TAB:
Fact and Fiction

Should a testing, adjusting, and balancing (TAB) professional rely solely on experience?

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Over the years, certain beliefs have taken hold in the HVAC and testing, adjusting, and balancing (TAB) communities. Clearly, some of these beliefs are rooted in scientific principles and are valuable tokens of wisdom. Others, however, though they may have a factual basis, can be fiction if misapplied. This article examines three longstanding beliefs in the TAB industry.

Fact or fiction: Mathematical rounding of airflow measurements is falsification of data.

Fiction. Is it better to record a measurement exactly as it is reported by an instrument, or is it more appropriate to round it to the closest increment of 5 (or 10) cfm before putting it on a TAB report?

The first consideration in the analysis of this issue is the accuracy of the measurement. Take, for example, an airflow hood. The stated accuracy of a commonly used brand is ±3 percent of a reading, plus an additional 7 cfm. This means that actual airflow could be higher or lower than measured airflow by the quantity of:

$$\left(\text{measured airflow}\right)\times\left(0.03\right)+7$$

Therefore, a measured airflow of 100 cfm would have a calibrated accuracy no greater than ±10 cfm, or ±10 percent.

As can be seen in Table 1, until airflow readings become larger (near 300 cfm), the accuracy of an airflow measurement by a flow hood is no greater than ±6 percent. This issue is pervasive, whether airflow, velocity, or pressure is measured. For instance, a velocity reading of 1,004 fpm using the same instrument (but with a pitot tube) has an actual

Measured airflow, cfm	± range, cfm	Low cfm	High cfm	Actual accuracy, ± percent
10	7.3	2.7	17.3	73
25	7.8	17.3	32.8	31
50	8.5	41.5	58.5	17
75	9.3	65.8	84.3	12
100	10.0	90.0	110.0	10
125	10.8	114.3	135.8	9
150	11.5	138.5	161.5	8
175	12.3	162.8	187.3	7
200	13.0	187.0	213.0	7
250	14.5	235.5	264.5	6
300	16.0	284.0	316.0	5
350	17.5	332.5	367.5	5
400	19.0	381.0	419.0	5
500	22.0	478.0	522.0	4
750	29.5	720.5	779.5	4
1,000	37.0	963.0	1,037.0	4

TABLE 1. Actual accuracy of airflow measurements using a flow hood.

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accuracy of ±37 fpm. While this leads to an improved actual accuracy of 3.7 percent, questions pertaining to rounding—Would a reported velocity of 1,005 fpm be any less accurate? What about 1,010 fpm?—remain.

A second consideration regarding rounding is repeatability. One can expect a different measurement almost every time a reading is taken. Table 2 shows the results of a rudimentary field test, which included four repeated trials of four individual airflow measurements (taken with the technician facing east, west, north, and south) for a single four-way-pattern ceiling diffuser in a constant-volume system. Out of 16 airflow readings, only 11 were unique. Of those that repeated, only one was measured three times (296 cfm). Note that this test was repeated with another ceiling diffuser with similar results.

Trial	Direction N	E	S	W
1	313	287	308	296
2	296	283	300	313
3	310	309	281	293
4	308	283	296	286

TABLE 2. Sample airflow readings from a ceiling diffuser using an airflow hood.

Between measurement accuracy and repeatability, it can be stated that an exact translation of data from instrument to paper is no more accurate than its rounded counterpart. It could be argued that the reporting of an exact value implies an accuracy that does not exist.

A final consideration concerns the theory of significant figures. Significant figures ensure that reported accuracy is no better than the limitations of the data or the equipment used to obtain the results. While the application of significant figures would provide a scientific answer to this problem, the issues of measurement accuracy and repeatability remain.

In conclusion, mathematical rounding of airflow data is an acceptable practice. However, it can be deployed only if the TAB professional's policy is logical, known, and applied consistently.

Fact or fiction: The standard ±10-percent balancing requirement can lead to a system that does not meet its functional requirements.

Fact. However, when this is the case, the cause rarely is an insufficient effort on the part of the TAB professional.

There are growing numbers of specifications that require final airflows to be within a smaller range of deviation than the standard ±10 percent. Some remove the low range entirely and call for +10 percent, 0 percent. Based solely on the previous section's discussion of accuracy and repeatability, these new requirements are questionable. But why is an accepted industry standard being changed? Perhaps for the following reasons:

• The designed airflow quantity is the minimum airflow required for acceptance by the owner (acceptance criteria).

This is common for mechanical systems in pharmaceutical and other "high-end" applications. The client's acceptance criteria are clear from the beginning of the project. They usually include a minimum-supply-airflow requirement, stated in terms of air changes per hour (ACH). Failure to meet this requirement might render the system unusable.

Recognizing the critical nature of this criteria (along with temperature, relative humidity, pressurization, etc.), one should design and construct a system to be capable of something greater than the minimum value for acceptance. For example, if the requirement is 15 ACH, a system should be designed for no less than 17 ACH. If it is, TAB procedures using the 10-percent standard would be sufficient.

• The belief that low airflow is worse than high airflow.

This is not always true. For example, higher airflow will reduce a system's ability to dehumidify and maintain required dew-point levels. If cooling-coil velocities are above 500 fpm, an overflow condition could increase the likelihood of condensate carryover, creating a ponding effect in the fan section. Higher airflow will create higher pressure loss and consume more fan energy than lower airflow. The application will determine whether a high tolerance is more critical than a low one or if both are equally critical. There is no standard.

The American National Standards Institute; the American Society of Heating, Refrigerating and Air-Conditioning Engineers; the National Environmental Balancing Bureau; and the Sheet Metal and Air Conditioning Contractors' National Association have established the ±10-percent standard. If a project requires closer adherence to design quantities than normal, ensure that the project is designed and built to account for it. In short, modify the project, not the industry.

Fact or fiction: Outside airflow is determined by subtracting return airflow from supply airflow.

Depends. Mathematically, this is true in most installations. As such, it almost has become the default technique used by many TAB professionals because supply- and return-airflow measurements can be obtained in most installations with reasonable accuracy, whereas the same is not true of outside airflow. However, in more complex installations, misuse of this simplified version of a general mass-balance equation will lead to error.

For the system shown in Figure 1, the following equations are valid:

$$Q_{RA1} = Q_{EA} + Q_{RA2}$$
$$\left(\text{equivalent to } Q_{RA2} = Q_{RA1} - Q_{EA}\right)$$
$$Q_{SA} = Q_{RA2} + Q_{OA}$$
$$Q_{SA} = Q_{RA1} + Q_{OA} - Q_{EA} \left(\text{by substitution}\right)$$

As can be seen, neglecting exhaust airflow (Q_{EA}) in determining outside airflow (Q_{OA}) would lead to a system with an insufficient amount of outside airflow. This mistake may seem unlikely with the benefit of Figure 1. However, if provided with Figure 2, it seems more plausible.

Actually, there is no mechanical difference between these two systems. However, Figure 2 describes the system less clearly, increasing the chance of error.

Another downside of this method is that measurement error is additive. In other words, the product of multiple measurements has greater potential for error than direct measurements.

For example, assume that design supply-, return-, and outside-airflow quantities are 10,000, 8,500, and 1,500 cfm, respectively. The supply and return ducts are the same size: 36 in. by 26 in. (6.5 sq ft). The TAB professional decides to determine outside airflow by subtracting the recirculation airflow (Q_{RA2}), measured at Point 4, from the supply air (Q_{SA}), measured at Point 11. After initial readings and multiple adjustments, final airflows are determined to be within 10 percent of design at 10,010 and 8,450 cfm, which results in an outside airflow of 1,560 cfm, which also is within 10 percent of the design. However, when the instrument's velocity-measurement error ([measured velocity] x [0.03] + 7 fpm) is considered, the supply airflow could range from 9,650 to 10,350 cfm, while the return airflow could be between 8,200 and 8,800 cfm. As such, the actual outside airflow could be between 850 and 2,150 cfm, which is well outside of the 10-percent range.

The point of this analysis is not to discredit the subtraction method, but rather to point out its limitations. Presented below are some additional methods of determining outside airflow:

• Direct measurement. The most reliable method of determining outside airflow is direct measurement via an airflow traverse. In Figure 1, the traverse would take place at Point 5a or 5b. In practice, this method typically is unavailable because of insufficient lengths of ductwork upstream and downstream of suitable traverse locations.

Although outside airflow can be measured directly through an alternate method, including the use of an averaging pitot-tube array or differential-pressure measurement of louvers or dampers, these methods are less reliable and should be avoided, unless confirming data are obtained.

• Recirculation-air-pressure measurement. This technique allows the TAB professional to determine outside airflow from only supply airflow and recirculation-air static pressure. Step 1 in this method is to seal the outside-air duct or intake (points 5a, 5b, and 5c in Figure 1) so there is no possibility of air entering through the outside-air damper when the fan is on. With the fan on, the next step is to measure supply airflow and the recirculation-air static pressure directly upstream of the mixing section (Point 4 in Figure 1). This has established the flow/pressure relationships (C_v, K, or C constants).

Clearly, in this initial measurement, Q_{RA2} is equal to supply airflow. Using the squared relationship between flow and pressure (affinity laws), the TAB professional can determine the design pressure at Point 4 at design recirculation airflow. As such, and through iteration, the outside-air damper can be opened until the design pressure at Point 4 is met. Confirming that supply airflow is at design quantity (±), the TAB professional can be confident that recirculation airflow and outside airflow are at design quantities (±).

• Mixed temperature measurement. This method allows calculation of outside airflow from outside-air, recirculation-air, and supply-air temperatures and recirculation and supply airflows according to the following equation:

$$Q_{OA} = \frac{\left[(T_6)(Q_{SA}) + (T_4)(Q_{RA2})\right]}{T_{5c}}$$

A problem with this method is temperature stratification in the mixing section (Point 6 in Figure 1), where recirculation air and outside air combine. Should this method be the most appropriate for the application, stratification can be overcome by taking a temperature traverse and calculating the weighted average using the following equation and Table 3.

$$T_{avg\text{-}wtd} = \left[(T_{1A})(V_{1A}) + (T_{1B})(V_{1B})\right] + \left[(T_{2A})(V_{2A})\right] + \left[(T_{3A})(V_{2A})\right] + \left[(T_{4A})(V_{4A})\right] \div \text{sum of all velocity readings}$$

Outside airflow then would be calculated using the following equation:

$$Q_{OA} = \frac{\left[(T_{avg\text{-}wtd})(Q_{SA}) + (T_4)(Q_{RA2})\right]}{T_{5c}}$$

Note that these calculations assume a uniform temperature profile for the return air (Point 4) and outside air (Point 5c). If this assumption is incorrect, then these also will require the use of weighted-average temperature in lieu of single-point measurements.

SUMMARY

Most beliefs, whether in the TAB industry or any other, are based on fact under a certain set of circumstances. As such, when confronted with those circumstances, one should use the beliefs whenever possible to reduce the time and effort required to complete a task. However, misapplication of techniques based on the beliefs can lead to erroneous results. Unfortunately, there is no book on how to avoid all of the potential pitfalls. The best chance a professional has is to understand the operation of a system and its requirements and be open to asking for assistance when required. The professional's worst chance is to believe that 20 or 30 years of experience equals perfection.

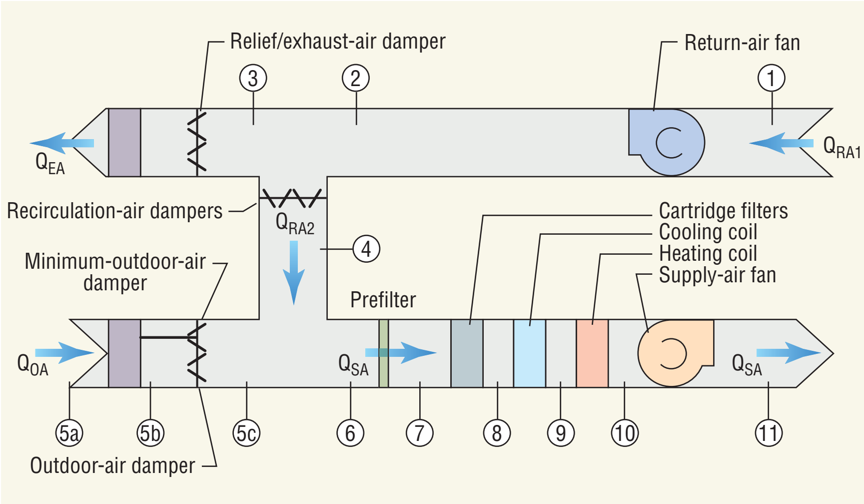

FIGURE 1. Air-handling unit.

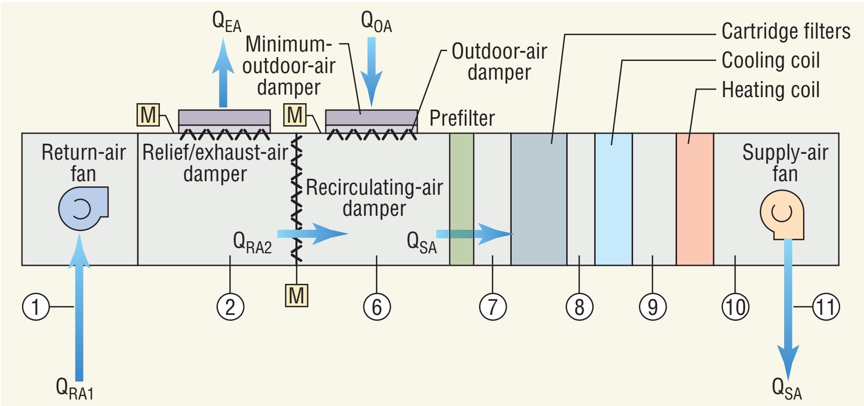

FIGURE 2. Alternate schematic of air-handling unit in Figure 1.

		Data	Horizontal position			
			A	B	C	D
Vertical position	1	Temperature, F	65	70	72	68
		Velocity, fpm	500	500	550	600
	2	Temperature, F	65	68	68	65
		Velocity, fpm	500	425	600	625
	3	Temperature, F	50	45	50	52
		Velocity, fpm	550	475	490	575
	4	Temperature, F	35	30	30	35
		Velocity, fpm	500	375	400	400

TABLE 3. Sample temperature traverse sheet.