.* goes into the importance of correct airflow monitoring stations and the problems that can arise if these are not maintained and cleaned properly.

Also in this issue, Douglas R. Meacham, TBE of *Kahoe Air Balance Company* looks at how room construction can affect the design pressure criteria.

In “Project Specifications: Friend or Foe?” the Baltimore Air Balance Company makes the case for a thorough review of plans and specifications before beginning any project.

Don Burk, TBE of *PBC, Inc.* discusses who should bear labor and material costs in test and balance projects that require fan sheave and belt replacements.

Lowell T. Hedrick, Jr., TBE, CxA and Patrick E. Young, TBE, CxA of *Test and Balance Corporation* compare the differences between true RMS and non-true RMS digital multimeters in digital volt-amp meter measurements.

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!

The system pump flow rate should include the filter flow requirement. The result will be flow rates at the terminals not being compromised by keeping the system clean.

The Dirt on MICRON FILTERS

Mike Van Wiechen, TBS
Airwaso Canada Inc.

Many hydronic systems have filtration systems in place that clean and filter the water. Typical systems use strainers and micron filters to achieve cleanliness.

Strainers are usually found at the pump inlet. Mesh size may be 1/8 inch. Startup strainers are used during the filling, flushing and cleaning stage to remove construction contaminates.

Initial TAB system pre-checks require the TAB technician to inspect this strainer to ensure it is clean, and that the correct mesh size is in place.

Micron filters are either installed side stream to the pump, or more commonly across the pump with flow from the pump discharge piping, to filter, to pump suction piping *(Figure 1)*. Typically, they are installed near the chemical feed pump and share common piping. Ball valves isolate the filter for easier cartridge replacement.

Cartridges are made with a variety of materials (Cotton, Glass, Nylon, Polyester, Rayon), and with different pore sizes (1 to 150 microns). Most common are cartridges made of Cotton string, wound on a 2 ½" diameter by 10 inch spool and have a retention from 5 to 20 microns. Pressure drops can vary from 1 psi to 20 psi depending on flow rates and particle loading.

During the TAB balancing of the system, the filter is normally isolated from the system. Design engineers usually size the circulation pump flow rate, for the design sum of the terminals (coils, convectors, radiant panels, chilled beams, etc.).

No consideration is usually given for the flow rate required for the filter to keep the system clean. Micron filter flow may divert 10% or more of the pump flow rate. We have never encountered a project where an engineer had specified the filter flow rate, or sized a system pump to include filter flow rate.

In some cases a CBV (circuit balance valve) was installed in the micron filter piping, but this will not maintain a constant flow rate as the filter loads.

In other instances a sight glass (with floating ball indicator) was installed at the outlet side of the filter. Maintenance staff can manually adjust one of the isolation valves daily to ensure flow.

The best solution would be a sight glass and a pressure independent flow control valve (automated balancing valve). This would be selected based on the required flow rate of the filter, and be installed on the downstream side of the filter. With this set up, flow rate remains constant as the filter loads and pump motor VFD ramps up and down.

The maintenance mechanic has an indicator of when the filter is fully loaded when the sight glass is not indicating flow. Once again, the system pump flow rate should include the filter flow requirement. The result will be flow rates at the terminals not being compromised by keeping the system clean.

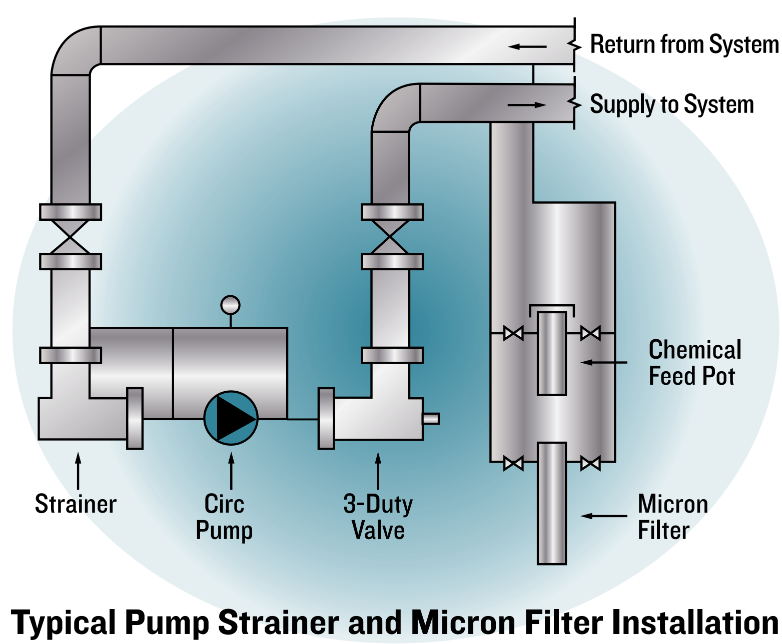

Figure 1.

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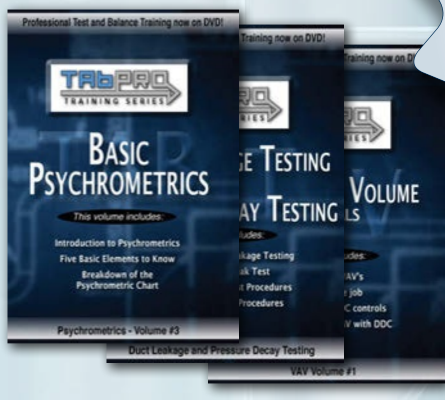

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The Importance of Correct Airflow Monitoring Stations: A CASE STUDY

Benjiman J. Link, TBE
United Testing & Balancing, Inc.

How does a healthcare facility that has been occupied only seven months, balanced and recalibrated by the same TAB agency have such severe pressurization problems?

With the high cost of investigative testing and troubleshooting, the vital role of Airflow Monitoring Station cannot be overlooked. The following example may help TAB agencies, Mechanical Engineers, and Owners find a simple fix to a serious problem.

A healthcare facility requested assistance with a problem where spaces had become improperly pressurized with high humidity levels, leading to excessive condensation. The ceiling tiles throughout the facility were in need of replacement, sometimes on a daily basis, due to staining from the excessive moisture. Overall, the facility was in an uncomfortable condition.

The initial visit confirmed the owner and occupant's complaints – an isolation room had water dripping from the ceiling and was under an extreme negative pressure. Initially, the temperature set point had been lowered in the space to help with patient comfort, but the room had to be vacated due to the conditions. Above the ceiling, all of the exposed ductwork was covered in condensation.

The second room observed had the same pressure and humidity issues as the first. The supply duct in the area had condensation which filled the duct-mounted smoke detector casing. The relative humidity level measured above the ceiling was 81%. At this point, condensation was dripping from the duct through the ceiling to the floor.

The building was relatively new and had been completely balanced within the last year. It is equipped with an Energy Management System (EMS) where multiple values including air quantities, temperatures, etc., could be monitored. During initial review of the EMS, all of the control set points, outside air volumes, and unit operations appeared to be correct. The sum of the outside air introduced to the building was 42,000 CFM, in accordance with the original design. All of the discharge air temperatures were maintaining set points, and there were no other apparent issues that could cause the types of problems observed in the patient rooms.

Further investigation revealed the area of the building with the most prevalent moisture issues was -0.06" w.c. in relation to the atmosphere. This information lead to examination of the associated RTU. The outside air design quantity for the unit was 7,000 CFM. The same correction factors recorded in the TAB report were still in place and had not been changed in the Airflow Monitoring Stations (AFMS). The visual displays for the AFMS confirmed the airflow offset was 7,000 CFM between the supply and return fans, and the EMS confirmed the same information.

Velocities at the fresh air intake were found to be very low. A suggestion was made to the owner that the AFMS should be recalibrated using the same method performed for the original TAB project. After recalibration, the space was +0.03" w.c. in relation to the atmosphere.

Although the issues with the RTU had been corrected, whenever adjacent doors were opened to other areas in the facility, the negative conditions would present themselves again. Following a complete pressure survey of the facility, it was found that all areas had negative pressure problems related to AFMS that were not controlling to the correct outside air quantity.

It was finally recommended to the owner that all of the AFMS be cleaned and recalibrated. When this was completed, the overall building was positively pressurized at +0.03" w.c. The pressure relationships between different areas of the building were correct. Room pressure monitors were now within normal ranges, humidity levels were as expected for July and there were no problems with condensation.

So how does a healthcare facility that has been occupied only seven months, balanced and recalibrated by the same TAB agency have such severe pressurization problems?

The particular airflow sensing devices for this facility are glass bulb thermal dispersion sensors. During the cleaning of the flow stations for the recalibration purposes, some of the bulbs were found to have a buildup of dirt and lint. This continual buildup was causing incorrect airflow

CORRECTION FACTORS		
RTU #	Original Correction	Area Correction
RTU-1 (Supply)	2.83	2.83
RTU-1 (Return)	2.17	2.56
RTU-2 (Supply)	4.07	4.55
RTU-2 (Return)	4.17	5.00
RTU-3 (Supply)	2.47	2.07
RTU-3 (Return)	1.40	1.62
RTU-4 (Supply)	4.57	3.63
RTU-4 (Return)	3.03	3.75
RTU-5 (Supply)	8.60	6.16
RTU-5 (Return)	4.33	5.03
RTU-6 (Supply)	4.70	4.70
RTU-6 (Return)	4.15	3.90
RTU-7 (Supply)	1.44	1.78
RTU-7 (Return)	1.23	1.52
RTU-8 (Supply)	1.96	2.34
RTU-8 (Return)	1.85	1.99
RTU-9 (Supply)	5.63	6.44
RTU-9 (Return)	4.93	5.64
RTU-10 (Supply)	2.11	2.35
RTU-10 (Return)	1.85	1.80
RTU-11 (Supply)	4.04	3.47
RTU-11 (Return)	2.90	2.90
RTU-12 (Supply)	3.49	2.70
RTU-12 (Return)	2.26	2.34

measurements, and in this situation, a vast reduction of outside airflow. Based on all of the information gathered it was assumed that the combination of construction completion, and the introduction of new linens to the facility had caused an unusually high level of airborne particulates. This resulted in a need for flow station cleaning long before expected.

The provided chart, (*left*), shows the original TAB report correction factors and the new correction factors after the cleaning and recalibration. It was also noted that there may have been buildup of particulates, due to temporary filters and normal construction practices that affected the original TAB correction factors.

RECOMMENDATIONS

Even with a short duration of occupancy, some circumstances will require cleaning of Airflow Monitoring Stations long before an experienced TAB agency would expect. The following recommendations offer guidance to help prevent issues with AFMS.

- Always follow the manufacturer's recommendations for maintenance and cleaning of AFMS.
- Detailed inspection of flow stations for cleanliness before initial TAB is recommended.
- Periodic verification of Airflow Monitoring Stations by a certified AABC TAB agency is always a good idea.

If there is a pressure requirement, then a room construction requirement needs to be specified as well.

Issues with Specified Room Pressures

Douglas R. Meacham, TBE
Kahoe Air Balance Company

Not all rooms are built the same. Testing room pressures at a hospital, research lab, or even at a pharmaceutical manufacturing facility can reveal differences. Technicians can encounter room pressure requirements specified for a room without specified criteria as to the tightness of the room's envelope. If the room is not treated as a vessel, with all penetrations accounted for, then that room will have issues attaining any stated pressure. It is necessary for the design engineer and the architect to have the same intention on the space requirements. If there is a pressure requirement, then a room construction requirement needs to be specified as well. This can be accomplished with stating that all areas surrounding pressure zones need a rated barrier or wall.

With this requirement, attention to items—such as cinderblocks needing to be sealed, all conduit penetrations requiring sealing, walls needing to extend to the deck, and doors needing to be gasketed—are part of the original design documents. Now a calculated offset can accomplish the required function, creating a pressure relationship. With all of the energy-saving requirements that are driving the construction industry today, having a space that requires additional supply or exhaust to satisfy a room pressure will require additional energy to operate because it was not appropriately designed architecturally.

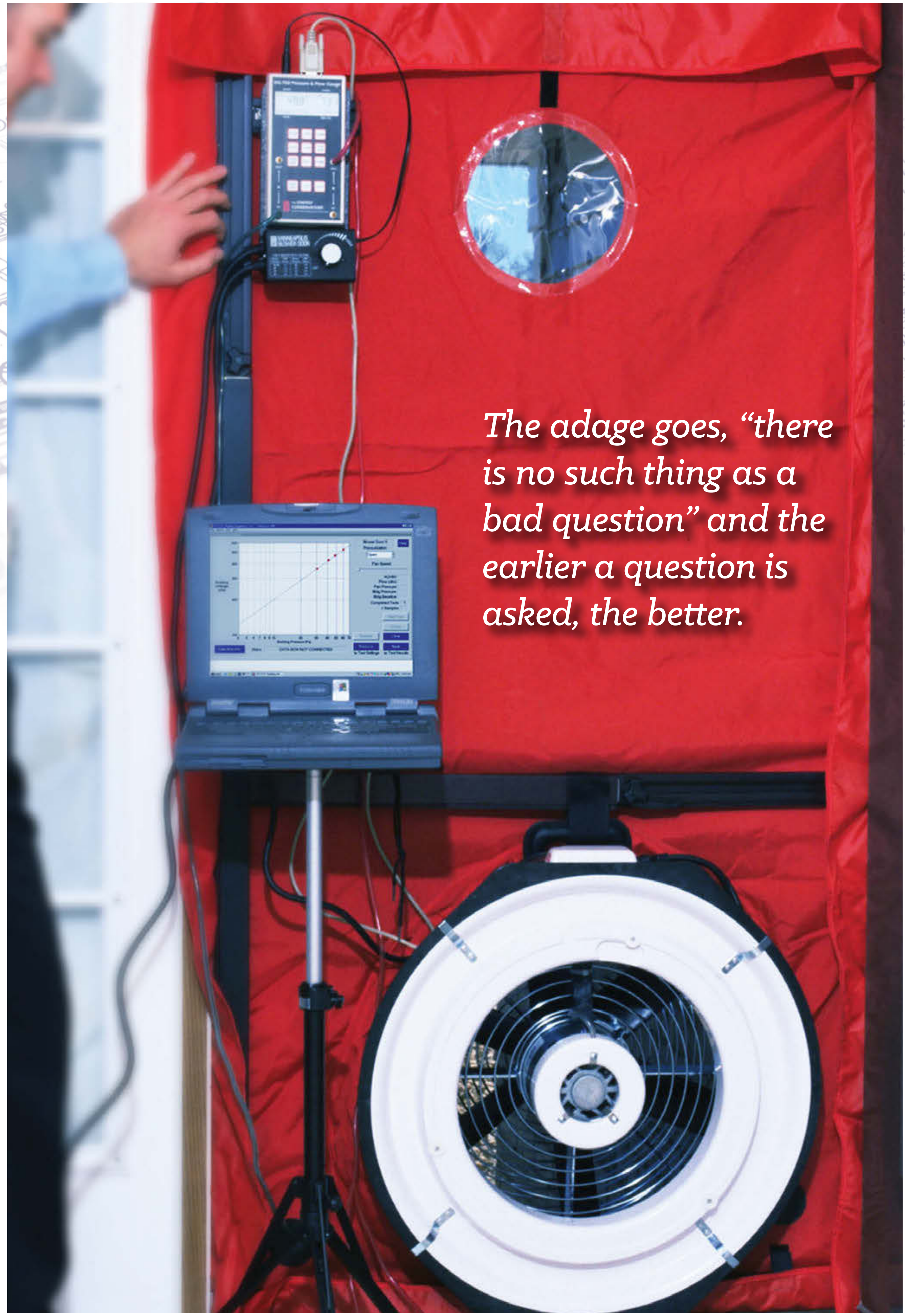
For instance, a recent project involved a lab that was designed to be negative 0.05" w.c. with a 100 CFM offset and utilizing a supply AV and an Exhaust EV for control. This room had no construction specifications other than that of a typical office wall with standard doors and an acoustical drop ceiling. To achieve a negative pressure in the room, it was directed that the supply be reduced by 10% and exhaust be increased by 10%. However, the exhaust system was already operating at its maximum speed and a 10% increase in the exhaust could not be achieved. Under the new conditions the room pressure was -0.01" w.c., with close to a 200 CFM offset. Using Blower Door, it was calculated that an offset of 354 CFM was needed to achieve a -0.05 "w.c. room pressure.

Inspecting the room and surrounding areas it was determined that the space needed to be constructed tighter. To accomplish this the facility maintenance contractor did the following:

- Installed a foiled back ceiling tile into a gasketed ceiling grid
- Added door sweeps and gaskets
- Sealed off light switches and electrical outlets

The room was re-tested using the Blower Door to verify that there were positive results after the space changes. The room achieved a -.05" w.c. room pressure with a 90 CFM offset. The space conditions were met with even less airflow making it more efficient.

It is always a good idea to verify how the room construction will affect the design pressure criteria.



The adage goes, "there is no such thing as a bad question" and the earlier a question is asked, the better.

PROJECT SPECIFICATIONS: Friend or Foe?

Baltimore Air Balance Company

For all contractors, every project should start with a thorough review of the project's plans and specifications. As TAB agencies, the plans show what needs to be balanced, and the specification (specs) gives the details on the testing, balancing and other test requirements.

A proper review of the specifications starts with the estimator. It is the job of the estimator to cover all the requirements set forth by the design engineer. After all, it is the design engineer that will be reviewing the submittals and final TAB report at the end of the project. Do not take shortcuts before a project is even started. Have everything covered that is required by the project's specifications in the proposal. With that being said, there can be some unorthodox requirements in the TAB spec. As strange as it may seem to take vibration measurements on a fractional horsepower motor, if it's in the spec, make sure it is covered in the proposal. When bringing attention to something provide separate pricing for the testing that is deemed "over the top". That way the requirements of the project are accounted for.

The next step begins when a project is awarded. Again review the plans and specs and make sure the team understands what is required. Many specs call for qualifications to be submitted along with the procedures used to test and balance each system. The AABC certifications cover all the qualifications that are needed as an independent Testing and Balancing agency. The procedures are a little more involved, but do not take shortcuts if the specs require the submission of the procedures used to balance the project. Provide detailed procedures for each system on the project. Some specs ask for a detailed design review as well. It is a good practice to perform a design review even if the requirement isn't detailed in the TAB section of the specifications.

This last step before the actual balancing can save lost time at the end of the project, waiting for responses from the engineer of record on questions that could have been asked early on in the job. The main things to call out are the need for additional balancing devices, the amount of diversity in each variable volume system, differences between the total connected flow and the output of fans and pumps in constant volume systems, possible building pressurization problems, etc. The adage goes, "there is no such thing as a bad question" and the earlier a question is asked, the better.

When called to start balancing the TAB Technician should again review the specifications before actually starting any test and balance work. This will make sure it is understood what scope of work must be performed. This will also ensure the Technician has all the equipment to take the measurements that are required. The attention to detail saves time in the long run, avoiding a possible return to the site after balancing is complete.

Lastly, after the TAB work is done and the report complete, it is the Test and Balance Engineer's turn to review the specifications before certifying the TAB report. This is all in the name of providing what is required, while also delivering a quality TAB report in the most efficient way possible. If there isn't time to do it right the first time, when will there be time to do it a second time?

Is the testing and balancing specification section a TAB Agency's friend or foe? As with many things in the world, it is a matter of perspective. Someone who takes shortcuts may find the specs unreasonable when the design engineer forces them to take all the required data. However, if all the data is recorded thoroughly the first time around, specs can be a good guideline to the entire team working to provide the building owner with a quality product.

Sheave & Belt Change Issues

Don Burke, TBE
PBC, Inc.

During the process of testing and balancing air systems, it is not uncommon to find that achievement of design air requirements necessitates the changing of the fan's sheave and belts. In these cases, three questions that need to be answered often come up: **1)** Who should size the new sheave and belts, **2)** who should install them and **3)** who should pay the labor and material costs?

In the vast majority of cases, the answer to the first two questions is the TAB agency. Experience has taught that sizing new sheave and belt combinations based on actual field test results is the way to assure expediency with the fewest mistakes. Installing the new sheave and belt combination should also be done by the TAB agency, since they must retest the system afterwards.

Regarding the third question: Who should pay the labor and material costs—the answer is not as simple.

To come up with a sensible answer based on practical realities, the following postulates should be considered:

1. Most TAB agencies do not include the cost of sheave and belt changes in their price unless specifically called for by specification—i.e., existing AC-1 will require a sheave and belt change in order to increase/decrease air output to the new design requirements.

2. The reason TAB agencies cannot include the costs of sheave and belt changes in their price is simple. Many times, fans that require sheave and belt changes are relatively simple systems, systems for which the total price to balance is less than the costs of labor and material to change the sheave and belt. Even for many more extensive systems, the costs of sheave and belt changes can amount to 25% or more of the cost to test and balance the system.

Obviously, the potential cost of sheave and belt changes cannot be included in the balancing price nor imposed upon the balancing agency without an unfair financial burden upon the TAB agency.

3. Who should bear the cost of sheave and belt changes? Customarily the mechanical contractor pays these costs. Since the gross costs of a system including the air handlers, the piping, ductwork, etc., are all normally included in the mechanical contractor's price,

the cost of a sheave and belt change is usually a very small percentage of overall costs; and, therefore, not too significant either at bid time or relative to profit margin.

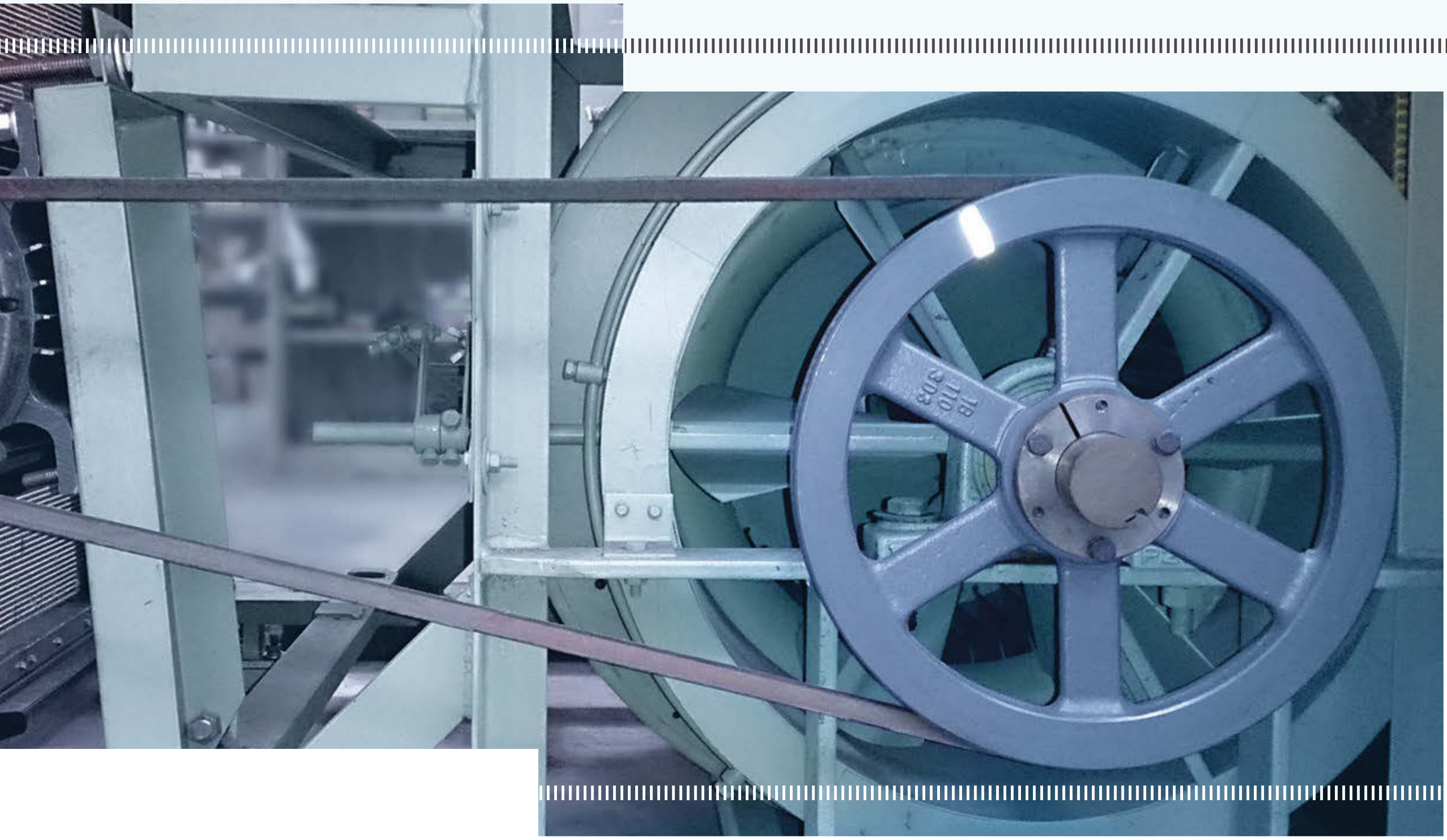
Although the mechanical contractor customarily pays for sheave and belt changes, that does not clearly answer the question of who should pay. That question has several different answers depending on the reason a sheave and belt must be changed.

Following are several examples:

a. The fan's design calls for 5000 CFM at 1.0" ESP and 1000 RPM. Tests show the fan delivers 6000 CFM at 1.5" ESP and 1200 RPM, and the adjustable sheave is at minimum adjustment. Who should be responsible for the sheave change? The fan manufacturer or his representative, because design RPM cannot be reached.

b. The fan's design is the same as in a., above. Tests show the fan delivers 4000 CFM at 1.5" ESP and 1075 RPM and the adjustable sheave is at maximum adjustment. Evidence shows that because of unforeseen field conditions, some of the ductwork had to be modified in such a way as to create more than anticipated resistance. Who should be responsible for the sheave change? Most likely the mechanical contractor, because it is for this type of situation that the design engineer wants protection for himself and the owner. The designer expects the mechanical contractor to allow for situations such as this in his price so that the owner need not be charged additionally.

c. The fan's design calls for 5000 CFM at 2.0" S.P. and 1000 RPM. Tests show the fan delivers 6000 CFM at 0.5" ESP and 925 RPM, and the adjustable sheave is at minimum adjustment. Who should pay for the sheave and belt change? If the duct system is installed essentially as designed, the design engineer should be



responsible for the sheave and belt change. If actual ESP is less than 0.5” at design air (5000 CFM) but design stated 2.0” ESP, the error is in the design.

In no case should the costs of sheave and belt changes be imposed upon the TAB agency unless it was known before bid time that specific units would require these changes, and the TAB agency was asked to include these costs in their bid.

SUMMARY

Reasons Why Sheave And V-Belt Changes Are Necessary

1. Field conditions necessitate duct changes that impose more static pressure (SP) resistance.
2. The design engineer overestimated or underestimated the SP resistance of the system.
3. The fan is shipped from the manufacturer with a sheave and belt combination that does not permit adjustments capable of achieving design RPM’s.
4. The fan is capable of running at or above design RPM and is seeing design or less than design SP, yet does not deliver design CFM.
5. The fan’s connected airflow requirement is significantly above or below the fan’s design CFM.
6. The fan is an existing system and the design engineer states in the balancing specification that sheave and belt changes, if necessary, are to be included as part of the balancing.
7. System effect factors result in CFM output deficiencies.

COMMENTARY

Except for a situation such as described in number 6, above, there is no circumstance in which the cost of a sheave and V-belt change should be imposed upon the balancing agency.

IS YOUR AMP METER LYING TO YOU?

A Field Test Experiment to Highlight the Differences of True RMS and Non-True RMS Digital Multimeters

Lowell T. Hedrick, Jr., P.E., CxA and **Patrick E. Young, TBE, CxA**

Test and Balance Corporation

INTRODUCTION

The two methods generally used in Digital Volt-Amp Meter (DVM) measurement electronics are:

1. Non-True RMS (root mean square) - AC average rectified measurement.
2. True RMS measurement.

Troubleshooting motors fed by adjustable speed controllers can be difficult if you do not utilize the right meter. Solid state motor drives and controls often conduct non-sinusoidal (distorted) current. Distorted current waveforms occur in short pulses rather than the smooth sine wave seen with a standard induction motor or a resistive heater. The measurement differences between true RMS and non-true RMS meters can affect proper motor adjustments. A non-true RMS meter inaccurately measures non-pure sinusoidal waveforms due to limitations of the electronics and poor signal representation. These meters are specified to be "average responding-RMS indicating." They capture the rectified average of an AC waveform and scale the number by 1.1 to calculate the RMS value. In other words, the value they display is not a true value, but is a calculated value based on an assumption about the wave shape. The average responding method works for pure sine waves, such as heaters and induction motors, but can lead to large reading errors up to 40 percent, when a waveform is distorted by nonlinear loads such as adjustable speed drives or computers. For this article, motor amperages will be compared using a True RMS DVM meter and a DVM using averaging rectified amperage and voltage measurements. The test will demonstrate that using a non-true RMS meter can result in "overloading" a motor above its full load amperage (FLA) eventually causing failure.

EXPERIMENT

The experiment is designed to show the differences of true RMS and non-true RMS (AC rectified averaging) digital multimeters. The test starts with a fan motor that had failed, was replaced with a new motor and placed back in operation. Measurements will be taken as the speed of the fan is adjusted by use of a single phase diode rectifying controller. This test will also show the correlation of the fan's speed, amperage and airflow. Equipment used in this experiment include:

1. A Greenheck Model CUE 121-A-G Fan, The motor is rated at ½ HP, 115 volt and 6 full load amps, equipped with Vari-Speed Controller which generated a non-pure sinusoidal waveform from the AC incoming line voltage.
2. A Shortridge Digital Multimeter with Vel-Grid attachment and static probe.
3. A Southwire 21010N AC average rectified digital multimeter.
4. A Fluke 376 True RMS digital multimeter.

Troubleshooting motors fed by adjustable speed controllers can be difficult if you do not utilize the right meter.

Meters were clamped around same conductor to measure the output current from the speed controller to the fan motor. These results were consistent and repeatable within the tested equipment and instrument tolerances, and can be used as a quick guide to differentiate between true RMS and average rectified multimeters.

The test was conducted on an oven hood fan supplied with a Greenheck CUE 121 equipped with a variable speed controller. Described below are three different methods to determine the oven hood airflow. The average of all three was calculated to be 1,492 CFM. The fan static pressure was measured at .92" w.g. at 6.0 amps and 123.5 volts. This would indicate that the ½ horsepower motor was fully loaded. Plotting the airflow data on the manufacturer's fan curve shows it does not fall properly on the curve. It is believed that this discrepancy can be accounted for in that the 10" round duct rises to the bottom of the curb where it opens to the full curb size, thus most likely causing some level of "system effect" on the fan. This does not account for the electrical issues noted below with regard to the speed controller.

Figure 1.

Measuring the same motor current with two meters: Which one is correct? The motor circuit above feeds a motor via a speed controller with a non-linear load with "distorted" current wave. The true-RMS clamp reads correctly but the average responding clamp reads low by 27 percent.

MECHANICAL MEASUREMENTS OF AIRFLOW - GREENHECK CUE 121-A-G

The velocity was measured at the 10" round inlet using Shortridge Multimeter and airflow was noted as 2,647 FPM multiplied times 0.545 sq. ft. = 1,443 CFM. Second, the 10" round duct connection was measured using a Shortridge Multimeter with the static probe as noted on hood label. The baseline value noted on the nameplate tag was that 0.40" which equates to 900 CFM. Since 1.15" w.g. was measured at the duct collar, this would calculate to 1,526 CFM. A third check was completed using the Shortridge Multimeter and the Vel-Grid attachment on the face of the perforated face of the hood intake. Avg. Velocity = 317 FPM x 4.75 sq. ft. (Intake area 57" X 12") = 1,506 CFM. The static pressure was measured at 0.92"w.g.

ELECTRICAL MEASUREMENTS

The steady state operating conditions of the oven fan motor and "Vari-Speed" controller were utilized. The motor is rated at ½ HP, 115 volt and 6.0 full load amps. Using the current triac controls, the Vari-Speed would allow operating amperage up to approximately 10 amps or 167% of rated full load amps. The motor speed/amp curve, as was measured, is an inverted "V" with "peak" amps actually ramping from high speed (6.0 full load amps) up to 10 amps as the motor is slowed down and then as the speed continues to drop down, the amps decreases to approximately 7.2 amps at the lowest speed (which is above FLA and appears to cause the motor to overheat). The Mechanical Field Engineer stated that at the rated full load amps (6.0), the fan CFM was higher than was desired based on the Design Engineer's parameters. Even though the airflow was above design, the unit was left running at the speed where the motor was operating at full load amps to prevent motor damage. The following table documents the recorded measurements.

ELECTRICAL MEASUREMENTS OF OVEN FAN

Speed	Amps / Non-Rms	Amps / Rms	Volts At Disconnect
Speed Position As Found	6.96	9.54	123.6
Lowest Speed Position	5.41	7.21	122.3
Highest Speed Position	5.10	6.89	122.5
Speed (Final Setting)	4.56	6.00	123.5

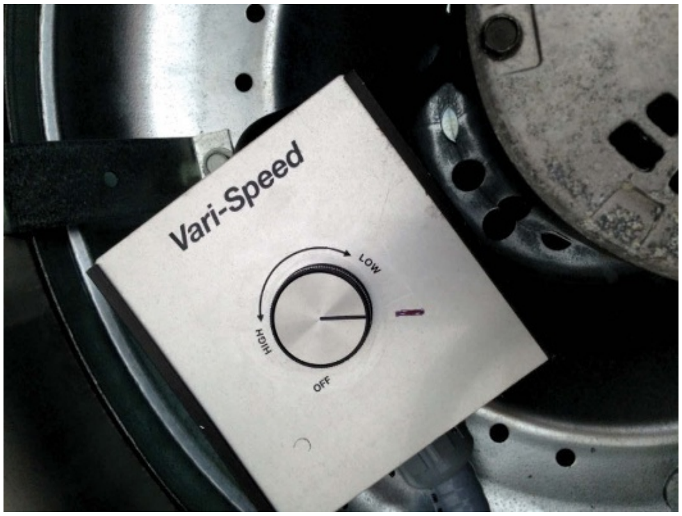


Figure 2.
Greenheck CUE 121 Fan - Fan motor with adjustable speed controller.

CONCLUSIONS

A non-true RMS meter will indicate acceptable amperage drawn when the current is actually above the nameplate rating. When measuring current from a variable speed controller, a True RMS meter must be used to obtain consistent correct voltage and amperage measurements. The correlation factors between Vrms/Arms and Vave/Aave are neither linear nor constant. This experiment illustrates the linearity between the two meter readings can be close at points, but still not consistent. A true RMS meter should be used by Field Engineers to ensure accurate, consistent, repeatable and safe measurements. The experiment also illustrates that fluctuation of incoming current does affect the output motor via speed controls. This is important because motor/fan speed affects air flow. Without confidence of a stable measurable current both the motor and the air flows are affected. This can lead to improper air flow, heat removal issues and motor damage.

A recommendation was made to consult the fan manufacturer and resolve the electrical issue involving the use of this Vari-Speed controller with the specified motor. In the "as found" condition, the fan should simply be adjusted to a "higher than required" airflow to maintain amperage at or below the nameplate rating.

References

1. *True RMS Definition, application note 106, Linear Technology*
2. *Fundamental Electrical and Electronic Principle 3rd edition by Christopher R. Robertson, Newness 2008.*
3. *Fluke Application Notes, Why True-RMS? 2002 Fluke Corporation*

ROOT MEAN SQUARE

The RMS value of a set of values (or a continuous-time waveform) is the square root of the arithmetic mean (average) of the squares of the original values (or the square of the function that defines the continuous waveform).

In the case of a set of n values, $\{x_1, x_2, \ldots, x_n\}$ the RMS value is given by this formula:

$$x_{rms} = \sqrt{\frac{1}{n}\left(x_1^2 + x_2^2 + \ldots + x_n^2\right)}.$$

The corresponding formula for a continuous function (or waveform) f(t) defined over the interval $T_1 \leq t \leq T_2$ is

$$f_{rms} = \sqrt{\frac{1}{T_2 - T_1}\int_{T_1}^{T_2} [f(t)]^2 \, dt},$$

And the RMS for a function over all time is:

$$f_{rms} = \lim_{T \to \infty} \sqrt{\frac{1}{T}\int_{0}^{T} [f(t)]^2 \, dt}.$$

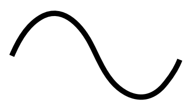
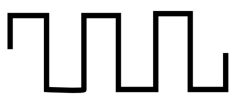
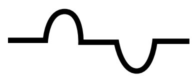
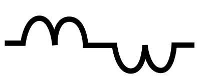
COMPARISON OF NON-TRUE RMS AND TRUE RMS METER RESPONSES				
MULTIMETER TYPE	RESPONSE TO SINE WAVE	RESPONSE TO SQUARE WAVE	RESPONSE TO SINGLE PHASE DIODE RECTIFIER	RESPONSE TO 3 PHASE DIODE RECTIFIER
Wave Form				
Average Responding Non-True RMS	Correct	10 % high	40 % low	5-30 % low
True-RMS	Correct	Correct	Correct	Correct

Figure 3.
Comparison of Average Responding and True RMS Units. Fluke Application Notes, Why True-RMS? 2002 Fluke Corporation.

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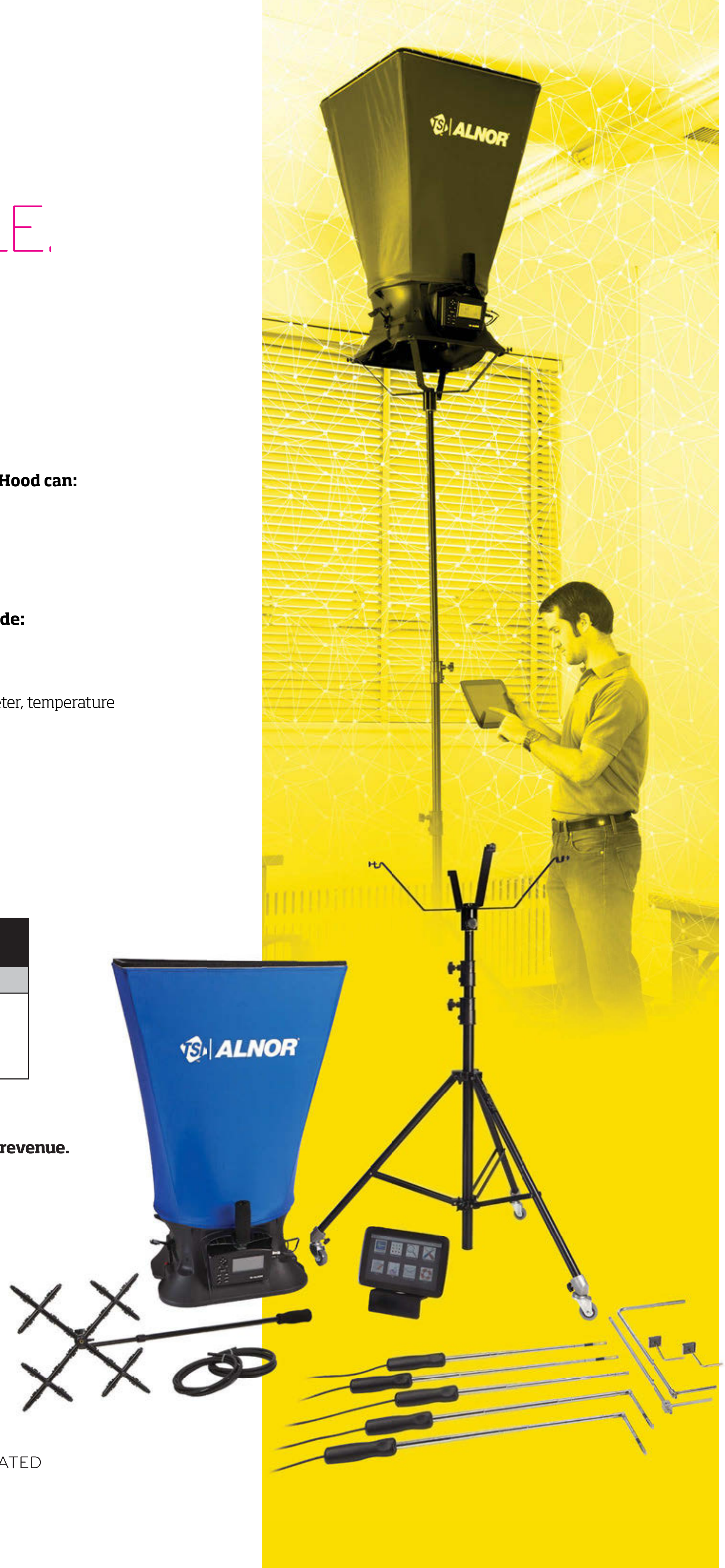

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