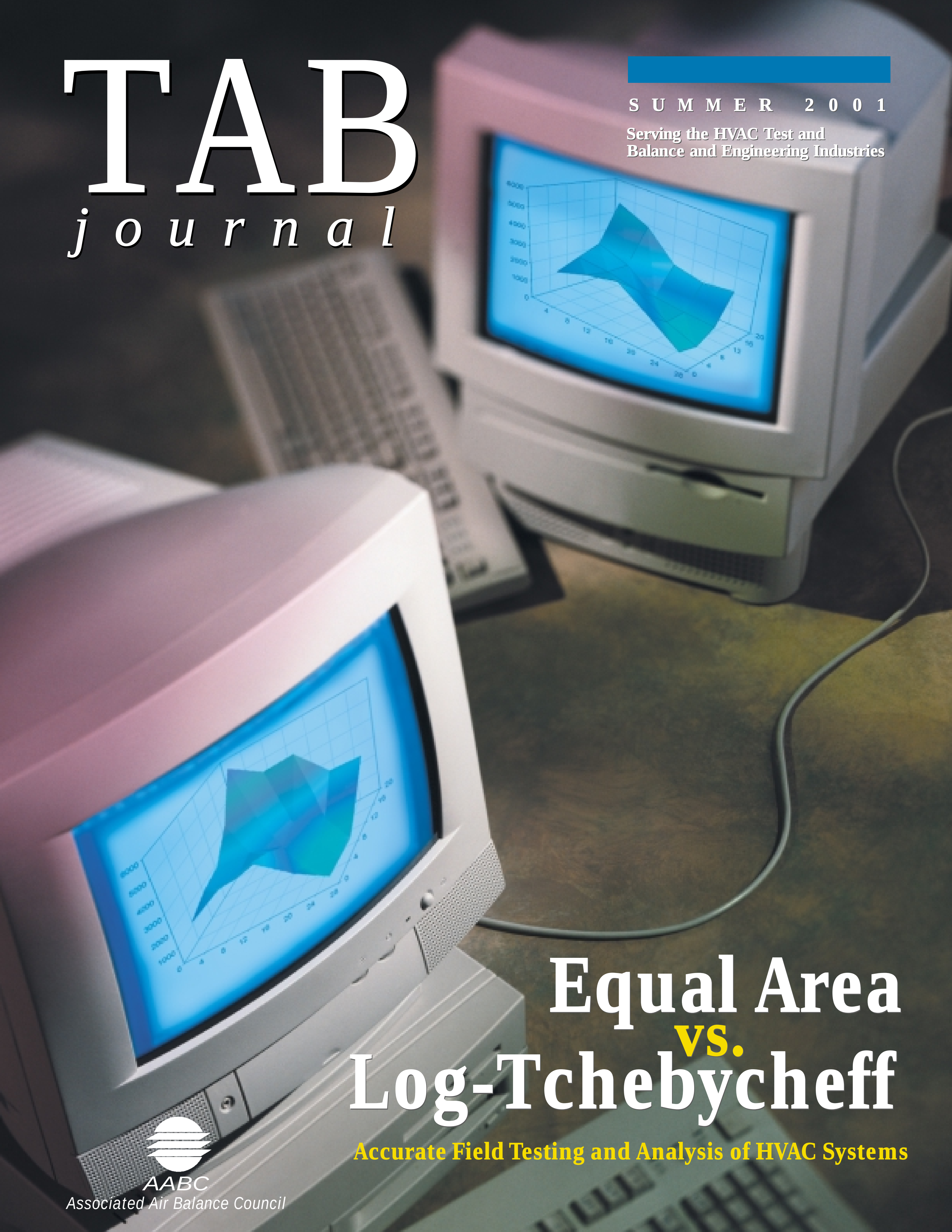
TAB journal

SUMMER 2001

Serving the HVAC Test and Balance and Engineering Industries



Equal Area vs. Log-Tchebycheff

Accurate Field Testing and Analysis of HVAC Systems

AABC
Associated Air Balance Council

TAB journal

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From the Publisher

We are pleased to present the Summer 2001 issue of *TAB Journal*. This issue delves into an ongoing debate in the test and balance community: Equal Area vs. Log-Tchebycheff. Several members have contributed articles exploring the competing methods for duct velocity traversing.

Gaylon Richardson presents two separate analyses of duct velocity traversing, in "Traversing for Flow Correction Factors" and "Traversing for Accuracy in a Rectangular Duct." Meanwhile, Joe Baumgartner offers his take on duct velocity traversing in "To Use Log-Tchebycheff or Not to Use Log-Tchebycheff…Is That the Question?"

Also included is an article reprinted from HPAC Engineering, which was written using input from AABC member Marty Pieper of Systems Testing and Balancing. Our thanks to HPAC Engineering for lending us this informative article.

This issue's *Tech Tips* section features a tip from David Parker and Bernie Moltz of Bernie Moltz, Inc., while Dean Jukam of Systems Management & Balancing, Inc. has submitted a humorous story by an anonymous writer at the University of Iowa. Also included is a follow-up from Richard Miller of Systems Testing and Analysis to an article printed in the Fall 2000 issue of *TAB Journal*.

We would like to thank those members who contributed articles for sharing their views with the *TAB Journal* readership. If you have an article or comment you would like to contribute, please contact AABC National Headquarters.

To Use Log-Tchebycheff or Not to Use Log-Tchebycheff…Is That the Question?

Joseph E. Baumgartner, III, P.E.
Baltimore Air Balance Company

Mr. Ernest MacFerran, P.E. wrote an article comparing duct velocity traverse methods that was published in the December 1999 issue of *HPAC Engineering*. The article has generated much debate. Open and honest discussion of methods and procedures is necessary for the advancement of our industry. If that was not true, buildings under construction today would be heating with one pipe steam and opening the windows for air conditioning.

That being said, misrepresentation of data, inaccurate statements, and inadequate analysis do not lay the foundation for informed discussion. Mr. MacFerran's article, unfortunately, contains all three elements.

Misrepresentation of Data

Test Configuration

Mr. MacFerran does not specifically state where the measurements he used for his analysis were taken. In correspondence to a colleague, Mr. MacFerran confirmed that the initial measurements in the rectangular duct were taken two feet from the discharge end, placing the traverse plane six feet from the round to rectangular transition. At that initial test point, Mr. MacFerran further stated that the equal area traverse method in the rectangular duct yielded the same CFM as the traverse in the round duct. Mr. MacFerran then took measurements in the rectangular duct five and a half feet from the discharge end, placing the traverse plane one and a half feet from the round to rectangular transition. At this second measurement location, Mr. MacFerran's equal area traverse of the rectangular duct did not yield the same cfm as the round duct traverse. This measurement location, one and a half feet downstream of the round to rectangular transition, is the basis for Mr. MacFerran's data and conclusions. This traverse plane location would not be acceptable in any of the publications cited by Mr. MacFerran throughout his article. "Regions immediately downstream from elbows, obstructions and abrupt changes are not suitable traverse plane locations." [1]

Velocity Point Locations

In his description of traversing a duct, Mr. MacFerran states that the Log-Tchebycheff method dictates three holes be placed in the 12" side of the duct. However, in his test measurements, Mr. MacFerran actually shows readings at five locations along the 12" side. Additional readings taken in the equal area method would also have identified the velocity distribution as it existed. While the equal area method only requires two readings in the 12" side, knowing the traverse plane location, one versed in field measurements would have increased the number of measurements to try to improve the accuracy. "If the flow conditions at the traverse plane are less than satisfactory, the accuracy of the flow rate determination may be improved by increasing the number of measurement points in the traverse plane." [2]

From further analysis of Mr. MacFerran's article, the values for the Log-Tchebycheff velocities were not even measured. Mr. MacFerran calculated the Log-Tchebycheff velocities from a graph he developed from the measured equal area velocities. How can two methods be compared if one is never actually performed? Further, the graphs do not correspond to the data and what is listed as the height in the Log-Tchebycheff table is actually the spacing algorithm.

Instrumentation

Mr. MacFerran does not indicate what instruments he used for his measurements. In the photograph accompanying the article of Mr. MacFerran's test assembly, a Magnehelic® gauge is shown on top of the duct connected to a pitot tube. A Magnehelic is a differential pressure gauge employing a dry type bellows and a calibrated spring. A Magnehelic gauge should not be used in performing a duct velocity traverse. Again—none of the references cited by Mr. MacFerran recommend this instrument for duct velocity traverses. "The instruments recommended for use in measuring velocity are a Pitot-static tube and an inclined manometer or electronic instruments of comparable accuracy."[3]

Inaccurate Statements

- Mr. MacFerran states "Through my tests, I discovered that results from the equal area method are always in error…" His initial tests do not support that statement.
- Mr. MacFerran states "…no contracted AABC or NEBB company has used or will use the Log-Tchebycheff method for rectangular ducts." There are in fact AABC firms that use the Log-Tchebycheff method.
- Mr. MacFerran states "Figures 1 & 2 show the traversal points for a 30-in. square duct using the Log-Tchebycheff and equal-area methods respectively. These figures do not show the correct spacing for either method. For the Log-Tchebycheff, both sides should have six readings; Mr. MacFerran's figure shows six on one side and five on the other. For the equal-area method, Mr. MacFerran only shows four readings for each side for a total of sixteen measurements. Actually, there should be five readings on each side for a total of twenty-five measurements.
- Mr. MacFerran states "…the equal-area method overstates air flow, which can be attributed to the measurement and averaging of only the air velocities of the interior." His figures 1 & 2 dispute that. From his figures, eight of the Log-Tchebycheff readings are in the black area totaling 26% of the total. For the equal area four of the readings are in the black area totaling 25% of the readings.
- Mr. MacFerran overstates the "exclusive endorsement" of the Log-Tchebycheff method.
- Mr. MacFerran, in his test data, indicates that the velocities at zero inches and twelve inches (the duct walls) are greater than zero. In all viscous fluid flow, the wall velocities are actually zero.

Inadequate Analysis

- Mr. MacFerran's sole conclusion from his test set up is that the equal-area traverse is wrong. He concludes that because the rectangular duct traverse does not give the same result as the round duct traverse. Actually from his test data there could be four possibilities:
  1. Both readings are wrong,
  2. Both readings are correct,
  3. Only the round duct is correct,
  4. Only the rectangular duct is correct.

From his test set up, nothing can be definitely concluded, because there is no independent verification of the fan airflow.

- In Mr. MacFerran's example of the two story elementary school, his sole conclusion again is that the equal-area traverses are wrong. There is no discussion of how the air terminals were measured. There is no discussion of instrumentation. Mr. MacFerran did make an allusion to duct leakage but only to say the duct had been pressure tested during construction, essentially ruling that out. No mention was made if the duct was subsequently inspected for leakage.
- Mr. MacFerran attributes all building airflow problems to equal-area duct traverses. It is great that eighty recent projects have no problems, but can the sole source be Log-Tchebycheff traverses? We do not know because no other variables are introduced.
- Mr. MacFerran concludes that velocity point location is the sole reason for variance in measurements. There are other factors that can contribute that need to be considered in an analysis. Some of which are how the pitot tube is held, variances in fan speed over time due to electrical distribution, and pulsing airflow created by fan cut off blades.

Mr. MacFerran proposes an interesting comparison in his article but he does not provide sound engineering data or analysis to support his conclusion.

To continue the discussion, we can examine the motion of fluid.

Air flows experienced in HVAC work are nearly always turbulent, or at least in transition with the Reynolds number well exceeding 2000. "Laminar flow may be analyzed analytically, but turbulent flows require experimental results (combined with analytical) for complete analysis." [4] From Navier-Stokes equations and Prandtl's boundary law theory, velocity profiles for fully developed, non-compressible, turbulent flow in ducts approach the form of

$$\frac{V}{V_{max}} = \left(\frac{Y}{R}\right)^{1/n}$$

where V = air stream velocity at point Y from duct edge
V_{max} = maximum velocity of air stream
R = radius of duct

Power law theory places n = 7. Actually, values of n experimentally determined by J. Nikuradse vary from 6 at Reynolds number = 4×10^3 to 10 at Reynolds number = 3.2×10^6" [5] Based on this equation, we can calculate theoretical point velocities for the equal-area method and the Log-Tchebycheff method. Since volume flow rate is equal to the average velocity times the area, we can compare the average velocity of each method, at the same area, to explore differences in flow rates.

Using the same rectangular duct sizing and hole spacing employed by Mr. MacFerran, we calculate the following velocities at the two extremes n=6 and n=10:

12 INCH SIDE

Equal Area

point	y/R	v@n=6	v@n=10
3"	.5	.89 Vmax	.93 Vmax
9"	.5	.89 Vmax	.93 Vmax
Average velocity		@n=6:	.89 Vmax
Average velocity		@n=10:	.93 Vmax

Log-Tchebycheff

point	y/R	v@n=6	v@n=10
.89"	.15	.73 Vmax	.83 Vmax
3.46"	.58	.91 Vmax	.95 Vmax
6.0"	1.0	1.0 Vmax	1.0 Vmax
8.5"	.58	.91 Vmax	.95 Vmax
11.11"	.15	.73 Vmax	.83 Vmax
Average velocity		@n=6:	.86 Vmax
Average velocity		@n=10:	.91 Vmax

Comparing Equal-Area to Log-Tchebycheff

@n=6 .89 Vmax / .86 Vmax = 1.03
@n=10 .93 Vmax / .91 Vmax = 1.02

48 INCH SIDE

Equal Area

point	y/R	v@n=6	v@n=10
3"	.125	.71 Vmax	.81 Vmax
9"	.375	.85 Vmax	.91 Vmax
15"	.625	.93 Vmax	.95 Vmax
21"	.875	.98 Vmax	.99 Vmax
27"	.875	.98 Vmax	.99 Vmax
33"	.625	.93 Vmax	.95 Vmax
39"	.375	.85 Vmax	.91 Vmax
45"	.125	.71 Vmax	.81 Vmax
Average velocity		@n=6:	.87 Vmax
Average velocity		@n=10:	.92 Vmax

Log-Tchebycheff

point	y/R	v@n=6	v@n=10
2.5"	.104	.69 Vmax	.80 Vmax
9.7"	.404	.86 Vmax	.91 Vmax
17.6"	.733	.95 Vmax	.97 Vmax
24"	1.0	1.0 Vmax	1.0 Vmax
30.4"	.733	.95 Vmax	.97 Vmax
38.2"	.404	.86 Vmax	.91 Vmax
45.5"	.104	.69 Vmax	.80 Vmax
Average velocity		@n=6:	.86 Vmax
Average velocity		@n=10:	.91 Vmax

Comparing Equal-Area to Log-Tchebycheff

@n=6 .87 Vmax / .86 Vmax = 1.01
@n=10 .92 Vmax / .91 Vmax = 1.01

Based on these conditions, the equal-area method of traversing would yield measured airflows that were 1% to 3% higher than those measured by the Log-Tchebycheff method. However, this exercise assumed a fully developed velocity profile of the air stream. In the field, fully developed flow is not always present.

The hole spacing of the Log-Tchebycheff method is based on the assumption of fully developed flow. When the flow is not fully developed, the Log-Tchebycheff method can err because too much weighting is given to boundary velocities that do not reflect the actual profile make up of the air stream. *(See figure 1)*. Taking point measurements at equal spacing across the full cross sectional area of the air stream, as in the equal-area method, allows one to see the make up of the velocity profile that actually exists. No assumptions have to be made. Analysis decisions can then be made based on this information.

So returning to the title of this article—*To Use Log-Tchebycheff or Not to Use Log-Tchebycheff…Is That the Question?*—the answer is no that is not the question. The question is—What is the proper approach for accurate field testing and analysis of HVAC systems.

At the Associated Air Balance Council our answer is **Total System Balancing**. In our methodical approach, a duct velocity traverse is a tool for analysis. While it can be a major tool, it is only one tool and it is not an end in itself. We view the system as a whole. In addition to duct traverse measurements, fan data will be obtained (static pressures, rpm, bhp), mixed air temperatures will be measured,

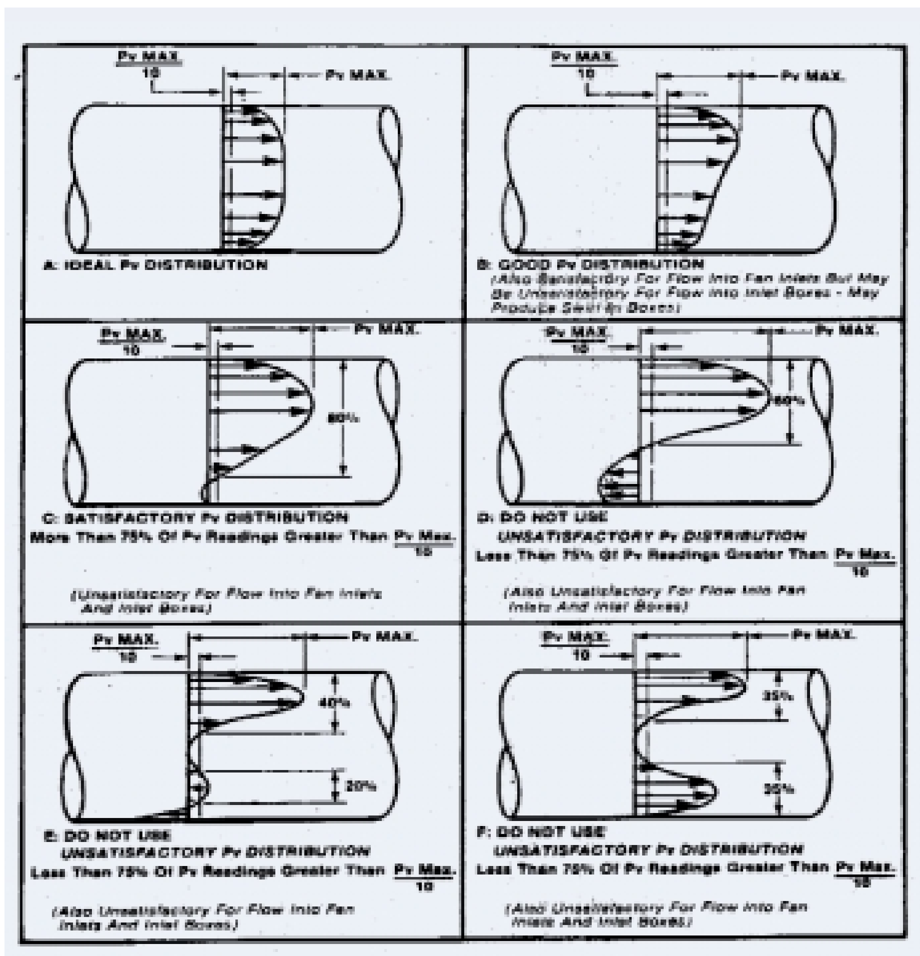

Figure 3: *from ANSI/ASHRAE Standard 111-1988*

air terminal flows will be gathered, space and building pressures will be monitored. All of this data will be reviewed as a whole to accurately determine and evaluate system performance. From this point, systems can be balanced to deliver design intent and operate at an optimum level.

In conclusion, accurate field data is at times difficult to obtain. In those instances, care must be taken to not exclude possible contributing factors that affect the results.

References

[1] AMCA Publication 203 *A Guide to the Measurement of Fan-System Performance in the Field.*

[2] AMCA Publication 203 *A Guide to the Measurement of Fan-System Performance in the Field.*

[3] ANSI/ASHRAE 111-1999 *Practices for Measurement, Testing, Adjusting and Balancing of Building Heating, Ventilation, Air-Conditioning, and Refrigeration Systems.*

[4] Olsen, Ruben M. *Essentials of Engineering Fluid Mechanics.*

[5] Olsen, Ruben M. *Essentials of Engineering Fluid Mechanics.*

reader feedback

Letter to the Editor

TAB Journal welcomes submissions for publication. TAB Journal is published quarterly by the Associated Air Balance Council. Send letters or articles to: Editor, TAB Journal 1518 K Street, NW, Suite 503 Washington, DC 20005

In the Spring 2001 issue several mistakes were made in my article "Understanding the Design Intent of Variable Volume Lab Controls and Pressurization Systems" when printed:

- In the example on page 3, the Supply box maximum is 1000 CFM not 700.
- On page 4, Fig. 1, the supply CFM in the room is 650, not 530 CFM.
- Also, on page 4, Fig. 2, the supply CFM in the room is 650 not 860 CFM.

Please advise our readers as the present values are confusing.

Sincerely,

William A. Derse, T.B.E.
Professional System Analysis, Inc.

EQUAL AREA vs. LOG-TCHEBYCHEFF Revisited

In a study initiated by *HPAC Engineering*, methods of measuring air flow in rectangular ducts are put to the test.

EDITOR'S NOTE: In December 1999, HPAC Engineering *published "Equal Area vs. Log-Tchebycheff," an article by Ernest L. MacFerran, PE, championing the little-known Log-Tchebycheff method of measuring air flow in rectangular ducts, which, the author claimed, produces more-accurate results than does the widely used Equal Area method. The article generated much response from readers. Some vowed always to specify the "Log-T" method for test-and-balance reports, while others dismissed the differences in accuracy as insignificant. In an effort to further the discussion,* HPAC Engineering *asked the Iowa Energy Center to test the two methods. The results are presented here.*

Marty Pieper of Systems Management and Balancing Inc. measures duct velocities at Traverse Plane No. 1. Note the difference in measurement-point locations between the Equal Area (top) and Log-Tchebycheff methods.

By **CURTIS J. KLAASSEN, PE,**
and **JOHN M. HOUSE, PhD,**
Iowa Energy Center

Pitot-tube traverses commonly are used during test-and-balance procedures to determine volumetric air-flow rates in ductwork. For rectangular ducts, there are two accepted methods of determining the grid of locations where measurements should be taken, namely, the Log-Tchebycheff method adopted by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE)[1,2] and the Equal Area method supported by the Associated Air Balance Council (AABC).[3] Both methods determine duct air velocity by sampling velocity pressure at individual points in the traverse plane. Where they differ is in the rules that prescribe the location of those points. The Log-Tchebycheff method purports greater accuracy because the loca-

Curtis J. Klaassen, PE, is the manager of and John M. House, PhD, is the research engineer for the Iowa Energy Center's Energy Resource Station (ERS), a research, testing, demonstration, and training facility for building energy systems. The ERS is located on the campus of Des Moines Area Community College in Ankeny, Iowa. Klaassen has over 20 years of experience in the design of HVAC systems and the application of energy-efficient technology. House formerly was with the National Institute of Standards and Technology, for which he served as a project leader in the area of building controls. Klaassen can be contacted via e-mail at curtk@energy.iastate.edu, *while House can be contacted at* jhouse@energy.iastate.edu. *For more information on the ERS, visit* www.energy.iastate.edu.

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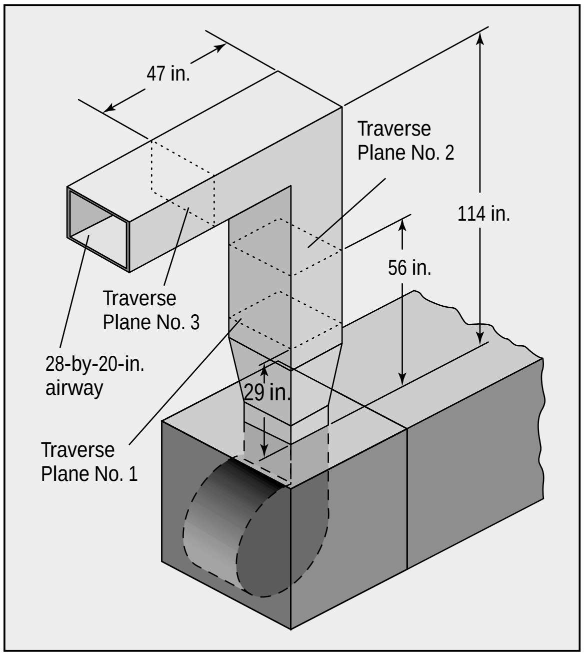


FIGURE 1. Schematic of ductwork and traverse-plane locations.

tion of its points accounts for friction loss at the duct walls.[1]

This article compares air-flow rates obtained with the Log-Tchebycheff and Equal Area methods and examines the influence traverse-plane location had on the measurements. Testing was conducted at the Iowa Energy Center's Energy Resource Station (ERS), which supports two commercial-scale air-handling systems serving matched pairs of test rooms and one general-service system serving the remainder of the building.

The testing was part of an effort to identify duct-velocity profiles and calibrate air-flow-measuring stations for the general-service air-handling system. The tests were intended to provide a comparison of the traverse methods under the less-than-ideal flow conditions frequently encountered in the field. The testing was limited to one main-supply-duct size and a specific set of operating conditions.

TEST CHARACTERISTICS

Ductwork. Figure 1 is a schematic of the air-handling-system supply-air ductwork. The air-handling-unit upblast discharge provides air directly to this main supply-air-ductwork section. The dimensions of the ductwork go from the 21 in. by 18 in. of the air-handling-unit outlet to the 30 in. by 22 in. of the sheet-metal duct, where the measurements were taken. A 1-in. liner reduces the duct's interior dimensions to 28 in. by 20 in. Although not shown in the diagram, turning vanes are installed in the 90-degree elbow.

The locations of the three traverse planes (a traverse plane is located at the tip of a Pitot-tube probe) are shown in Figure 1. For the velocities anticipated, 100-percent effective duct length corre-

The general-service air-handling system used for the tests. It serves the classrooms, offices, and common areas of the Energy Resource Station with a nominal capacity of 7,800 cfm.

sponding to a uniform velocity profile would be expected at two-and-one-half equivalent duct diameters downstream from the fan outlet.

The three traverse planes can be summarized as follows:

• System effect and the effect of a fan discharge are represented at Traverse Plane No. 1, which is approximately 50-percent effective duct length from the outlet of the fan.

• Traverse Plane No. 2 is located approximately 100-percent effective duct length from the outlet of the fan.

• The duct elbow with turning vanes introduces an upstream disturbance for Traverse Plane No. 3 at a distance slightly greater than one equivalent duct diameter. Approximately 32 in. downstream from Traverse Plane No. 3 is the first

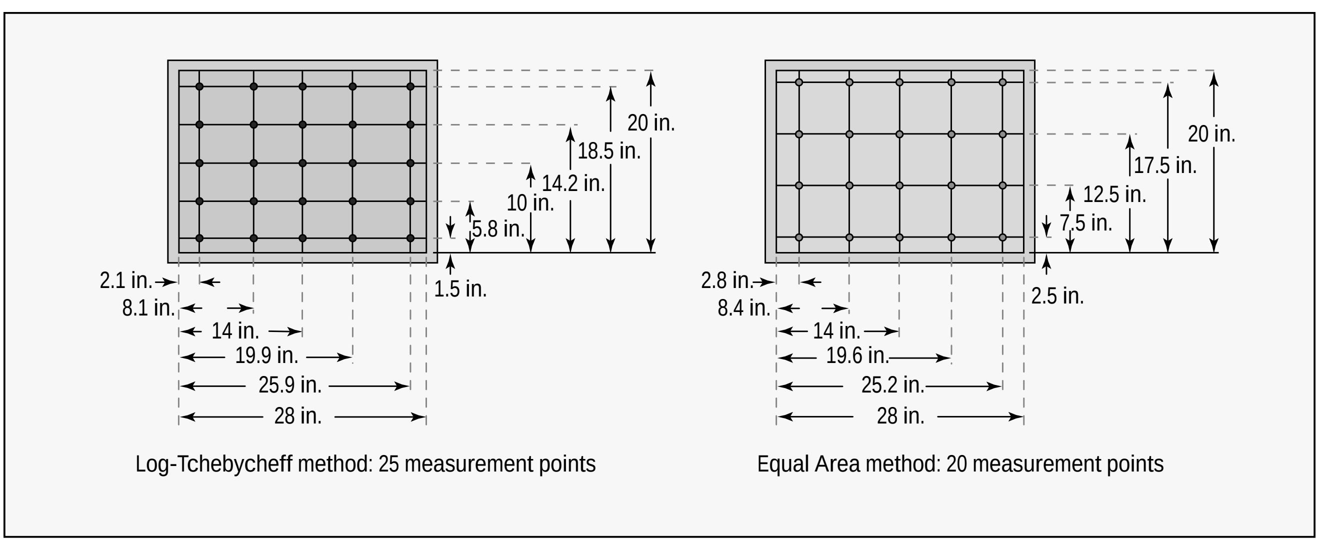


FIGURE 2. Log-Tchebycheff and Equal Area traverse grids for a 28-by-20-in. airway.

branch duct takeoff from the main supply duct.

Measurement grids. For a duct with a 28-by-20-in. airway, the Log-Tchebycheff method calls for a five-by-five grid of unequally spaced measurements,[1] while the Equal Area method requires a five-by-four grid[3] with the distance between measurements no more than 6 in. The locations of the measurement points for both methods are shown in Figure 2.

Air-handling-system operation. Prior to and throughout the test period, the general-service air-handling system was operated in a steady-state, constant-volume mode. The supply and return fans were overridden to fixed-speed operation, and the outside-, return-, and exhaust-air dampers were positioned for 100-percent return air. The fan-powered, variable-air-volume box dampers were fixed at the full open position, with the fans disabled. To determine the stability of system operation, an electronic flow-measuring-station signal was recorded each minute. The system maintained a stable air-flow rate, with a peak-to-peak range consistently less than 2.3 percent of the mean flow.

A floor-up view of the supply-air ductwork, showing the direction of air flow from the fan discharge. Traverse Plane Nos. 1 and 2 are in the vertical section of the ductwork, while Traverse Plane No. 3 is in the horizontal section.

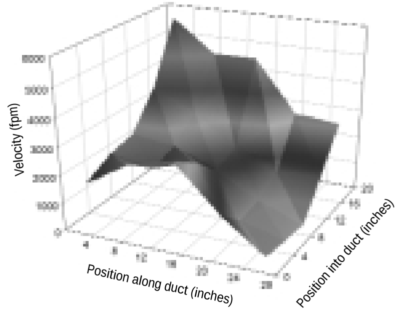
Performing measurements. The measurements were performed using a Shortridge Airdata Multimeter Model ADM-860 with a Certificate of Recalibration dated seven weeks prior to the tests. This instrument provides automatic pressure compensation to account for non-standard conditions. Attaching a temperature probe to the instrument provides temperature compensation.

The instrument was operated in a differential-pressure mode, with velocity computed internally in units of feet per minute (fpm). Using the calibration data sheet, the uncertainty of the velocity measurements was estimated to be ±3 percent of the reading.

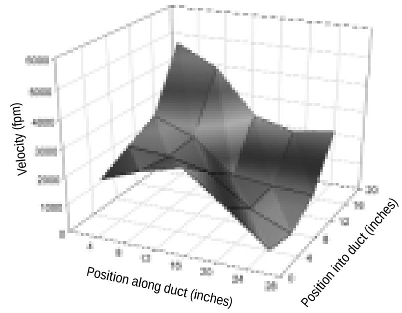
To minimize measurement error resulting from instrument operation, the services of a testing-and-balancing engineer were enlisted. Well-qualified with 17 years of field experience, Marty Pieper of Systems Management and Balancing Inc. performed all of the measurements reported in this article.

Data sets. Measurements were made at each of the traverse planes shown in Figure 1 using both the Log-Tchebycheff and the Equal Area measurement locations. At each location, three measurements of air velocity were obtained consecutively and then averaged to establish a mean velocity for that location. The entire procedure was repeated to produce 12 data sets based on accepted standards defined by ASHRAE and AABC.

It was determined that the most uniform velocity profile was located in the horizontal section of duct at Traverse Plane No. 3. Ideally, the reference air-flow rate would have been established by measuring differential pressure across a primary instrument, such as a flow nozzle. For this experiment, such a measurement was not practical. Instead, the reference air-flow rate was determined using a Pitot-tube traverse of a much higher resolution. In particular, measurements were taken with a 14-by-10 grid, with the Pitot tube positioned at the center of 2-by-2-in. squares. For the reference case, only a single measurement was taken at each location.



Log-Tchebycheff method



Equal Area method

FIGURE 3. Velocity profiles obtained at Traverse Plane No. 1, Run No. 2.

RESULTS

Results of all of the tests are presented in Table 1, with velocity profiles for the shaded cases plotted in figures 3-5. Both ASHRAE and AABC provide guidelines regarding the acceptability of velocity profiles. These guidelines say that for a velocity distribution to be acceptable, 75 percent or more of the velocity measurements must be greater than 1/10 of the maximum velocity of that profile. The ASHRAE guideline further states that for a distribution to be considered ideal, 80 to 90 percent of the velocity measurements must be greater than 1/10 of the maximum velocity of that profile. At Traverse Plane No. 1, 80 to 90 percent of the velocity measurements were greater than 1/10 of the maximum velocity, while at both of the other traverse planes, 100 percent of the velocity measurements were greater. By the above criteria, then, all of the profiles recorded at all three traverse locations satisfy the requirement for an ideal distribution.

The velocity profiles obtained with

		LOG-TCHEBYCHEFF		EQUAL AREA	
Traverse Plane No.	Run No.	Air-flow rate (cfm)	Relative error (%)[a]	Air-flow rate (cfm)	Relative error (%)[a]
1	1	7,811	-0.04	7,288	-6.73
	2	8,204	4.99	7,623	-2.44
2	1	7,620	-2.48	7,352	-5.91
	2	7,639	-2.24	7,187	-8.02
3	1	7,700	-1.46	7,838	0.31
	2	7,740	-0.95	7,843	0.37
3	Reference[b]			7,814	

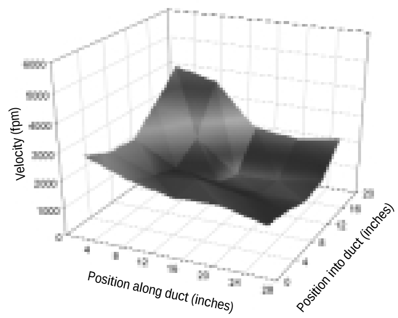
[a] The relative error is determined from:

$$\frac{Q\text{–Reference}}{\text{Reference}} \times 100\%$$

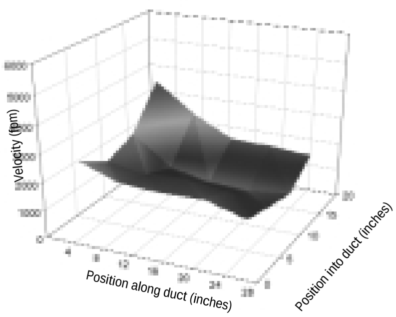
where:
Q = Equal-Area- or Log-Tchebycheff-method air-flow rate.

[b] The reference air-flow rate was obtained using a 14-by-10 grid. All other Equal-Area-method results were obtained with a five-by-four grid, while all Log-Tchebycheff-method results were obtained with a five-by-five grid. Recommended grids for both methods are dependent on duct size.

TABLE 1. Results of the air-flow measurements.



Log-Tchebycheff method



Equal Area method

FIGURE 4. Velocity profiles obtained at Traverse Plane No. 2, Run No. 2.

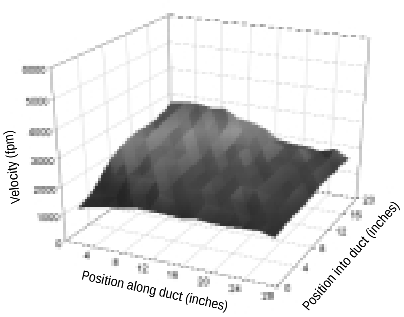


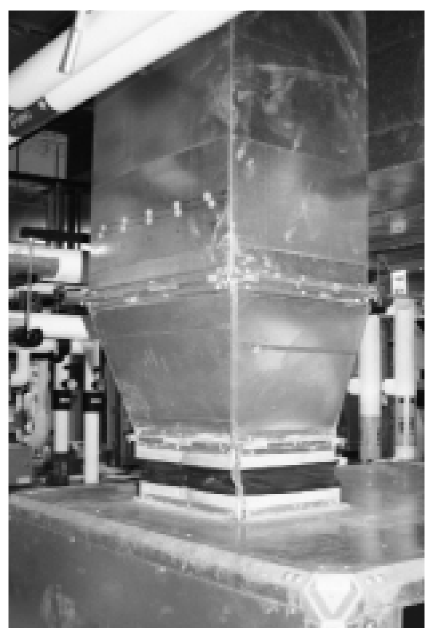
FIGURE 5. High-resolution Equal-Area-method velocity profile obtained at Traverse Plane No. 3.

the Log-Tchebycheff and Equal Area methods at Traverse Plane No. 1 are presented in Figure 3. Although the profiles are very non-uniform, they are consistent between the two methods. Velocities on the far left side at the back of the duct (position along the duct close to 0 in. and position into the duct approaching 20 in.) approach or exceed 5,000 fpm, while velocities in the front right corner (position along the duct close to 28 in. and position into the duct approaching 0 in.) are very low. In fact, velocities at some locations in the front right corner are negative with both methods and were recorded as zero in accordance with the ASHRAE standard.[1]

The non-uniformity of the profiles at Traverse Plane No. 1 was expected given the abrupt transition disturbance just upstream. The highest velocities occurred at a location directly in line with the fan discharge, while the lowest velocities occurred at a location directly in line with the most severe transition. The air-flow rates at Traverse Plane No. 1 showed a wide variation both between the two methods and between the two runs performed with each method.

Figure 4 shows the velocity profiles obtained with the Log-Tchebycheff and Equal Area methods at Traverse Plane No. 2. Although, as with Traverse Plane No. 1, the profiles are very similar, the range of velocities is substantially smaller. The profiles are interesting in that they have the appearance of an inverted “D.” Instead of the highest velocities being at the center of the duct, as is the case with fully developed turbulent flow in straight ducts, the highest velocities are near the walls.

Table 1 shows that while the air-flow rates obtained with both methods at Traverse Plane No. 2 are less than the reference value of 7,814 cfm, the rates obtained with the Log-Tchebycheff method are more consistent between the two runs (7,620 cfm and 7,639 cfm) and are within 2.5 percent of the reference value.



The vertical portion of the main supply-air ductwork of the general-service air-handling system. The yellow duct plugs identify Traverse Plane No. 1.

The high-resolution Equal Area profile obtained at Traverse Plane No. 3 is shown in Figure 5. This profile, obtained with a grid of 140 measurement points, shows that the velocities, although still not displaying the classic "D" shape, are much more uniform. Because the profiles obtained with the Log-Tchebycheff method (five-by-five grid) and the Equal Area method (five-by-four grid) also were highly uniform, they are not presented.

Table 1 shows that the two air-flow rates obtained with the Log-Tchebycheff method at Traverse Plane No. 3 differ from one another by only 40 cfm and differ from the reference value by less than 1.5 percent, while the two air-flow rates obtained with the Equal Area method are nearly the same and differ from the reference value by less than 0.4 percent. Even though the Log-Tchebycheff measurements slightly underpredict the reference value, and the Equal Area measurements slightly overpredict it, both are very satisfactory. In fact, the differences in the results obtained with the two methods and those obtained with the high-resolution Equal Area grid are well within the estimated uncertainty of the velocity measurements. The implication is that, with the results from Traverse Plane No. 3, no conclusion can be made regarding which method is more accurate.

CONCLUSIONS

The primary conclusion that can be drawn from these tests is that the uniformity of the velocity profile offered by the traverse-plane location has a more significant influence on an air-flow measurement than does the method (Log-Tchebycheff or Equal Area) used to determine the measurement grid.

At Traverse Plane No. 3, where the velocity profiles are very uniform, the Log-Tchebycheff and Equal Area methods produce results that are in excellent agreement with the reference air-flow rate determined using a high-resolution grid traverse. At Traverse Plane No. 2, the velocity profiles are less uniform, with the average measurement of the Log-Tchebycheff method approximately 2.4-percent less than the reference value and the average measurement of the Equal Area method approximately 7-percent less than the reference value. At this location, the additional measurement points of the Log-Tchebycheff method provide the resolution necessary to capture the velocity profile. At Traverse Plane No. 1, the velocity profiles are the least uniform, and the results are the least consistent. This is the only location at which negative readings were obtained, a factor that may have contributed to the inconsistency of the measurements.

The variances identified at traverse planes 1 and 2 occur under velocity-distribution conditions considered ideal by the criterion that 80 to 90 percent of the velocity measurements be greater than 1/10 of the maximum velocity. This reinforces the importance of this criterion in determining acceptable velocity profiles for the traverse-plane location selected. Improved confidence in the measured values is expected as the 1/10 threshold increases.

The testing reported here considers only a single duct size and air-flow rate; therefore, it is not possible to draw any conclusions about the generality of the results. The results do, however, suggest that additional research aimed at comparing the accuracy of the Log-Tchebycheff and Equal Area methods is merited. In particular, the scope of the comparisons should be extended to consider a range of air-flow rates, duct sizes, and configurations, with measurements taken under field conditions.

ACKNOWLEDGMENT

The authors wish to acknowledge the Dept. of Mechanical Engineering at The University of Iowa for assistance with the data analysis.

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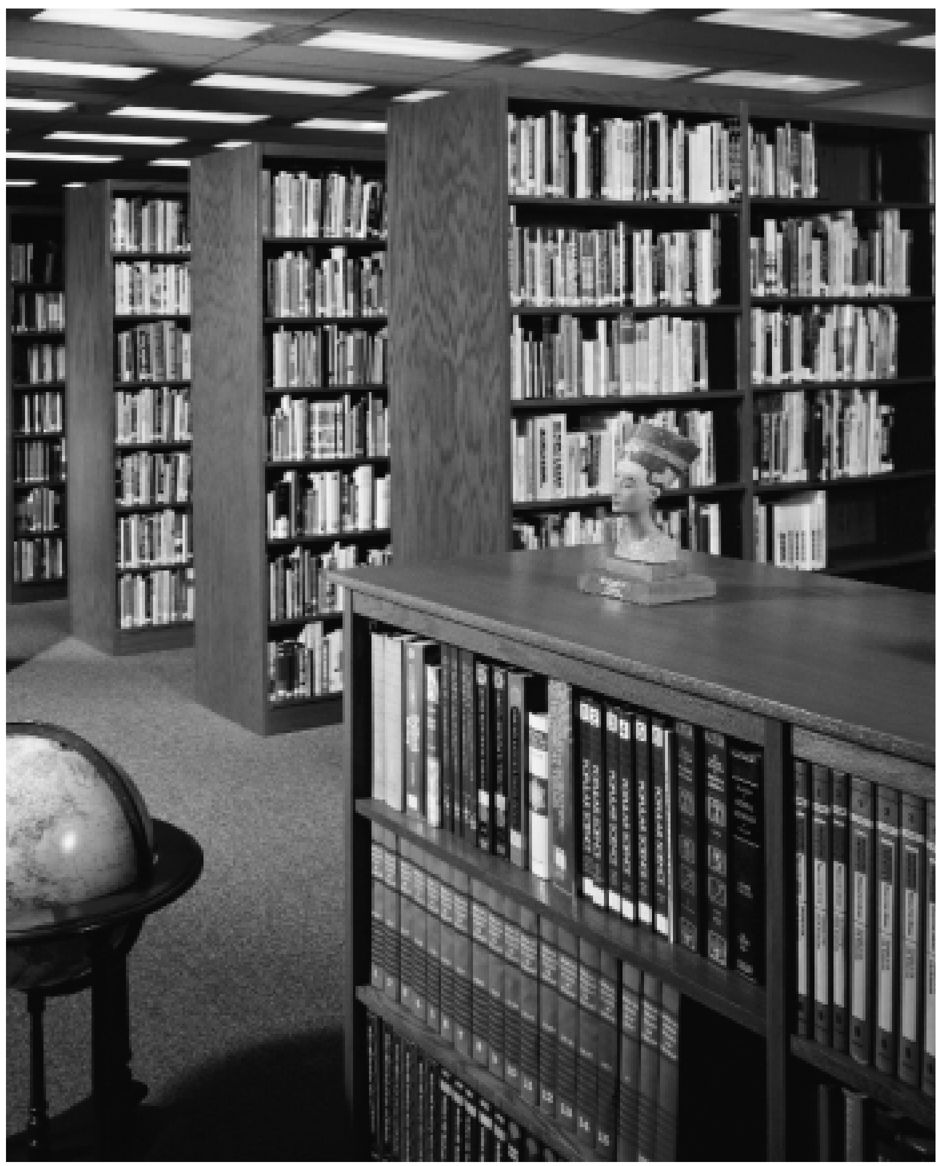
The Balancer

This humorous story was written by an anonymous author working at the University of Iowa and submitted by Odean Jukam of Systems Management & Balancing, Inc.

July's heat had solidly shoved me into midsummer doldrums. The dental students had taken leave of the college for the month and my office was strangely quiet without them. My office mate was on vacation, and I busied myself getting ready for the fall term. What better time than now to install the new air-conditioning system made necessary by the total disintegration of the old one.



Outside my office door, the workman removed a section of the ceiling and pulled an air duct down through it. Cold air, approximately 62 degrees cold, pulsated through my office door. The first two days were great, invigorating. After that I began to freeze my garbanzos off.

I called the building supervisor. "When is this cold air going to be regulated?" I asked in my most determined peremptoral manner.

"Well," he answered, "Hold on for a few more days. The Balancer will come around when all the duct work is installed."

I was perplexed, bewildered, mystified. A Balancer? Did I recognize the term? Hmmm. Bouncer? No, he works in taverns. Leveler? Doesn't he cement brick? No, I didn't recognize the word.

"The Balancer?"

"Yeah!"

"What is a Balancer? A defector from a Russian bear juggling act?"

"Heck, no," he chuckled. "A Balancer is the guy who regulates the heat, cold and thermostats after the new system has been installed."

Silly me! I thought this guy had real talent. "Well, when is this Balancer supposed to show his face in my office?"

"I already answered that…in a day or two."

I began to wonder what are the physical requirements to become a Balancer. Is he tall? Short? Bald? Skinny? Maybe the building super lied to me. Maybe he was really a she. Maybe the Balancer was a balance beam reject from the UI's women's gymnastic team, and maybe this gymnast moonlights as a Zambonie operator in the winter season. I finally decided the Balancer has to be small and agile with an abnormal body thermostat. How else could he/she work in the small and cold ceiling holes left open for this magic trick. I then began to question every small workman who might look like a Balancer.

"Are you the Balancer?"

"Nope! I think he is down on the second floor."

"Why the hell is he down there?"

"Lady! He is working his way up through the building."

A week passed and I had not seen or met the Balancer despite my steady inquiry. I now began wearing socks and long sleeved shirts. I began to question co-workers from other clinics.

"Have you seen or do you know the Balancer?"

"No, but I wish he would make it to our office. It's 92 degrees in there today."

I raced there to warm up.

My desire to meet the Balancer remains… unbalanced.

By now, my curiosity was overwhelming. Who was this guy and where was he? I readied my office for his arrival. First, I would let him balance the cold air and then set the thermostat. Then I would casually show him another cold air duct…in a remote part of the clinic. There I would stuff his tiny body into the air duct and seal him up in his own special time capsule with his own duct tape. Years from now, his body would be found frozen like an extinct woolly mammoth ready to spring to life and start balancing once more.

Slowly paranoia seized my mind. Several of the workmen would laugh whenever I went by. I was sure they knew my dire intent for the Balancer and were hiding him from me, never mind it was mid July and I looked pretty silly in my bulky knit sweater and winter weight slacks. I carried hot coffee wherever I went. I sucked it up by the quarts. If I had been a dog at the vet's, the vet would have felt my cold nose and pronounced me healthy.

I again called the building supervisor to complain. Again I was assured that the Balancer was 1) in the building and 2) would soon be in my area. Lies! All lies!! I was sure our squeeny-eyed governor had cut the Balancer's position from the state payroll. All of us at the Dental College were doomed. We would all freeze to death…slowly at our work stations…one by…one.

The next day, I arrived at work to find some of the ceiling holes closed. A number of workmen were replacing ceiling tiles at the clinic. The temperature in my office felt…for want of a better word…balanced. The Balancer was here! He had to be here!

"Okay! Folks! Where is the Balancer? I want to talk to him!"

"Oh, he's been here and gone."

"What!?! Where did he go? I want to see that fella."

"Sorry, Lady, he's gone."

Just like that. Gone. What a dirty rotten little sneak. His delayed visit forced me, prematurely, into my winter long johns, and it is still only July. Now he was gone. That illusive little fart had escaped me and his just fate.

July became August. The dental students returned. I started wearing normal summer clothes, and my abnormal fear of the dew point returned. But my desire to meet the Balancer remains, for want of a better word…unbalanced. On a clear day, I can look out the plate glass windows near my office on the fourth floor and see a large part of the UI campus. I know the Balancer is out there, somewhere, in some obscure corner of a large stone building, peeping into cold air ducts performing his magic act. I hope the little weasel gets stuck in a duct and blue-green fungus covers his tiny body. Maybe the hospital could use him to produce penicillin. Yeah! Justice at last!!


COVER
THE WATERFRONT]
Alnor's new HM 650 Hydronic Manometer sets a new standard for precision and versatility
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• easy valve control speeds up purging and zeroing
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Tech tips

Occasionally, AABC receives short "case study" technical papers from our members. These papers usually focus on observations made by AABC members working on a project "in the field," in which they explain a certain problem they have encountered, and what corrective actions they instigated to overcome that problem. Each of these papers presents certain problems or challenges to the test and balance professional, and provides insight into how these situations can be resolved.

These papers are relatively short but may hold special appeal for others involved with the every-day experience of testing and balancing. We therefore decided to publish these papers as a collection of articles in Tech Tips, a technical newsletter inside TAB Journal that can be removed for your convenience.

Do you have a "Tech Tip" that you would like to share with our readers? If so, please contact AABC at:

Associated Air Balance Council

1518 K Street NW, Ste 503
Washington, DC 20005
Phone: 202.737.0202
Fax: 202.638.4833
E-mail: aabchq@aol.com

Tech Tips are written for and by our readers. We thank them for sharing their valuable experiences and providing solutions to problems in our industry.

Designing Outside Air Systems

David C. Parker and Bernard S. Moltz
Bernie Moltz, Inc.

It appears there are still some engineers designing variable volume systems who are not totally familiar with the design process. A case in point is the design of the outside air portion of the system.

The minimum ventilation air requirement must always equal the minimum volume settings of the variable volume terminals. As an example, if 4,000 CFM is the minimum ventilation requirement, the minimum air at the terminals must equal that amount.

We have observed on many occasions that the total ventilation air at the terminals is far less than the design total outside air. Obviously one of the quantities is incorrect, raising the question as to how the test report can confirm the design.

In the case of a system with preconditioned forced outside air, the volume that is in excess of the minimums will be forced through the return system resulting in occupant discomfort, humidity problems and building pressurization problems.

It appears that the engineer's desire to provide safety factors is misapplied when considering the outside air systems in addition to the other parts of the system. If a safety factor is intended for the outside air, it must be applied to both the total intake as well as the total at the terminals. It must be recognized that the terminal supplier will match the terminal specification not the total volume.

Traversing for Flow Correction Factors

Gaylon Richardson
Engineered Air Balance Co., Inc.

It appears that in the field of testing and balancing, everyone has forgotten that it is an art, not a science. We now have digital manometers, digital hoods, digital anemometers, and digital controls to measure airflow. In the field, which is the most accurate?

The accuracy question has to start with the Pitot tube traverse. So naturally, the question that arises is this: Is the digital manometer accurate? The positive and negative side of the digital manometer must be verified against an analog manometer in the range of the traverse. What is the accuracy of the digital instrument? Does the digital manometer take a time-weighted average? When compared to analog traverses, is the answer the same? Other factors that also play a role in accuracy before comparing or taking a traverse are:

- Is the outlet connection air-tight so that leakage will not affect the accuracy of the velocity reading and the traverse readings?
- Is the duct straight and of sufficient length to obtain an accurate traverse?
- Is there a need for a density correction?
- Is the Pitot tube free of leaks?
- Are the hoses air-tight?
- Are the controls set to a fixed quantity?

The question of accuracy has to be answered in the following manner for digital manometers:

- They have an accuracy of 2% to 5% at different velocities.
- Take a single point in a traverse plane with a digital and an analog manometer and record the velocity every five (5) seconds. Does the velocity change over 5%, 10%, or 15%?
- Is the velocity high enough where 0.01 i.w.g. in velocity pressure change is not significant?

Exploring the issue of velocity pressure change suggests the plane of the traverse must be taken at velocities high enough to not cause error. AABC, AMCA, and ASHRAE have all noted that traverses must be above a velocity pressure of 0.023 i.w.g. which corresponds to a velocity of approximately 600 FPM for air of 0.075 lb/ft^3 density. Taking this idea a step further, what is the difference at low velocity pressures? *See Figure 1.*

Difference at Low Velocity Pressures

VP	Velocity
0.03	694
0.035	749
0.04	801
0.045	850
0.05	896
0.055	939
0.06	981
0.065	1021
0.07	1060
0.075	1097
0.08	1133
0.085	1168
0.09	1202
0.095	1234
0.10	1266

Figure 1: *This writer would round the velocities to the nearest 5 to suggest there is no way to obtain accuracies to the nearest 1.*

By taking the difference between 0.03 to 0.035 and dividing by 694, the percentage of difference is 7.9%. By taking the difference between 0.095 and 0.10 and dividing by 1234, the percentage of difference is 2.6%.

Assuming the flow factor is established accurately, then the instrument used in the field to measure velocity is actually being field calibrated. (Because of this, the manometer and hood will be the main focus). The issue with using the flow factor is the velocity measuring instrument must be positioned exactly in the same place and direction for each reading.

Case Study

A series fan powered box system using 4' slot diffusers was balanced by the following procedure:

- Traverse with a digital manometer on the discharge of a few boxes to determine the hood's flow factor.
- Readout the slot diffusers with the digital hood's flaps open.
- Use a flow grid 14" x 14" to establish equalized flows between the primary air and discharge so the induction opening will be 0 velocity.
- Adjust the minimum flow to design by deducting the flow grid reading from the discharge.

The traverses and the velocity readings indicated that the flow hood, with the flaps open, had no correction factor. The flow hood manufacturer stated for linear slot diffusers with airflow less than 100 CFM per lineal foot, the hood readings should be taken with the flaps closed. The location of the traverse was within 4 to 5 feet off the fan powered box discharge and the hood reading agreed within 10%, but was usually 10% to 20% higher than the air volume of the fan powered box. The new technology of the ECM motor was used which calculates air volume by the torque and RPM based on a demand signal.

The manufacturer of the box used AMCA nozzles to test the airflow of the hood reading with the slot diffuser supplied for the project. The results using a 1' x 4' hood top and the meter which was used in the field are found in *Figure 2.*

Under controlled conditions in the lab, the hood (with flaps open on a two-way horizontal diffuser pattern), varied from 12.5% to 1.6% difference from the setpoint. With the hood flaps closed, the difference from setpoint varied from 2.0% to -6.8%. Readings taken with flaps closed and one hood centered varied between 0% to -5.5%. Readings taken with the hood flaps open and the hood in the front position varied between 1.6% and 7.5%. With vertical down flow, the readings with flaps closed varied between 2.0% and -3.1% with the hood at the front position, and 2.5% to -3.0% with the hood at the center position.

In an independent lab, two different sized boxes were tested. A discharge traverse was performed with approximately 8 diameters upstream and 2 diameters downstream. The first traverse had an average velocity of 817 FPM in a 12" round duct. The traversed airflow was 642 CFM compared to an orifice reading of 562 CFM which is 14% higher. The other traverse was in a 16" round duct. The average velocity was 691 FPM, resulting in 964 CFM. The traversed airflow of 964 compared to an orifice reading of 898 CFM was 7% higher.

4' Linear Slot Diffuser With 2-Way Horizontal Flow					4' Linear Slot Diffuser With 2-Way Vertical Flow				
CFM Setpoint	Hood Location	Flap Position	Flowhood CFM Reading	% Difference From Setpoint	CFM Setpoint	Hood Location	Flap Position	Flowhood CFM Reading	% Difference From Setpoint
200	Back	Open	225	12.5	200	Back	Open	223	11.5
200	Back	Closed	204	2.0	200	Back	Closed	206	3.0
200	Front	Open	215	7.5	200	Front	Open	217	8.5
200	Front	Closed	202	1.0	200	Front	Closed	204	2.0
200	Center	Open	221	10.5	200	Center	Open	219	9.5
200	Center	Closed	200	0.0	200	Center	Closed	205	2.5
250	Back	Open	278	11.2	250	Back	Open	274	9.6
250	Back	Closed	249	-0.4	250	Back	Closed	255	2.0
250	Front	Open	262	4.8	250	Front	Open	271	8.4
250	Front	Closed	247	-1.2	250	Front	Closed	254	1.6
250	Center	Open	272	8.8	250	Center	Open	275	10.0
250	Center	Closed	248	-0.8	250	Center	Closed	255	2.0
300	Back	Open	318	6.0	300	Back	Open	318	6.0
300	Back	Closed	289	-3.7	300	Back	Closed	296	-1.3
300	Front	Open	308	2.7	300	Front	Open	315	5.0
300	Front	Closed	286	-4.7	300	Front	Closed	299	-0.3
300	Center	Open	316	5.3	300	Center	Open	317	5.7
300	Center	Closed	287	-4.3	300	Center	Closed	294	-2.0
350	Back	Open	375	7.1	350	Back	Open	365	4.3
350	Back	Closed	335	-4.3	350	Back	Closed	339	-3.1
350	Front	Open	358	2.3	350	Front	Open	376	7.4
350	Front	Closed	331	-5.4	350	Front	Closed	341	-2.6
350	Center	Open	365	4.3	350	Center	Open	365	4.3
350	Center	Closed	332	-5.1	350	Center	Closed	344	-1.7
400	Back	Open	426	6.5	400	Back	Open	432	8.0
400	Back	Closed	382	-4.5	400	Back	Closed	396	-1.0
400	Front	Open	407	1.8	400	Front	Open	420	5.0
400	Front	Closed	376	-6.0	400	Front	Closed	394	-1.5
400	Center	Open	420	5.0	400	Center	Open	422	5.5
400	Center	Closed	378	-5.5	400	Center	Closed	388	-3.0
425	Back	Open	449	5.6	425	Back	Open	445	4.7
425	Back	Closed	398	-6.4	425	Back	Closed	416	-2.1
425	Front	Open	432	1.6	425	Front	Open	445	4.7
425	Front	Closed	396	-6.8	425	Front	Closed	412	-3.1
425	Center	Open	444	4.5	425	Center	Open	446	4.9
425	Center	Closed	403	-5.2	425	Center	Closed	419	-1.4

Figure 2

Conclusion

Field traverses for flow factors should be taken above 1250 FPM to keep the error in the range of 2.5%. The outlet should be measured the same each time. With flow hoods, the measurement should be taken with deflection downward. If the velocities are too low to traverse, flow factors should be established using orifice tubes. In the case study, the number game was played but the best results would have been to use the airflow established by the fan powered box ECM calculation.

The AABC 2001 National Standards indicate the system to be balanced by the following methods:

Procedure: The entire air handling system must be fully operational: all inspections performed as described in Chapter Six, final filters installed, and all controls fully operational with all outlet dampers fully open. Set the system for balancing in the following manner:

- Put each air valve to full cooling and observe that the correct thermostat controls the correct air valve.
- Record the air handling unit model and serial number, the motor nameplate data, the sheave and belt data, the filter sizes and conditions, the starter data, and thermal overload protection sizes and ratings.
- Record the actual RPM and verify the correct rotation of the fan.
- Record actual operating amps and volts and compare to motor nameplate. If amperage is above nameplate, slow fan RPM until the amperage is at nameplate.
- Verify minimum outside air is set close to design.
- Verify the most remote air valve has the minimum static pressure required.
- Starting with the fan powered box closest to the air handling unit, adjust the thermostat to full heating. Proportion the outlets with the primary air valve at zero flow and adjust the fan control device to deliver the design cfm. Set the primary air to maximum flow and adjust its controller so the primary airflow matches the fan's airflow. This is accomplished by partially covering the return opening and observing with a 4" (100 mm) vane anemometer 0 flow of the return air plenum. Remove the covering and read the flow sensor pressure differential at maximum airflow for the primary air valve. Determine the minimum airflow sensor pressure differential with Equation 8.1.

Equation for Determining the Minimum Airflow

$$P_d MIN = P_d MAX \times \left(\frac{CFM\ MIN}{CFM\ MAX}\right)^2$$

Where:
CFM MIN = Minimum Design Airflow
CFM MAX = Maximum Airflow Measured
P_d MIN = Unknown Minimum Differential Pressure
P_d MAX = Maximum Pressure Differential Measured

Set the minimum airflow on the air valve with the calculated pressure differential. Follow the same procedure for the next fan powered box until all the fan powered boxes and outlets are proportioned.

- In order to prevent over-pressurization of the system, all air valves must be in control with at least one air valve controlling approximately 80% - 90% open when the system is at maximum cooling airflow.
- When taking final Pitot tube traverses and there is diversity, the air valves closest to the air handling unit will be set to their design minimum to simulate diversity. All the remaining air valves will be set at design maximum airflow. The air valves used for diversity will be recorded on the air traverse data sheet.
- With the system in maximum airflow, or diversity as applicable, record VFD settings. If below 60hz, adjust the sheave package so that the VFD will operate at 60hz, provided there is not future expansion to the system or concern for filter loading.

Record static pressure at the static pressure controller. Verify the controller static pressure reads as the measured static pressure. This will be used as the controller setpoint and will be set by the person responsible for the control system. Record the inlet static pressure at the air valve on the end of the system. Put all air valves to minimum flow and record the static pressure at the sensor to verify that the controller is maintaining the system static pressure as the fan volume modulates. Record the minimum outside air CFM (l/s) at minimum flow.

- Record the final measured data with the air valves set for maximum cooling and at design minimum outside air. Re-take the information with the economizer cycle set for 100% outside air. Test the economizer cycle as described in Chapter Six.

REPORT

At the completion of balancing, record and report the following final conditions:

- The air handling unit manufacturer's model and serial number
- Motor nameplate data
- Sheave and belt data
- Filter sizes and conditions
- Starter data and thermal overload protection sizes and ratings
- Design and actual supply airflow (by Pitot tube traverse) normal and actual economizer modes
- Design and actual return airflow (by Pitot tube traverse)
- Design and actual minimum outside airflow
- Design airflow
- Motor(s) actual voltage, current, BHP (W), and RPM
- Fan(s) design and actual RPM
- Static pressure profile and static pressure at the end of the system
- Coil capacity test with each coil set for design airflow and water flow
- Static pressure controller setpoint and inlet static pressure of remote air valves
- Fan powered boxes' motor(s) actual voltage, and current
- Outlet airflows per fan powered box (full cooling minimum and maximum, and full heating)
- Fan powered box manufacturer, size, model, heater size (if electric), design and actual airflow for full cooling with zero return, and full heating with minimum primary airflow.

A Response to "Qualitative Testing of Laboratory Fume Hoods"

Richard Miller, P.E.
Systems Testing and Analysis

I want to express my feeling regarding the article, "Qualitative Testing of Laboratory Fume Hoods" published in the Fall 2000 issue of *TAB Journal*.

In the article on page 19, it is written that the technician must stand to the side of the hood, out of the hood airflow pattern. Pages 17 and 18 photos (a photo is worth a thousand words), indicate the technician blocking the air stream by standing in front of the opening. These photos also show the hoods being used as storage cabinets. This should not be done. Photos should be utilized to indicate the placement of equipment in the hoods, so the measurements can be repeated, if required. Equipment stored in the hoods does affect airflows.

Photo from Fall 2000 issue of TAB JOURNAL

ASHRAE Applications, 1999, Chapter 13 states: the measurements should be taken with a device that is accurate in the intended operating range and "an instrument holder" should be used to improve accuracy and be able to provide repeatable results. NSF requires a stand to hold the measuring instrument. Holding in your hand is ***not*** acceptable to any lab hood testing criteria.

If photos are utilized, and I believe they enhance the article, we should be very careful that we are performing the testing correctly or we are ***not*** presenting AABC in a favorable light to persons knowledgeable and we can be thought of as incompetent.

When a hood is tested with the sash full open, every halving of open area approximates a doubling of velocity unless the fume hood is equipped with a variable volume exhaust system. The hood should then be checked at 12" opening of the sash to verify the inflow velocity has remained constant with the full open inflow velocity, namely 100 fpm.

Excessive inflow velocities are harmful and potentially dangerous. See SEFA, page 15, appendix E, general information paragraph E1.2.1 and Prudent Practices page 200, paragraph 2 which states the same about excessive turbulence. Prudent Practices states such air turbulence can cause vapors within the hood to spill out into the general laboratory atmosphere. Page 204 expands on this statement.

Industrial Ventilation pages 3-17, Section 3.7 also repeats this warning.

There is no guide that I am aware of that permits testing of an open sash approximately 2" below the bottom level holding the velocity meter by hand.

NSF states 3 heights to the sash setting of 25%, 50%, and 75% of the opening height.

We, at Systems Testing and Analysis test fume hoods with sashes at 12" heights because this is repeatable and all are done in a standard that anyone in the company can return and re-verify these measurements.

When the system is a VAV exhaust we verify open and 12" open.

The ASHRAE/ANSI 110 requires testing with the sash at various openings but this is to verify containment of the sulphur hexafluoride.

Traversing for Accuracy in a Rectangular Duct

Gaylon Richardson
Engineered Air Balance Co., Inc.

How accurate is the Equal Area Method prescribed by the Associated Air Balance Council versus the Log Tchebycheff. To determine which method was more accurate, a series of traverses were taken in a laboratory using AMCA nozzles off a wind tunnel. The duct sizes tested were 48" x 12" duct and 24" x 24". Each duct was tapped into the wind tunnel's discharge plenum. The traverses were taken 6' from the entrance of the duct and 2' from the exit. Procedures prescribed in the AABC National Standards 2001 were used.

Pitot Tube Traverses

- To accomplish repeatable traverse measurements, take the measurements in a specific, measured pattern.
- Duct size must not change in a traversed section.
- Face the Pitot tube into the airstream and parallel to the ductwork at each measurement point and measure the velocity pressures.
- Convert velocity pressure to fpm velocity before averaging. Verify the traverse is taken at standard conditions.
- Take traverse measurements at actual conditions and actual cubic feet per minute (ACFM). Correct ACFM to standard CFM (SCFM) when specified by using the density correction.
- Verify that velocity measurements are acceptable. A traverse plane is suitable for flow measurements if more than 75% of the velocity pressure readings are greater than 1/10 of the maximum velocity measurement and are not negative.
- Show all traverses in the final report which will show duct size, static pressure and corresponding velocity, duct area, and the airflow. If the traverse is taken in other than standard conditions, show barometric pressure and temperature. Show density corrections for each traverse.

Square or Rectangular Duct Traverses

- Performing a Pitot tube traverse of a square or rectangular duct, the minimum spacing of the readings in the duct, and the markings on the Pitot tube are determined using the following method:
- The minimum number of readings taken in a square or rectangular duct is four (4). This would be for a duct with the height and width under 4".

Duct Side Less Than or Equal To:	Minimum Number of Readings:
4" or less	2
15"	3
24"	4
35"	5
48"	6
63"	7
80"	8
99"	9
100"	10

- For any duct with a side greater than 100", the maximum distance between holes shall not exceed 12". For all readings, the first reading shall be located from the duct wall 1/2 the distance between readings. For example, a 12" duct width will have three (3) readings 4" apart with the first reading taken at 2" from the duct wall.

It should be noted that the AABC 2001 National Standards do not state that the traverse must be located at least 7.5 diameters downstream and 3 diameters upstream of any disturbance. The TAB Technician must use good judgement and understand velocity profiles for the traverse to be valid.

We established the following parameters to take traverses using the Equal Area Method for a minimum of 18 points and 32 points in the 48" x 12" duct, and 16 points and 24 points in the 24" x 24" duct at 1000, 1500, 2000, and 2500 FPM. We took traverses using the Log Tchebycheff method with 25 points, 36 points, and 49 points at the same velocities shown above.

FOR 48" SIDE SPACING THE READINGS WERE:

Reading #	Equal Area 18 points	Equal Area 32 points	Log T 25 points	Log T 36 points	Log T 49 points
1	4	3	$3\frac{9}{16}$	$2\frac{15}{16}$	$2\frac{9}{16}$
2	12	9	$13\frac{13}{16}$	$11\frac{1}{4}$	$9\frac{3}{4}$
3	20	15	24	21	$17\frac{9}{16}$
4	28	21	$34\frac{13}{16}$	27	24
5	36	27	$44\frac{7}{16}$	$36\frac{3}{4}$	$30\frac{7}{16}$
6	44	33	—	$45\frac{1}{16}$	$38\frac{1}{4}$
7	—	39	—	—	$45\frac{7}{16}$
8	—	45	—	—	—

FOR 24" SIDE SPACING THE READINGS WERE:

Reading #	Equal Area 18 points	Equal Area 32 points	Log T 25 points	Log T 36 points	Log T 49 points
1	3	2	$1\frac{3}{4}$	$1\frac{7}{16}$	$1\frac{1}{4}$
2	9	6	$6\frac{15}{16}$	$5\frac{5}{8}$	$4\frac{7}{8}$
3	15	10	12	$10\frac{1}{2}$	$8\frac{3}{4}$
4	21	14	$17\frac{1}{16}$	$13\frac{1}{2}$	12
5	—	18	$22\frac{1}{4}$	$18\frac{3}{8}$	$15\frac{1}{4}$
6	—	22	—	$22\frac{9}{16}$	$19\frac{1}{8}$
7	—	—	—	—	$22\frac{3}{4}$

FOR 12" SIDE SPACING THE READINGS WERE:

Reading #	Equal Area 18 points	Equal Area 32 points	Log T 25 points	Log T 36 points	Log T 49 points
1	2	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$
2	6	$4\frac{1}{2}$	$3\frac{7}{16}$	$2\frac{13}{16}$	$2\frac{7}{16}$
3	10	$7\frac{1}{2}$	6	$5\frac{1}{4}$	$4\frac{3}{8}$
4	—	$10\frac{1}{2}$	$8\frac{9}{16}$	$6\frac{3}{4}$	6
5	—	—	$11\frac{1}{8}$	$9\frac{3}{16}$	$7\frac{5}{8}$
6	—	—	—	$11\frac{1}{4}$	$9\frac{9}{16}$
7	—	—	—	—	$11\frac{3}{8}$

TRAVERSES

Traverse No. 1 Equal Area 18 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.055	940	0.060	980	0.060	980	0.065	1020	0.065	1020	0.060	980				
0.060	980	0.060	980	0.070	1060	0.065	1020	0.065	1020	0.060	980				
0.055	940	0.060	980	0.065	1020	0.070	1060	0.065	1020	0.065	1020				
	2860		2940		3060		3100		3060		2980				

Average Velocity	1000	FPM	X	Duct Area	4.00	SQ FT	=	CFM (Actual) 4000
								Flow Station CFM 3990
								Difference +10

Traverse No. 2

Equal Area 32 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.060	980	0.060	980	0.060	980	0.060	980	0.065	1020	0.060	980	0.060	980	0.060	980
0.060	980	0.060	980	0.065	1020	0.070	1060	0.065	1020	0.065	1020	0.065	1020	0.060	980
0.060	980	0.060	980	0.065	1020	0.070	1060	0.070	1060	0.065	1020	0.065	1020	0.060	980
0.055	940	0.055	940	0.065	1020	0.065	1020	0.060	980	0.060	980	0.070	1060	0.060	980
	3880		3880		4040		4120		4080		4000		4080		3920

Average Velocity	1000	FPM	X	Duct Area	4.00	SQ FT	=	CFM (Actual)	4000
								Flow Station CFM	3990
								Difference	−10

Traverse No. 3

Log T 25 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.050	895	0.055	940	0.060	980	0.060	980	0.050	895						
0.060	980	0.060	980	0.070	1060	0.060	980	0.060	980						
0.060	980	0.060	980	0.070	1060	0.065	1020	0.060	980						
0.060	980	0.060	980	0.065	1020	0.065	1020	0.060	980						
0.065	1020	0.060	980	0.060	980	0.060	90	0.055	940						
	4855		4860		5100		4980		4775						

Average Velocity	985	FPM	X	Duct Area	4.00	SQ FT	=	CFM (Actual)	3940
								Flow Station CFM	3990
								Difference	-50

Traverse No. 4

Log T 36 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.040	800	0.060	980	0.050	895	0.060	90	0.060	980	0.050	895				
0.060	980	0.065	1020	0.065	1020	0.065	1020	0.060	980	0.060	980				
0.060	980	0.060	980	0.070	1060	0.070	1060	0.065	1020	0.060	980				
0.060	980	0.060	980	0.070	1060	0.070	1060	0.065	1020	0.055	940				
0.060	980	0.060	980	0.065	1020	0.070	1060	0.065	1020	0.060	980				
0.050	895	0.055	940	0.060	980	0.060	980	0.060	980	0.055	940				
	5615		5880		6035		6160		6000		5715				

Average Velocity	985	FPM	X	Duct Area	4.00	SQ FT	=	CFM (Actual)	3940
								Flow Station CFM	3990
								Difference	-50

Traverse No. 5

Equal Area 18 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.055	940	0.060	980	0.060	980	0.065	1020	0.065	1020	0.060	980				
0.060	980	0.060	980	0.070	1060	0.065	1020	0.065	1020	0.060	980				
0.055	940	0.060	980	0.065	1020	0.070	1060	0.065	1020	0.065	1020				
	2860		2940		3060		3100		3060		2980				

Average Velocity	1000	FPM	X	Duct Area	4.00	SQ FT	=	CFM (Actual)	4000
								Flow Station CFM	3990
								Difference	+10

Traverse No. 6 Equal Area 18 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.150	1550	0.140	1500	0.140	1500	0.150	1550	0.160	1600	0.140	1500				
0.150	1550	0.140	1500	0.140	1500	0.140	1500	0.150	1550	0.140	1500				
0.150	1550	0.150	1550	0.130	1445	0.130	1445	0.140	1500	0.130	1445				
	4650		4550		4445		4495		4650		4445				

Average Velocity 1515 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 6060

Flow Station CFM 6030

Difference +30

Traverse No. 7 Equal Area 32 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.150	1550	0.150	1550	0.140	1500	0.140	1500	0.150	1550	0.160	1600	0.150	1550	0.140	1500
0.150	1550	0.150	1550	0.140	1500	0.140	1500	0.150	1550	0.150	1550	0.150	1550	0.130	1445
0.150	1550	0.150	1550	0.130	1445	0.130	1445	0.140	1500	0.140	1500	0.150	1550	0.130	1445
0.150	1550	0.140	1500	0.130	1445	0.130	1445	0.120	1385	0.130	1445	0.150	1550	0.130	1445
	6200		6150		5890		5890		5985		6095		6200		5835

Average Velocity 1510 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 6040

Flow Station CFM 6030

Difference +10

Traverse No. 8 Log T 25 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.140	1500	0.130	1445	0.140	1500	0.160	1600	0.150	1550						
0.150	1550	0.140	1500	0.140	1500	0.150	1550	0.160	1600						
0.160	1600	0.140	1500	0.130	1445	0.150	1550	0.150	1550						
0.160	1600	0.140	1500	0.130	1445	0.140	1500	0.140	1500						
0.150	1550	0.130	1445	0.130	1445	0.130	1445	0.130	1445						
	7800		7390		7335		7645		7645						

Average Velocity 1515 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 6060

Flow Station CFM 6030

Difference -30

Traverse No. 9 Log T 49 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.130	1445	0.140	1500	0.140	1500	0.150	1550	0.160	1600	0.150	1550	0.110	1330		
0.150	1550	0.150	1550	0.140	1500	0.150	1550	0.160	1600	0.150	1550	0.130	1445		
0.150	1550	0.150	1550	0.140	1500	0.150	1550	0.150	1550	0.160	1600	0.130	1445		
0.150	1550	0.140	1500	0.140	1500	0.140	1500	0.150	1550	0.150	1550	0.130	1445		
0.150	1550	0.140	1500	0.130	1445	0.130	1445	0.140	1500	0.150	1550	0.120	1385		
0.150	1550	0.140	1500	0.130	1445	0.130	1445	0.140	1500	0.140	1500	0.130	1445		
0.120	1385	0.130	1445	0.120	1385	0.110	1330	0.120	1385	0.120	1385	0.110	1330		
	10580		10545		10275		10370		10685		10685		9825		

Average Velocity 1490 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 5960

Flow Station CFM 6030

Difference -70

Traverse No. 10

Equal Area 18 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.290	2155	0.270	2080	0.260	2040	0.280	2120	0.280	2120	0.270	2080				
0.290	2155	0.260	2040	0.240	1960	0.270	2080	0.270	2080	0.250	2005				
0.280	2120	0.250	2005	0.240	1960	0.240	1960	0.250	2005	0.240	1960				
	6430		6125		5960		6160		6205		6045				

Average Velocity 2050 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 8200

Flow Station CFM 8100

Difference +100

Traverse No. 11

Log T 25 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.250	2005	0.240	1960	0.290	2155	0.280	2120	0.250	2005						
0.280	2120	0.250	2005	0.280	2120	0.280	2120	0.250	2005						
0.290	2155	0.240	1960	0.260	2040	0.270	2080	0.250	2005						
0.280	2120	0.240	1960	0.250	2005	0.270	2080	0.240	1960						
0.260	2040	0.240	1960	0.210	1835	0.240	1960	0.230	1920						
	10440		9845		10155		10360		9895						

Average Velocity 2030 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 8120

Flow Station CFM 8100

Difference +20

Traverse No. 12

Equal Area 18 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.410	2565	0.360	2405	0.370	2435	0.400	2535	0.410	2565	0.370	2435				
0.420	2595	0.360	2405	0.350	2370	0.380	2470	0.400	2535	0.380	2470				
0.420	2595	0.370	2435	0.340	2335	0.340	2335	0.370	2435	0.360	2405				
	7755		7245		7140		7340		7535		7310				

Average Velocity 2465 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 9860

Flow Station CFM 9680

Difference -180

Traverse No. 13

Log T 25 Points Width: 48 Height: 12

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.370	2435	0.370	2435	0.410	2565	0.440	2655	0.310	2230						
0.400	2535	0.370	2435	0.390	2500	0.410	2565	0.350	2370						
0.420	2595	0.350	2370	0.380	2470	0.390	2500	0.350	2370						
0.420	2595	0.340	2335	0.350	2370	0.370	2435	0.340	2335						
0.370	2435	0.340	2335	0.300	2195	0.340	2335	0.330	2300						
	12595		11910		12100		12490		11605						

Average Velocity 2430 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 9720

Flow Station CFM 9680

Difference +40

TRAVERSES 14-17

Traverse No. 14 Equal Area 16 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.055	940	0.060	980	0.070	1060	0.065	1020								
0.060	980	0.070	1060	0.080	1135	0.080	1135								
0.065	1020	0.070	1060	0.080	1135	0.075	1095								
0.060	980	0.060	980	0.060	980	0.055	940								
	3920		4080		4310		4190								

Average Velocity 1030 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 4120

Flow Station CFM 3980

Difference +140

Traverse No. 15 Log T 25 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.050	895	0.060	980	0.065	1020	0.070	1060	0.060	980						
0.060	980	0.065	1020	0.080	1135	0.080	1135	0.070	1060						
0.060	980	0.075	1095	0.075	1095	0.080	1135	0.080	1135						
0.060	980	0.070	1060	0.075	1095	0.080	1135	0.070	1060						
0.050	895	0.060	980	0.055	940	0.060	980	0.055	940						
	4730		5135		5285		5445		5175						

Average Velocity 1030 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 4120

Flow Station CFM 3980

Difference +140

Traverse No. 16 Equal Area 16 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.120	1385	0.130	1445	0.150	1550	0.140	1500								
0.130	1445	0.160	1600	0.170	1650	0.170	1650								
0.150	1550	0.170	1650	0.170	1650	0.170	1650								
0.140	1500	0.140	1500	0.140	1500	0.130	1445								
	5880		6195		6350		6245								

Average Velocity 1540 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 6160

Flow Station CFM 6050

Difference +110

Traverse No. 17 Log T 25 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.110	1330	0.120	1385	0.140	1500	0.150	1550	0.130	1445						
0.130	1445	0.140	1500	0.160	1600	0.170	1650	0.160	1600						
0.140	1500	0.160	1600	0.170	1650	0.180	1700	0.170	1650						
0.140	1500	0.160	1600	0.160	1600	0.170	1650	0.150	1550						
0.110	1330	0.130	1445	0.130	1445	0.140	1500	0.130	1445						
	7105		7530		7795		8050		7690						

Average Velocity 1525 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 6100

Flow Station CFM 6050

Difference +50

Traverse No. 18 Equal Area 16 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.200	1790	0.230	1920	0.300	2195	0.210	1835								
0.250	2005	0.290	2155	0.330	2300	0.270	2080								
0.270	2080	0.330	2300	0.340	2335	0.240	1960								
0.240	1960	0.300	2195	0.340	2335	0.240	1960								
	7835		8570		9165		7835								

Average Velocity 2090 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 8360

Flow Station CFM 8045

Difference +315

Traverse No. 19 Log T 25 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.180	1700	0.200	1790	0.270	2080	0.280	2120	0.180	1700						
0.220	1880	0.250	2005	0.310	2230	0.300	2195	0.240	1960						
0.270	2080	0.300	2195	0.330	2300	0.320	2265	0.240	1960						
0.250	2005	0.310	2230	0.340	2335	0.330	2300	0.220	1880						
0.200	1790	0.240	1960	0.330	2300	0.300	2195	0.210	1835						
	9455		10180		11245		11075		9335						

Average Velocity 2050 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 8200

Flow Station CFM 8045

Difference +155

Traverse No. 20 Equal Area 16 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.310	2230	0.340	2335	0.420	2595	0.370	2435								
0.380	2470	0.430	2625	0.490	2805	0.420	2595								
0.390	2500	0.510	2860	0.510	2860	0.420	2595								
0.350	2370	0.480	2775	0.470	2745	0.340	2335								
	9570		10595		11005		9960								

Average Velocity 2570 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 10280

Flow Station CFM 9825

Difference +455

Traverse No. 21 Log T 25 Points Width: 24 Height: 24

V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.	V.P.	VEL.
0.280	2120	0.320	2265	0.420	2595	0.430	2625	0.300	2195						
0.330	2300	0.370	2435	0.480	2775	0.470	2745	0.320	2265						
0.390	2500	0.460	2715	0.500	2830	0.500	2830	0.360	2405						
0.340	2335	0.480	2775	0.520	2890	0.480	2775	0.300	2195						
0.290	2155	0.390	2500	0.460	2715	0.420	2595	0.310	2230						
	11410		12690		13805		13570		11290						

Average Velocity 2510 FPM X Duct Area 4.00 SQ FT = CFM (Actual) 10040

Flow Station CFM 9825

Difference −215

TRAVERSE SUMMARY

48" x 12" Duct at 1000 FPM

	Equal	Equal	Log T	Log T	Log T
	Area 18 Pts	Area 32 Pts	Area 25 Pts	Area 36 Pts	Area 49 Pts
Traverse #	1	2	3	4	5
Traverse	4000	4000	3940	3940	3980
Nozzle	3990	3990	3990	3990	3990
Difference	10	10	-50	-50	-10
% off from Nozzle	0.25	0.25	-1.25	-1.25	-0.25

48" x 12" Duct at 1500 FPM

	Equal	Equal	Log T	Log T
	Area 18 Pts	Area 32 Pts	Area 25 Pts	Area 49 Pts
Traverse #	6	7	8	9
Traverse	6060	6040	6060	5960
Nozzle	6030	6030	6030	6030
Difference	30	10	30	-70
% off from Nozzle	0.50	0.17	0.50	-1.16

48" x 12" Duct at 2000 FPM

	Equal	Log T
	Area 18 Pts	Area 25 Pts
Traverse #	10	11
Traverse	8200	8120
Nozzle	8100	8100
Difference	100	20
% off from Nozzle	1.23	0.25

24" x 24" Duct at 1000 FPM

	Equal	Log T
	Area 18 Pts	Area 25 Pts
Traverse #	14	15
Traverse	4120	4120
Nozzle	3980	3980
Difference	140	140
% off from Nozzle	3.52	3.52

48" x 12" Duct at 2500 FPM

	Equal	Log T
	Area 18 Pts	Area 25 Pts
Traverse #	12	13
Traverse	9860	9720
Nozzle	9680	9680
Difference	180	40
% off from Nozzle	1.86	0.41

24" x 24" Duct at 1500 FPM

	Equal	Log T
	Area 18 Pts	Area 25 Pts
Traverse #	16	17
Traverse	6160	6100
Nozzle	6050	6050
Difference	110	50
% off from Nozzle	1.82	0.83

24" x 24" Duct at 2000 FPM

	Equal	Log T
	Area 18 Pts	Area 25 Pts
Traverse #	18	19
Traverse	8360	8200
Nozzle	8045	8045
Difference	315	155
% off from Nozzle	3.95	1.93

24" x 24" Duct at 2500 FPM

	Equal	Log T
	Area 18 Pts	Area 25 Pts
Traverse #	20	21
Traverse	10280	10040
Nozzle	9825	9825
Difference	455	215
% off from Nozzle	4.63	2.19

CONCLUSION

There is not over 2% difference in the two methods of Equal Area versus Log Tchebycheff. Equal Area is easier to use and is the accepted method by the Associated Air Balance Council. It should be noted that the readings were taken with an analog manometer and all velocities and CFM readings were rounded to the nearest 5" for easier use of the numbers and manometers cannot measure with any greater accuracy.

TAB journal

Classified Advertisement Reservation

Deadlines

Issue	Materials Deadline
Spring 2001	January 15, 2001
Summer 2001	April 15, 2001
Fall 2001	July 15, 2001
Winter 2002	October 15, 2001

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Classified ads are $50 for the first 100 words, and $1.50 for each additional word. There are three methods of payment available. 1.) You may fill out the credit card information below, and fax to the AABC Headquarters. 2.) Send a check or money order by mail, along with a copy of this reservation form, to the AABC Headquarters. 3.) Check the "Bill Me" option below. This option is only available to AABC members

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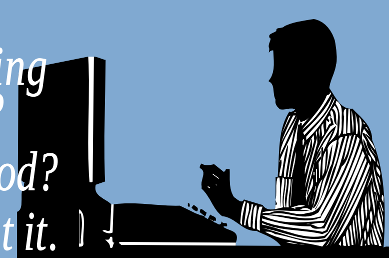

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