RESIDENTIAL AIRSIDE TESTING AND BALANCING

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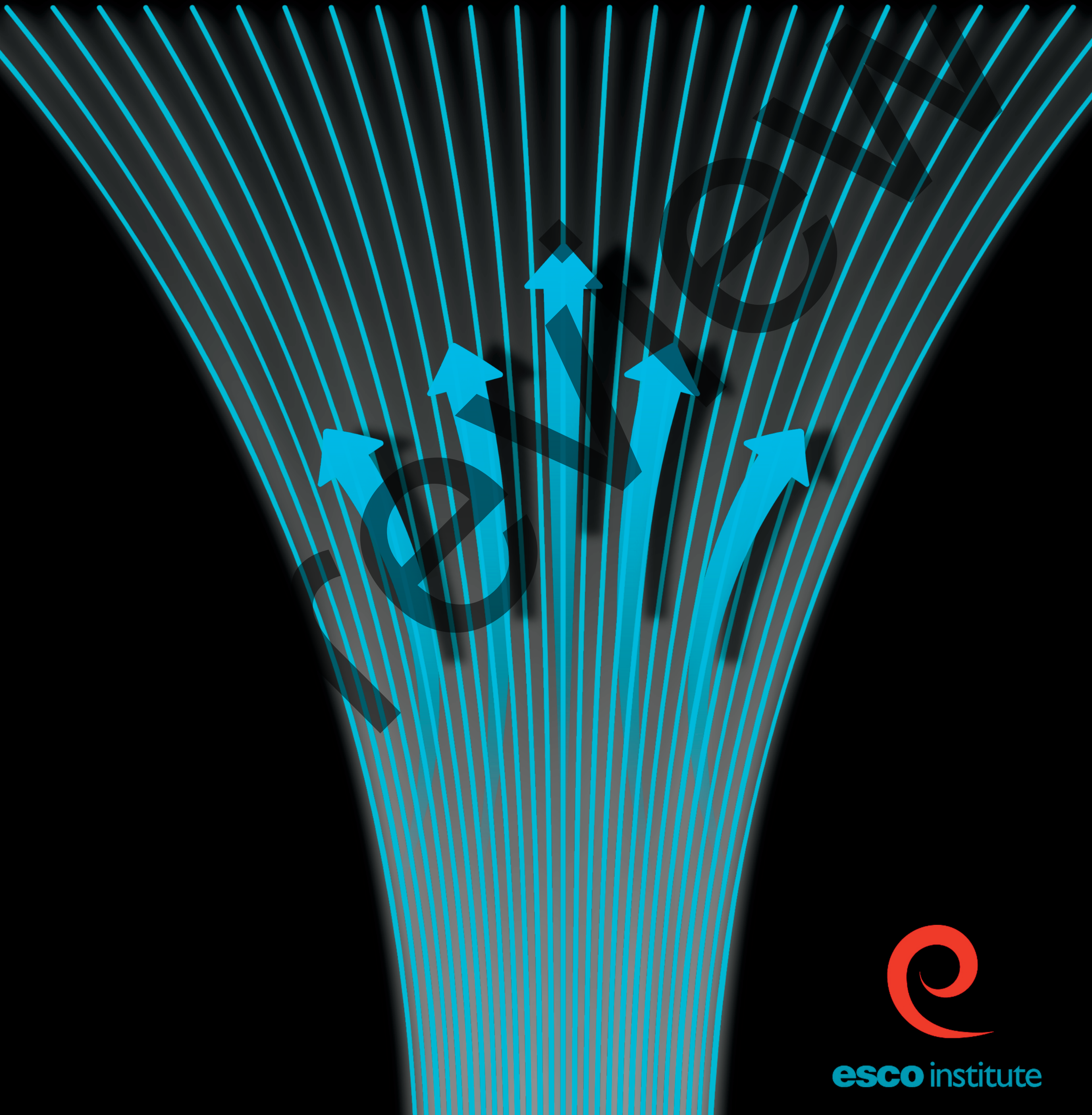

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This manual is designed to serve as a practical introduction to residential airside testing and balancing (TAB), providing technicians with the foundational knowledge and skills needed to measure and adjust airflow in residential HVAC systems. Proper airflow is essential to system performance, comfort, efficiency, and longevity, yet it is often one of the least understood aspects of HVAC installation and service.

As HVAC systems have evolved, so has the need for technicians to verify system performance through measurement. Just as combustion engines rely on correct fuel-air ratios to operate efficiently, HVAC systems rely on proper airflow to deliver the heating and cooling performance they were designed to provide. Without verifying airflow, even well-installed equipment may not perform to its full potential.

Training programs face the challenge of covering an ever-expanding body of knowledge within limited time. As a result, airflow testing and balancing has sometimes received less emphasis than other topics. However, as the industry continues to advance toward higher efficiency standards and improved comfort expectations, the ability to measure and adjust airflow is becoming an increasingly valuable skill for technicians.

By introducing the principles and practical methods of residential airside testing and balancing, this manual aims to help technicians better understand how air moves through a system and how proper measurement can lead to improved performance, comfort, and reliability. Integrating TAB concepts into HVAC training programs and field practices helps ensure that systems operate as intended and that technicians are equipped to deliver the highest level of service to their customers.

Looking back 50 years, effective and accurate testing and balancing (TAB) of residential air-conditioning and heating systems was, at best, difficult to achieve. The tools and instruments that were available in the 1970s, for example, were hard to use and the math required to translate the obtained readings and measurements into usable airflow data often incorporated significant errors. However, by the 1990s the quality of the available instrumentation greatly improved in accuracy, ease of use, and the algorithms used to perform internal calculations.

Now, more than ever, more advanced technologies have been integrated into the systems that are installed, making accurate and precise TAB measurements necessary for all HVAC installations. New-generation hi-tech tools and instruments have traditionally been expensive, however the cost of good testing equipment is coming down. Further, the costs associated with not using them and not properly testing and balancing our systems far exceed to cost of doing things right the first time. With proper instrumentation, it is far easier to accurately measure and predict how an HVAC system will operate under most conditions year-round. Potential system problems and deficiencies that are likely to occur are more easily identified and addressed before a customer complaint is made. Over the past few years, affordable accurate tools and test instruments that can be used for residential TAB have become available. Skilled TAB technicians can no longer afford to be without them.

The primary goal of this manual is to show technicians how, with the use of reliable and reasonably priced instrumentation, to verify that a residential HVAC system is operating as designed and within the equipment manufacturer's design specifications. Effective methods for measuring airflow will be shared along with step-by-step procedures to help ensure accurate, reliable, and repeatable results. It is strongly recommended that that more than one method or instrument be used for verification, especially if there are if there are any questions about the reliability of the first method or about an instrument's accuracy.

The first part of each section will cover the basic information needed to do the testing and balancing. For those going into more depth those basic instructions are followed in the section by the theory behind the hands-on field methods. At the end of each chapter in this book, there is a short quiz to test your understanding of the concepts covered. In addition, there is a hands-on project that is intended to help drive these concepts home.

As you proceed through this book, the concepts discussed will build on previously introduced topics in a logical, progressive manner. It is our hope that, by the time you complete this manual, you should be able to test and balance airflow in a residential HVAC system accurately and with confidence.

In residential air conditioning and heating systems, air is commonly used as the medium that acts as the conduit through which heat is either added to, or removed from, the structure. For this reason, it is important to make sure that the proper amount of air flows into and out of each room in a home. Figure 1-1 illustrates how air flows through a piece of air-conditioning equipment.

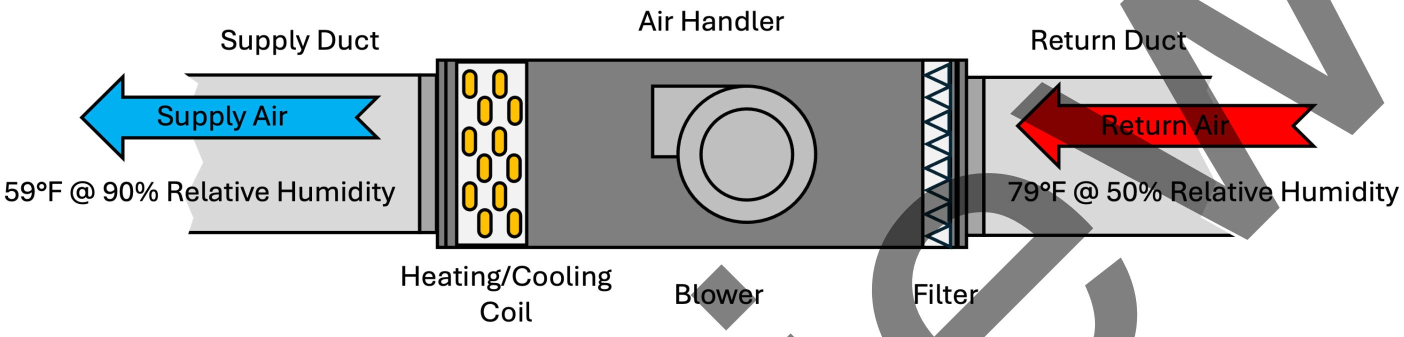


Figure 1-1: Sample airflow through an air handler for a cooling application.

Air returning to the equipment from the conditioned space is referred to as return air. At a dry-bulb temperature of 79°F and 50% relative humidity, return air flows back into the air handling equipment via the return portion of the duct system. At the air handler, the air is filtered and then passed through the blower and cooling in this example (or heating coil). The air, once cooled, leaves the air handler at a temperature of 59°F and a relative humidity of 90% or more. This cooled airstream, referred to as the system's supply air, is then sent into the occupied space. The supply air, once discharged to the space through supply registers, grilles and/or louvers, mixes with the air in the space.

A home's temperature may or may not be the same throughout the structure because the heating and cooling loads change based on occupancy, the sun's position, wind, and other factors. HVAC systems are designed for a snapshot in time, and system balancing is based on heating and cooling load calculations, which often require different amounts of air. A well-balanced system will maintain all of the rooms in a home as close to the thermostat's set point as possible.

Heat Transfer in Air-Conditioning and Heating Systems

When an airstream passes through a cooling coil, the dry-bulb temperature of the airstream is reduced. When an airstream passes through a heating coil, the dry-bulb temperature of the airstream is increased. Whenever the temperature of the airstream changes, a sensible heat transfer takes place. The general formula that is used to calculate the amount of sensible heat transferred in a process is:

$$\text{Sensible heat in Btuh} = 1.08 \times \text{airflow in cubic feet per minute} \times \text{air temperature change}$$

The 1.08 value is a standard air conversion factor.

From the Field

Homeowner to Technician:	“The upstairs bedrooms are hot in the summer and cold in the winter”
Technician to Homeowner:	“I’ll be more than happy to investigate this situation for you. Is there anything else I need to know before I begin?”
Homeowner to Technician:	“Yes. Another HVAC company told me that I need a new system because the one I have is undersized. It’s funny, though, as the company that told me that is the same company that sold me the systemin the first place! I would like to get you expert opinion before I make a decision.”

The technician performs his inspection. After about 30 minutes, the technician approaches the homeowner.

Technician to Homeowner:	“I finished my inspection and did not notice any return grilles upstairs.
Homeowner to Technician:	“And what does that mean.”
Technician to Homeowner:	“Without any return grilles, there is no pathway for the air upstairs to get back to the HVAC equipment” This is very likely the cause for inadequate cooling in the summer and heating in the winter.”
Homeowner to Technician:	“So, what do you suggest?”
Technician to Homeowner:	“Before we do anything major, let’s try something simple first. Tonight, leave the upstairs bedroom doors open and let us know within a day or two if that has any effect on the comfort issue.”

Two days later…

Homeowner to Technician:	“Your suggestion worked great! The upstairs was completely comfortable. However, I don’t want to keep the doors open all the time. What do you suggest?”
Technician to Homeowner:	“We have a few options for you and the good news is that, whichever solution you choose, you’ll be spending much less money than if you had to replace your entire system. We’ll prepare a couple of estimates for you and send them over within a day or two.”
Homeowner to Technician:	“Thank you! I look forward to receiving more information on the system modifications.”

8. Using the psychrometric chart, answer the following questions based on the following information:
   - Return air: 80°F dry-bulb temperature @ 40% relative humidity
   - Supply air: 60°F dry-bulb temperature @ 90% relative humidity
   - Airflow rate: 800 CFM

   a) Calculate the Q_S, in Btuh, for this system.

   b) Determine the ΔW, in grains/lb for this system.

   c) Calculate the Q_L, in Btuh, for this system.

   d) Calculate the Q_T, in Btuh, for this system.

   e) Calculate the Q_T for this system if the house was located at an altitude of 3,000 ft.

Field Exercise

Measure the supply and return air conditions on an operating system. If the airflow rate can be calculated or measured, use the obtained value. Otherwise, use 1,00 CFM for this exercise. Then:

- plot the supply and return air conditions on a psychrometric chart.
- calculate the ΔT, ΔW, and Δh values for the system.
- calculate Q_S.
- calculate Q_L.
- calculate Q_T.

Airflow Grids

Airflow grids, Figure 2-7, are designed to slide into the filter slot found in most residential forced-air air-conditioning and heating systems. To use an airflow grid to measure the aiflow rate through a furnace or air handler, the supply duct's static pressure must first be determined. This is accomplished by operating the system's blower on high speed and measuring the static pressure in the supply duct with the system's existing filter in place. The system's filter is then replaced with the flow grid. The system's actual cfm value is then calculated by the grid and its associated smart device application using the pressure drop across the flow grid, the supply duct static pressure with the flow grid installed, and additional information that is provided about the system configuration. Such additional information includes the following:

- Is the airflow being measured through an air handler or a furnace?
- What is the orientation of the unit? (Upflow, downflow, horizontal)
- What is the cooling capacity of the system?
- Where is the filter located? (In the equipment's filter slot or a filter grille)
- What are the climate conditions? (Humid: 350 cfm/ton, or Dry: 425 cfm/ton)

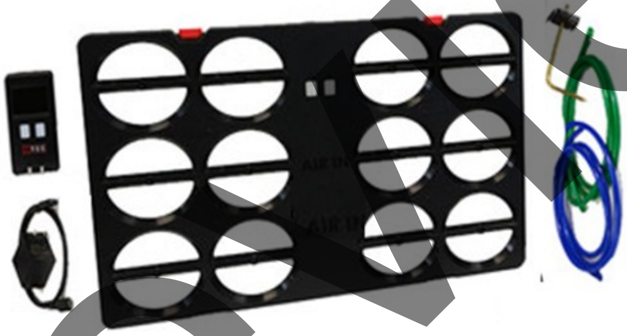


Figure 2-7: *Airflow grid.*

Duct Leakage Tools

Duct leakage tools, Figure 2-8, are used to pressurize sealed duct systems. Based on the amount of air that must be added to maintain the required pressure in the duct system, the amount of duct leakage is calculated. Duct leakage test equipment can also be used to measure airflow, in cfm. when the ducts are not sealed for pressure testing. If the system is determined to be operating with a large duct system leak rate, the air distribution system must be repaired before attempting to test and balance the system. Testing and balancing cannot be successfully done on a leaky duct system.

Figure 2-8: *Duct leakage test equipment.*

Duct System Pressures

The three pressures that are used to evaluate the performance of a duct system are the static pressure (SP), the velocity pressure (VP), and the total pressure (PT). The duct system's total pressure is equal to the mathematical sum of the system's static pressure and the system's velocity pressure. Duct system pressures are typically expressed in inches of water column, or inWC, where 1 psi is equal to 27.7 inWC. The following three formulas can be used to calculate any of the three pressures if the other two values are known:

TOTAL PRESSURE (TP) = VELOCITY PRESSURE (VP) + STATIC PERESSURE (SP)

VELOCITY PRESSURE (VP) = TOTAL PRESSURE (TP) - STATIC PERESSURE (SP)

STATIC PRESSURE (SP) = TOTAL PRESSURE (TP) - VELOCITY PERESSURE (VP)

Anatomy of a Pitot Tube

The pitot tube is constructed as two concentric tubes that allow for the direct measurement of a duct system's total and static pressures, Figure 2-11. The center tube has an open end that points directly into the airstream. The outer tube has a series of holes on its side and measure the pressure that is perpendicular to the direction of airflow.

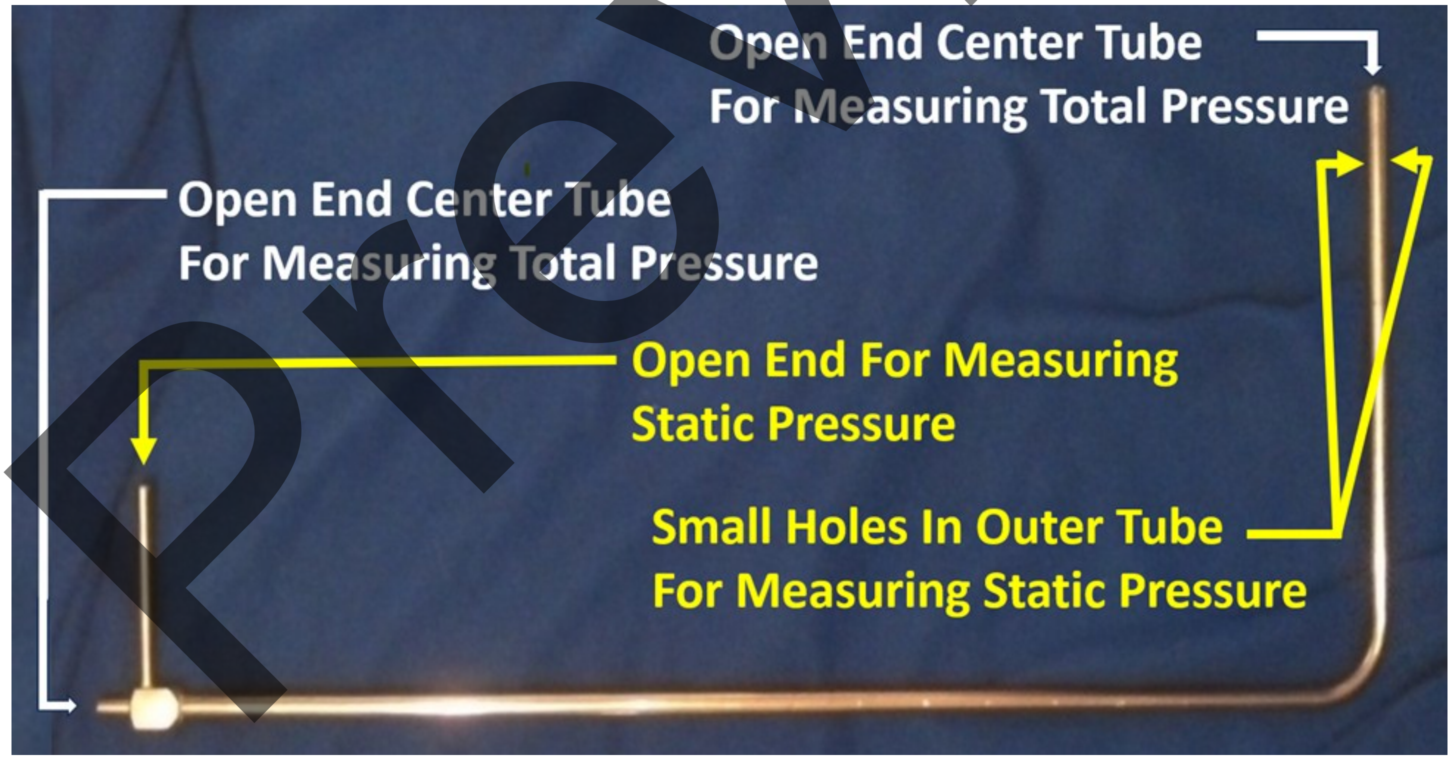


Figure 2-11: *The pitot tube is constructed as two concentric tubes.*

Using the result of this calculation, the damper can be closed while measuring the static pressure in the duct reaches the 0.236 IWC value. This eliminates the need to continually adjust the damper and measure the airflow, saving valuable time. ***Note:*** *this method of adjusting airflow to individual main supply branches will save you time, especially when balancing larger duct systems with many supply diffusers on them.*

Fan Law 3

The third fan law states that as the airflow rate (CFM) through a duct system changes, the motor's brake horsepower (BHP) will change. The formula for brake horsepower is:

$$BHP_{\text{TARGET VALUE}} = BHP_{\text{BEGINNING AS FOUND}} \left(CFM_{\text{NEW CFM}} \div CFM_{\text{BEGINNING AS FOUND}}\right)^3$$

Brake horsepower, by definition, is a measurement of a motor's power at the shaft before losing power due to resistance created by any components that are connected to the motor's shaft. In practice, the brake horsepower value is the amount of power a motor has before any devices or couplings are connected to it.

Sample Calculation

Consider that the airflow through a duct has increased from 500 cfm to 800 cfm. If the BHP at the 500 cfm airflow rate is 0.25, what will the BHP be once the airflow rate is increased?

From the information provided, we know the following:

- $BHP_{\text{BEGINNING AS FOUND}} = 0.25$
- $CFM_{\text{BEGINNING AS FOUND}} = 500\ CFM$
- $CFM_{\text{NEW CFM}} = 800\ CFM$

Substituting these known values into the formula for BHP_2, we get the following:

$$BHP_{\text{TARGET VALUE}} = BHP_{\text{BEGINNING AS FOUND}} \left(CFM_{\text{NEW CFM}} \div CFM_{\text{BEGINNING AS FOUND}}\right)^3$$

$$BHP_{\text{TARGET VALUE}} = 0.25 \times (800 \div 500)^3$$

$$BHP_{\text{TARGET VALUE}} = 0.25 \times (800 \div 500)^3$$

$$BHP_{\text{TARGET VALUE}} = 0.25 \times (1.6)^3 \text{ or } 1.6 \times 1.6 \times 1.6$$

$$BHP_{\text{TARGET VALUE}} = 0.25 \times 4.09$$

$$BHP_{\text{TARGET VALUE}} = 1.02$$

This result shows that, by increasing the airflow by 60% (from 500 CFM to 800 CFM), the BHP increased to a value that is over four times the original value. This indicates that pushing more air through a smaller duct will often require a larger blower motor operating at a higher speed. This is due to the higher static pressure in the duct. To resolve this situation without having to increase motor size or speed, it is often much easier, and in some cases, less expensive to increase the duct's cross-sectional area than increasing blower motor size and speed.

Tales From the Field

After repairing a very leaky duct system, the technician measured the airflow through the air handler and found it was below the manufacturer's minimum requirement. The tighter duct system had a higher static pressure, lowering the airflow rate through the system. After making sure all of the supply registers were open and the filter and coils were clean, it was determined that upgrades to the duct system would be required before the system could be balanced.

Fan Law #2

The second fan law states that as the airflow rate (CFM) through a duct system changes, the duct's static pressure (P) will change proportionally. Formula: $(CFM_1 \div CFM_2)^2 = (SP_1 \div SP_2)$

The relationship that exists between airflow and static pressure is:

$$(CFM_1 \div CFM_2)^2 = (SP_1 \div SP_2)$$

Where CFM_1 is the initial airflow rate
CFM_2 is the final airflow rate
SP_1 is the initial static pressure
SP_2 is the final static pressure

Note: this formula works only for positive static pressure values. As such, it is not intended to be used to evaluate the return air of an air-conditioning system.

Depending on which value must be determined, this fan law can be expressed to solve for each of the four variables as follows:

$$SP_2 = SP_1 \div (CFM_1 \div CFM_2)^2$$
$$SP_1 = SP_2 \text{ x } (CFM_1 \div CFM_2)^2$$
$$CFM_1 = CFM_2 \text{ x } (SP_1 \div SP_2)^{1/2}$$
$$CFM_2 = CFM_1 \text{ x } (SP_1 \div SP_2)^{1/2}$$

Note: the ½ exponent represents a square root and can be written as: $\sqrt{SP_1 \div SP_2}$

Tales From the Field

Once the balancing process has been completed, it is recommended that the arms or handles on the balancing dampers be screwed in place to prevent others from tampering with the damper positions. At a bare minimum, it is recommended that the final position of a damper's arm or handle be identified with a permanent marker. This way, if adjustments are made to the damper's position, the final post-balancing position can be easily restored.

Fan Law #3

The third fan law states that as the airflow rate through a duct system changes, the motor's break horsepower will change.

Formula: The relationship that exists between airflow and break horsepower is:

$$(CFM_1 \div CFM_2)^3 = BHP_1 \div BHP_2$$

Where CFM_1 is the initial airflow rate
CFM_2 is the final airflow rate
BHP_1 is the initial blower motor break horsepower
BHP_2 is the final break blower motor break horsepower

Overview of the Traversing Process

Traversing duct remains one of the most accurate ways to establish the total airflow passing through a duct. Calculating the airflow through a duct system or portion of the duct system is a straightforward, two-step process:

1. The average velocity of the airstream, expressed in feet per minute (FPM), is measured inside a duct.
2. The average velocity of the airstream is multiplied by the cross-sectional area of the duct, which is expressed in square feet (ft^2), to obtain the airflow rate, which is expressed in cubic feet per minute, or CFM.

In Chapter 2, the tools needed to properly traverse a duct were introduced. Traditionally, meters like an oil manometer or the analog pressure differential gauge only measured pressure differentials. The average pressure differentials were then converted to velocity, in feet per minute, using formulas and air correction factors. For quick, easy velocity measurements, it is strongly recommended that a digital meter with internal correction capabilities for altitude and air temperature be used for residential traverse applications. Some meters have the ability to record the individual measurements and provide an average for all of the readings.

Factors That Affect Airflow

There are many factors that affect airflow through a duct system. Although the technician performing the testing and balancing work may not be the one who designed and/or installed the air distribution system, it is important for the technician to be aware of the conditions that can negatively affect the obtained readings. Airflow can be affected by factors that include:

- Duct wall friction
- Duct sizes, shapes and configurations
- Duct fittings such as offsets, elbows, and transitions
- Turbulence created by filters, fans, and blowers

To help ensure that readings are as accurate as possible, it is strongly recommended that velocity readings be taken in straight duct runs in which airflow is laminar, which simply means smooth, parallel and orderly. For example, on a round duct system, readings should ideally be taken at a distance of over 7 duct diameters distance downstream and over 3 diameters upstream of any disturbance, Figure 4-3.

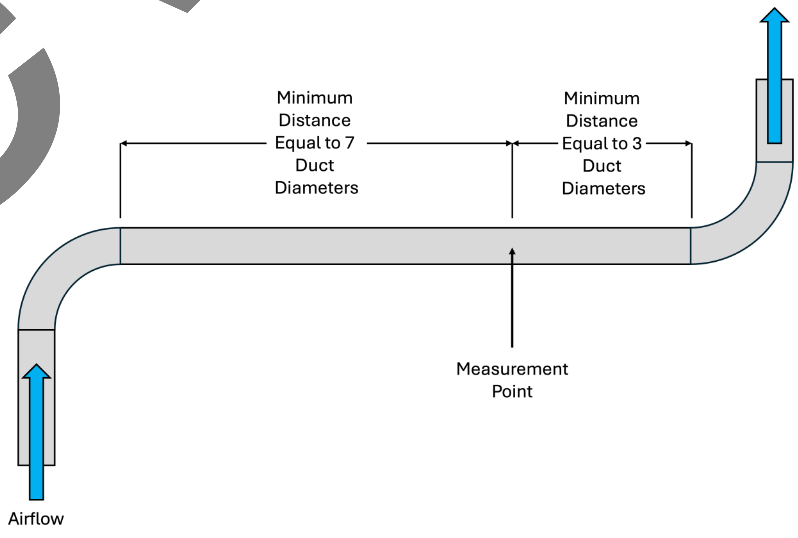
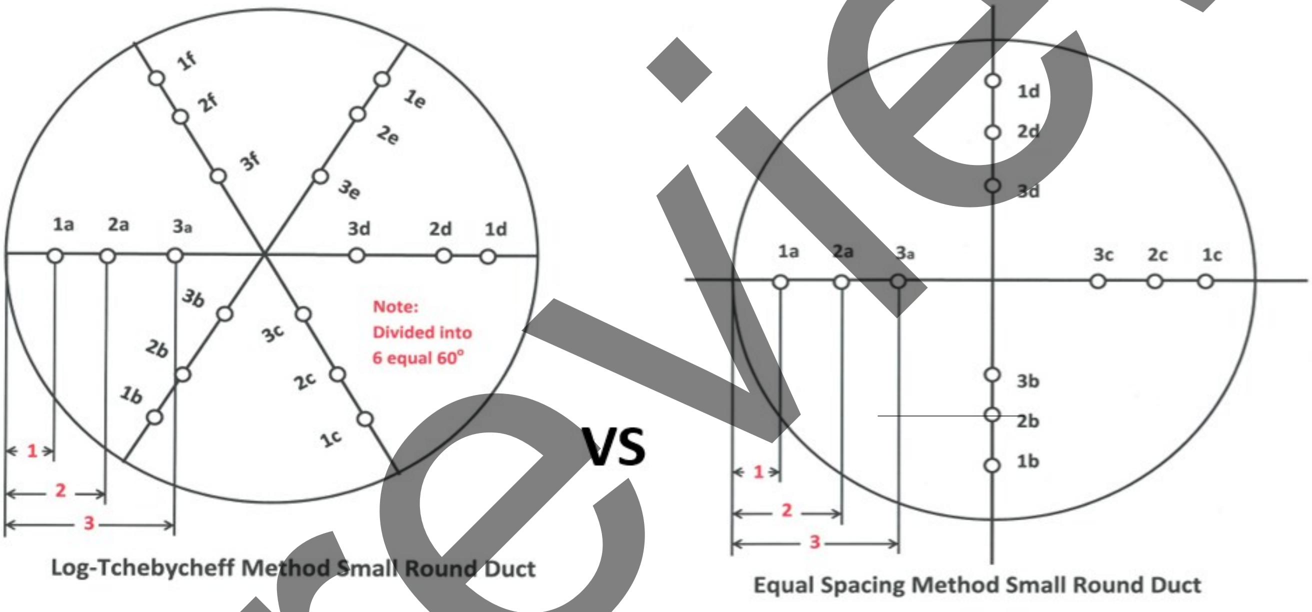
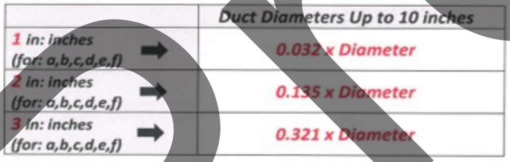


Figure 4-3: Minimum distances for accurate velocity measurements.

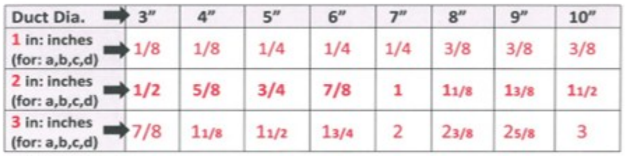
Traversing is not difficult, but finding a straight duct run that has enough clearance may be a difficult task. However, for residential air distribution systems, these distance guidelines can often be overlooked provided that none of the obtained readings (pressure or feet per minute) are negative.

When performing a duct traverse, it is important to get accurate measurement values. In the previous section the equal spacing method was shown. It is accurate enough for residential applications. However, technicians should know that there are two widely accepted methods for laying out the traverse patterns that will be followed; the Log-Tchebycheff method and the equal spacing method, Figure 4-4. The layouts shown are for small round ducts with diameters up to 10”. Larger ducts require more measurement points. The Log-Tchebycheff method is considered to be more accurate but, for residential applications, the equal spacing method has a degree of accuracy that is high enough for these applications. As such, this will be the method discussed in this book. In addition, the equal spacing method is easier to execute in the field because it requires that fewer holes be made in the duct and, as indicated in Figure 4-11, requires fewer readings.



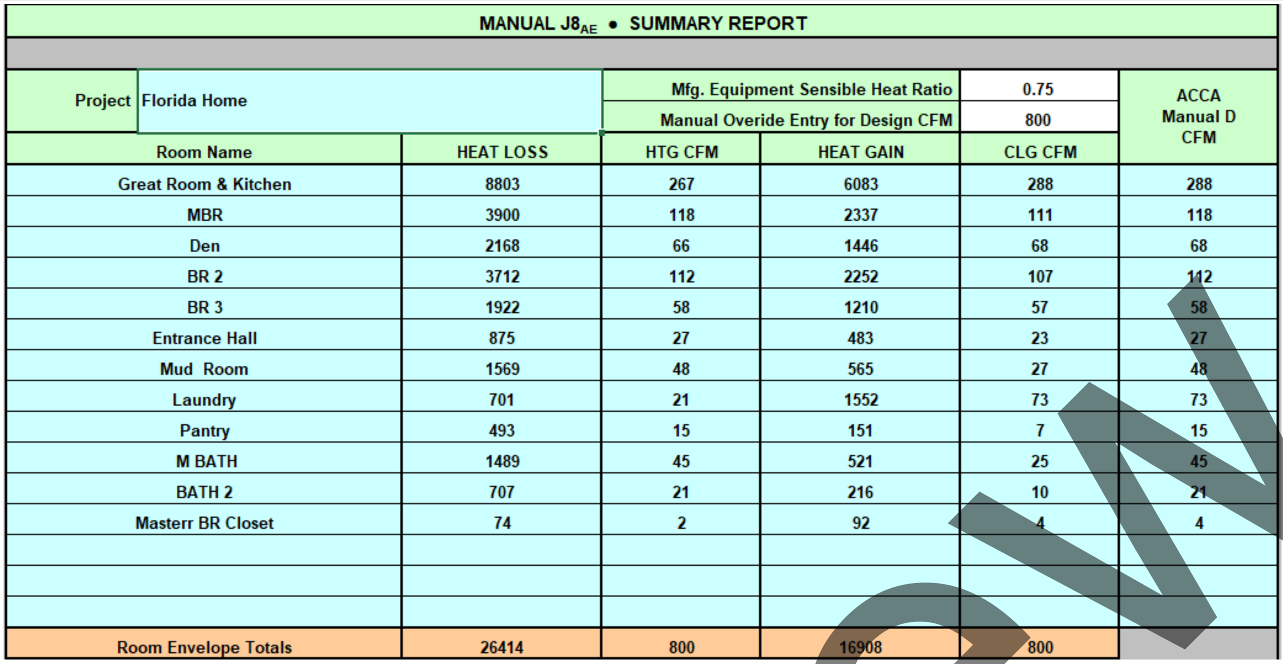


	Duct Diameters Up to 10 inches
1 in: inches (for: a,b,c,d,e,f) ➡	0.032 x Diameter
2 in: inches (for: a,b,c,d,e,f) ➡	0.135 x Diameter
3 in: inches (for: a,b,c,d,e,f) ➡	0.321 x Diameter



Duct Dia. ➡	3”	4”	5”	6”	7”	8”	9”	10”
1 in: inches (for: a,b,c,d) ➡	1/8	1/8	1/4	1/4	1/4	3/8	3/8	3/8
2 in: inches (for: a,b,c,d) ➡	1/2	5/8	3/4	7/8	1	1 1/8	1 3/8	1 1/2
3 in: inches (for: a,b,c,d) ➡	7/8	1 1/8	1 1/2	1 3/4	2	2 3/8	2 5/8	3

Figure 4-11. *(Left) The Log-Tchebycheff method. (Right) The equal spacing method.*



MANUAL $J8_{AE}$ • SUMMARY REPORT

Project	Florida Home		Mfg. Equipment Sensible Heat Ratio	0.75	
			Manual Overide Entry for Design CFM	800	

Room Name	HEAT LOSS	HTG CFM	HEAT GAIN	CLG CFM	ACCA Manual D CFM
Great Room & Kitchen	8803	267	6083	288	288
MBR	3900	118	2337	111	118
Den	2168	66	1446	68	68
BR 2	3712	112	2252	107	112
BR 3	1922	58	1210	57	58
Entrance Hall	875	27	483	23	27
Mud Room	1569	48	565	27	48
Laundry	701	21	1552	73	73
Pantry	493	15	151	7	15
M BATH	1489	45	521	25	45
BATH 2	707	21	216	10	21
Masterr BR Closet	74	2	92	4	4
Room Envelope Totals	26414	800	16908	800	

Figure 5-2: Estimated airflow rates based on heat gain and heat loss information.

Balancing Values

System designers will typically provide TAB technicians with the heat gain and heat loss summary sheet, the duct design paperwork, and a diffuser and grille value that reflects the 10% overage and the difference between heating and cooling values. The actual balancing value used by the technician is generally somewhere between the HTG CFM and CLG CFM values provided on the summary sheet. This subjectivity is why TAB work is often considered to be an art solution based on science. Generally adding the heating and cooling airflow values and dividing by two will provide a good starting point for establishing balancing values. When the final balance is completed, it should be within 10% of the values the TAB technician targets for each supply register and return grille. For example, the airflow for the great room and kitchen area, as indicated in Figure 5-2, ranges from 267 CFM to 288 CFM. The TAB technician would most likely balance this space at 278 CFM, which is midway between these two values.

Now, consider the mud room in Figure 5-2. This space has a heating airflow of 48 CFM and a cooling airflow of 27 CFM. An air balancer might balance the space closer to the 25 CFM, as indicated for cooling, since the mud room is close to an entrance. Because it is an entry point into the structure, this room will feel warm anyway when entering the house during the winter. As such, the lower airflow rate will likely suffice. In humid areas, where there is the likelihood of over cooling the area near an exterior door, selecting airflow rates on the lower end of the range is also likely. This is because higher airflow rates, in moister climates, can over cool rooms, and encourage mold growth. This happens when there is a high relative humidity and the room surfaces become cold enough to reach dewpoint. On the other hand, if the home is in a dry climate, the TAB technician would likely have selected a higher airflow rate to keep the room comfortable in the longer cooling season.

Tales From the Field

Referring to Figure 5-2, it can be seen that the closet in the master bedroom requires only 4 CFM of airflow. Running even the smallest of supply ducts to that space will most likely result in the overcooling of this area in the summer and the overheating of the area in the winter, even if the volume damper in that duct section is almost fully closed. To help get airflow to that space without overheating or overcooling this small space, it is

Duct Wheels and Slide Rules

Duct wheels, slide rules, and smart device applications are often used be TAB technicians to evaluate duct sizing when they are having air distribution and/or balancing issues, Figure 5-6. These devices provide the technician with airflow rate, air velocity, and duct size information at a given friction rate. With a friction rate of 0.15 for a 100-foot duct run, the duct wheel is able to adjust and provide values for duct runs with different equivalent lengths. For example, a duct run with an equivalent length of 400 feet, will have losses of 0.60 inWC, which is four times that of the 100-foot value. Similarly, a duct run with an equivalent length of 60 feet, will have losses of 0.0.9 inWC, which is 60% of the 0.15 inWC value.

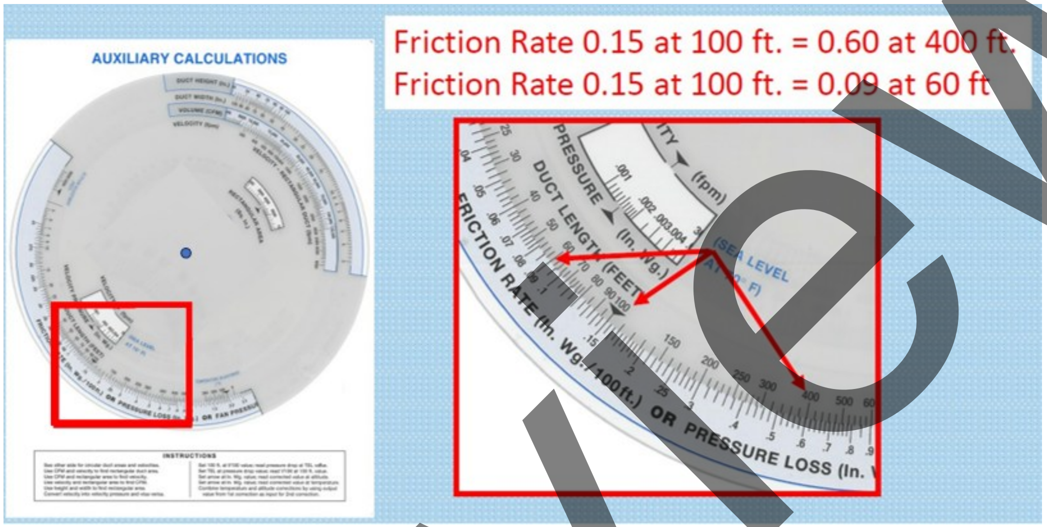


Figure 5-6: Sample friction rate adjustment for duct wheel values.

Summary

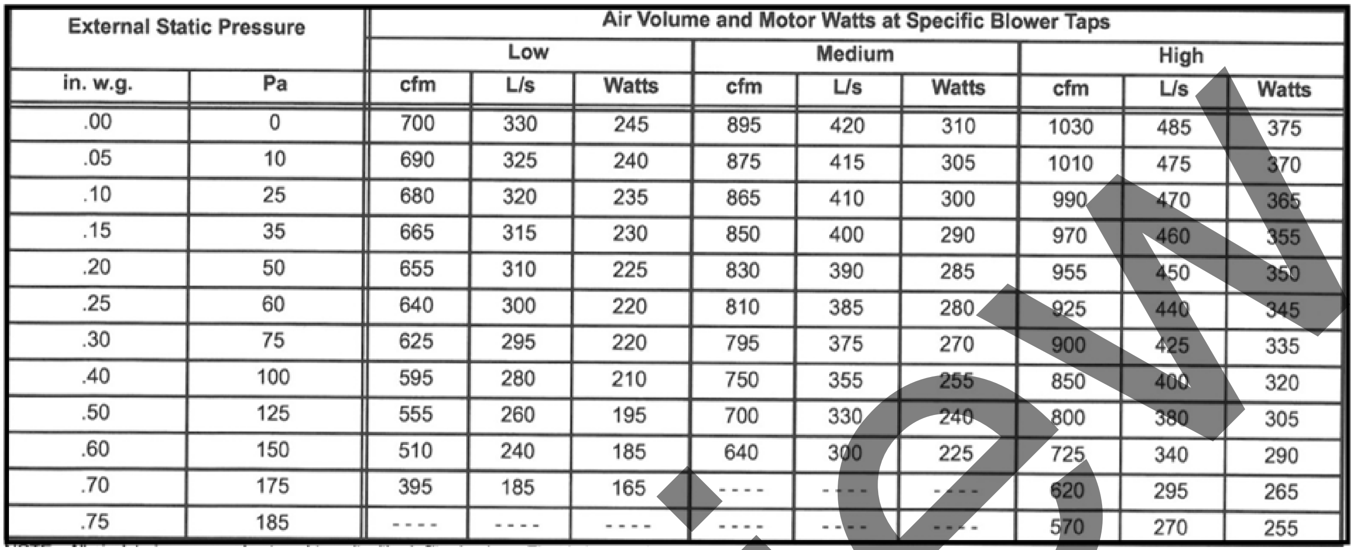
Testing and balancing (TAB) work is required to properly heat and cool a home. It is important that the airflow passing through the air handler is verified and is correct. With improper airflow, there is no way to verify that the refrigerant charge. Newly installed equipment loses approximately 30% of its efficiency due to improperly following and adhering to manufacturer's instructions, local codes and best industry practices.

Many leading contractors realize that performing system testing and balancing translates directly into fewer callbacks, increased profits, and improved customer satisfaction, even though there are higher up-front expenses associated with performing TAB work.

TAB technicians should understand how duct sizing controls how much air can be delivered to a space. Over time this understanding will grow, and most TAB technicians will be able to recommend design changes when they are needed using field measurements and duct wheel, slide rules, or smart device applications. The math and the number of calculations will seem overwhelming to many technicians, but the results are well worth the effort.

One important aspect of system testing and balancing is knowing the system's external static pressure (ESP) total component losses (CPL), available external static pressure (ASP), and friction rate. The friction rate refers to the resistance to flow for each 100 feet of equivalent duct length and is calculated by dividing the available static pressure by the total equivalent duct length and then multiplying this result by 100. The friction rate can then be used to size and evaluate air distributions systems.

Figure 6-3 shows a typical manufacturer’s airflow table and the differential pressure meter set up to measure ESP. Please always remember that the fan speed must be known before the table can be used. The filters and equipment coils need to be as clean as possible to ensure accurate readings.



External Static Pressure		Air Volume and Motor Watts at Specific Blower Taps								
		Low			Medium			High		
in. w.g.	Pa	cfm	L/s	Watts	cfm	L/s	Watts	cfm	L/s	Watts
.00	0	700	330	245	895	420	310	1030	485	375
.05	10	690	325	240	875	415	305	1010	475	370
.10	25	680	320	235	865	410	300	990	470	365
.15	35	665	315	230	850	400	290	970	460	355
.20	50	655	310	225	830	390	285	955	450	350
.25	60	640	300	220	810	385	280	925	440	345
.30	75	625	295	220	795	375	270	900	425	335
.40	100	595	280	210	750	355	255	850	400	320
.50	125	555	260	195	700	330	240	800	380	305
.60	150	510	240	185	640	300	225	725	340	290
.70	175	395	185	165	- - - -	- - - -	- - - -	620	295	265
.75	185	- - - -	- - - -	- - - -	- - - -	- - - -	- - - -	570	270	255

Figure 6-3: Typical manufacturer’s ESP table and differential meter setup to measure ESP.

It is important for technicians to know how to use the ESP/CFM tables in the manufacturer’s installation manuals. Also included in those manuals is how to change and set the fan speed. The charts will have notations on wet or dry coil and, if there is no air temperature noted, the standard air temperatures should be used. On the table in Figure 6-4, airflow for an ESP of 0.5 inWC varies by fan speed. At high speed, the airflow is 800 CFM, at medium speed, the airflow is 700 CFM, and at low speed, the airflow drops to 555 CFM.

The table is fairly linear so CFM values between table values can be found by interpolation. Interpolation accurately calculates the value between table values. A close enough value between table values can generally be estimated by technicians. For example, again referring to Figure 6-3, at medium fan speed and an ESP of 0.35 inWC, the airflow rate will be between 750 CFM and 795 CFM. Since 0.35 the exact midpoint between the ESP values of 0.3 and 0.4, the value at 0.35 can be interpolated as follows: (795 + 750 ÷ 2) = 1545 ÷ 2 = 772.5 CFM. For residential balancing, either 772 CFM or 773 CFM would be okay.

Don’t Forget the Paperwork!

No TAB project is complete until the paperwork is done It is important to fill out the TAB forms, Figure 6-4, completely and neatly. If there are more diffusers and grilles than there are places to record them on one form, two or more forms can be used. The set of forms should have page numbers put on all of the connected forms that show the page number and the number of pages. For example: Page 1 of 3, Page 2 of 3 and page 3 of 3.

TAB reports include supply air and a return air forms for every room. TAB Forms all have information identifying the tools used, the home location (identifier) and date of the TAB Testing along with spaces for the measurements to be recorded and evaluated.

Conclusion

TAB reports are not difficult to fill out. However, it is recommended that they be filled out in pencil or an erasable pen when you are starting out in TAB. Errors will be made and when they are found it is easier to erase one number than it is to rewrite the entire TAB report.

For the air handler/furnace report ,recording the method of testing includes recording all of the information the selected test method requires. For example, using the temperature differential method for combustion appliances numerous measurements may be required beyond the supply air and return air temperatures. For example, the combustion fuel pressures and the fuel Btuh content should be included to make sure the equipment is operating within the manufacturer’s chart’s specifications.

Always double check the math on TAB forms to make sure the column addition is done correctly. As noted before, the best practice is to enter the number into the calculator and then to verify it is correct before hitting the enter key. It is also a best practice to always do the math twice to make sure the correct answer gets recorded on the forms. Any TAB technician that tells you that they never make a math error when adding columns just didn’t find them!

TAB technicians should also verify that the duct system leakage rate is within acceptable ranges. They could use the air handler/furnace and the final balance totals to show there is no duct leakage and write that up on the sheets under notes. However, most duct leakage testing is done by sealing up return grilles and supply diffusers and recording measurements using a calibrated fan. Each tool manufacturer will have instruction and procedures that must be followed, and also have numerous measurements that have to be recorded on the TAB report. It is the TAB technician’s responsibility to read the instructions and follow them. All measured values need to be included in the TAB report.

Introduction

In this chapter we will look at how a TAB technician uses the various design airflow (CFM) values to establish the final TAB target values. Some decisions will need to be made with regards to how the home and its individual rooms will be used. Because there is no way to accurately predict how a homeowner will use their home, the final airflow rates delivered to each room may not perfectly align with the loads in these spaces. However, rest assured that the final CFM values will work for the vast majority of applications and uses.

Keep in mind that the goal of this manual is to help you become a better TAB technician and practitioner. To that end, if you understand, and can put into practice, even 70% of the balancing principles, practices, and procedures discussed in this manual, you will indeed be taking one step closer to mastering your craft. However, this manual was written so that you can easily adopt 100% of the practices in it and become a true residential TAB expert! To help you retain 100% of the airside balancing information, three important things that we've mentioned earlier in this book are worth covering again:

- When balancing a system, make sure the evaporator coil is clean and the air filter is clean and in like-new condition
- Make certain you verify that the air handler is delivering enough air. If there is not enough air to balance, the system can't be balanced!
- Before balancing a system, verify that a duct leakage test was done, as you'll need to establish that there is less than 10% leakage.

ANSI/ACCA 5 QI

The American National Standards Institute (ANSI) and the Air Conditioning Contractors of America (ACCA) have jointly established standards for designing, installing, and verifying residential HVAC systems. When airflow in a residential system is balanced to within 10% of the design airflow values, the minimum airflow requirements established in ANSI/ACCA 5 QI (*HVAC Quality Installation Specification)* will be exceeded. ANSI/ACCA 5 QI requirements state that airflow rates are to be within 20% of the design value.

The reason for the less stringent balancing values in the 5 QI standard is to leave "extra room" for inspectors whose job it is to verify and certify that a balance was done correctly. With these guidelines in place, projects will more easily pass the balancing inspection. Since different meters perform differently and the cleanliness levels of filters and coils at the time of testing may not be easily duplicated during a verification test, it was determined by ANSI and ACCA that the additional 10% was an acceptable tradeoff for evaluating the original system balance.

A Word About Design CFM Values

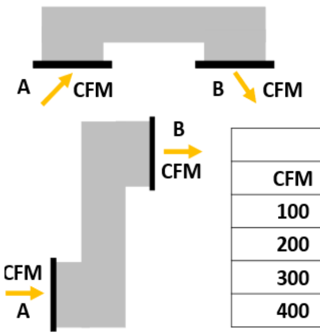
What follows is an example of how the balancing CFM values are established before the balancing process can begin. Figure 7-1 shows a load value worksheet (from Manual J), which includes CFM and the duct design values (from Manual D) in CFM, which are used for duct sizing. The column totals for the whole system CFM are also provided.

Return Duct Design Values

Return air pathways are critical for an HVAC system to move the correct amount of air back to the air handler from the areas being serviced. Always make sure there are return ducts. contrary to popular belief, return ducts can be balanced just like the supply ducts! However, there one critical difference; the static pressure fan law formula shortcut method will not work on return ducts because they operate with negative pressures.

Transfer Return Duct Design Values

Figure 7-5 show typical transfer duct sizing design values recommended in Manual D as well as their CFM ratings. To help ensure a high-quality balancing project, transfer ducts and grilles should also be verified to be working properly. Figure 7-6 shows the Department of Energy requirements for sizing transfer grilles and the acceptable pressure drop across a closed room door.



The A-B Component Pressure Drop (CPL):
- A-B Pressure drop is 0.05 IWC
- Two return grilles 0.02 IWC or less.
- Duct size based on grille frame size.
- Two 90^0 fittings EL of 15 ft or less.

Example Grille -Airway Size			
CFM	**Size**	**Size ft^2**	**FPM**
100	12”× 8”	0.666	150
200	14”× 10”	0.972	206
300	18”× 12”	1.50	200
400	20”× 12”	1.666	240

Figure 7-5: Transfer duct sizing chart based on Manual D.

The A-B Component Pressure Drop (CPL):
- A-B Pressure drop is 0.05 IWC
- Two return grilles 0.025 IWC or less.
- Duct size based on grille frame size.

Room Pressure Differential
To main part of house &
bedroom doors closed fan
on max-operating speed:
≥ -0.012 IWC ≤ 0.012 IWC
For rooms over 150 CFM
acceptable level:
≥ -0.020 IWC ≤ 0.020 IWC

Example Grille -Airway Size			
CFM	**Size**	**Size ft^2**	**FPM**
100	12”× 8”	0.666	150
200	14”× 10”	0.972	206
300	18”× 12”	1.50	200
400	20”× 12”	1.666	240

Figure 7-6: DOE transfer grille test requirements.

Conclusion

Balancing residential HVAC systems requires target supply and return CFM values for all areas. It also requires a return air pathway from all rooms or areas that can be isolated from a common hall return by closing a door. This section covered how the balancing CFM are established so TAB technicians would have an understanding of how they are calculated. As TAB technician’s career advances learning how to use a duct slide rule to fix design errors can save time and money for their contractor. TAB technician’s increase their earning potential and career opportunities by learning everything they can about the design and installation of HVAC systems.

airflow rate in this branch is significantly higher than the design requirements for this brance and is, therefore, a great branch to start on. As a matter of fact, the airflow in this branch is 28.5% (480 ÷ 373.5 × 100 = 128.5%) higher than needed.

Room Name	Heating CFM	Cooling CFM	Average CFM	AS Found CFM	Balanced CFM
Duct Branch 3					
Great Room & Kitchen	268	288	278	300	
Mud Room	46	25	35.5	40	
Pantry	14	6	10	40	
Laundry	20	80	50	100	
Totals	348	399	373.5	480	

Figure 8-2: *Airflow information for Branch 3 from Chapter 7.*

Proportional Balance Step 1:

Go to the two diffusers that are farthest from the main branch takeoff. For this example, it will be assumed that the diffusers listed in Figure 8-2 are organized, from top down, from the farthest from the main duct to the closest. From this figure, we obtain the following information:

Diffuser 1 (Great Room and Kitchen) Design CFM: 278 CFM
Diffuser 1 (Great Room and Kitchen) Measured CFM: 300 CFM
Diffuser 2 (Mud Room) Design CFM: 35.5 CFM
Diffuser 2 (Mud Room) Measured CFM: 40 CFM

Substituting these values into the base calculation guidelines in Figure 8-1, we get the following:

1A 300 ÷278 = 1.079 $\quad \frac{\textit{Diffuser 1 Measured CFM}}{\textit{Diffuser 1 Design CFM}}$

1B 40 ÷ 35.5 = 1.13 $\quad \frac{\textit{Diffuser 2 Measured CFM}}{\textit{Diffuser 2 Design CFM}}$

1C (1.079 + 1.13) ÷ 2 = 1.10 $\quad \frac{\textit{Line 1A + Line 1 B}}{2}$

1D 278 ×1.10 = 305 *Diffuser 1 Design CFM × Line 1 C*

1 E 35.5 × 1.10 = 39 *Diffuser 2 Design CFM × Line 1 C*

From these calculations, it can be seen that the target airflow values are 305 CFM for diffuser 1 and 39 CFM for diffuser 2. Since the measured CFM value for diffuser 1 is 300 CFM and well within the acceptable 10% range, no adjustment to this diffuser is needed. Similarly, no adjustment will need to be made to diffuser 2 but making a small adjustment here is recommended. Why? The other two diffusers on this branch will, based on the information in Figure 8-2, need to be closed, resulting in increased airflow through diffusers 1 and 2.

Name: ____ Date:_________________________

Read and answer the questions below.

1. For the balancing design CFM values for diffuser 1 of 200 CFM and diffuser 2 of 100 CFM, fill out the proportional balance for the first two diffusers below if the measured CFM is found to be 175 for diffuser 1 and 125 CFM for diffuser 2.

Line 1A: _________________ (Diffuser 1 Measured CFM) ÷ (Diffuser 1 Design CFM)

Line 1B: _________________ (Diffuser 2 Measured CFM) ÷ (Diffuser 2 Design CFM)

Line 1C: _________________ (Line 1A) ÷ (Line 1B)

Line 1D: _________________ (Diffuser 1 Design CFM) x Line 1C

Line 1E: _________________ (Diffuser 2 Design CFM) x Line 1C

2. For the filled-out sheet above, explain how you would adjust diffuser 1.

3. Using the information from the filled out proportional balancing diffuser 1&2 fill out the proportional balance sheet below for Diffuser 3 for a balancing design CFM of 146 and a measured value of 138 CFM.

Diffuser 3

2A _________________ $\frac{\text{Diffuser 3 Measured CFM}}{\text{Diffuser 3 Design CFM}}$

2B _________________ $\text{Line 1C} \times 2$

2C _________________ $\frac{\text{Line 2A + Line 2B}}{3}$

2D _________________ $\text{Diffuser 3 Design CFM} \times \text{Line 2C}$

4. Would you adjust the damper for Diffuser 3? Why or why not?

5. What would the difference be between the diffuser 3 proportional balance sheet and a proportional balance sheet for diffuser 4?

6. Why will the airflow change in a proportionally balanced HVAC system if the fan changes to a lower speed?

between the entering and leaving airstreams, while energy recovery ventilators transfer sensible and latent heat between the entering and leaving airstreams. Energy recovery ventilators are recommended for use in warm, humid climates, such as the southeast United States.

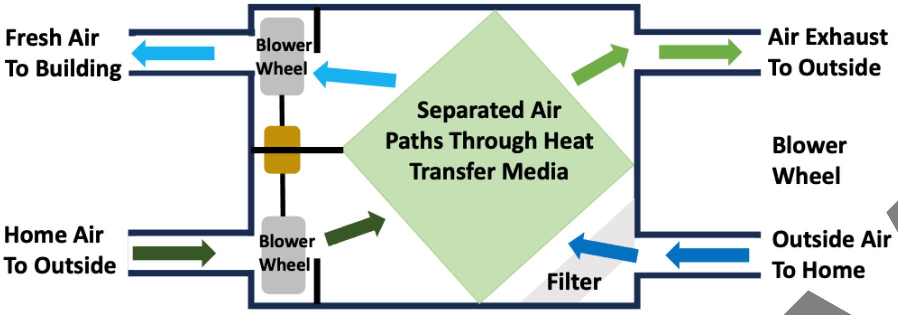


Figure 9-2: Cutaway illustration of a heat recovery ventilator.

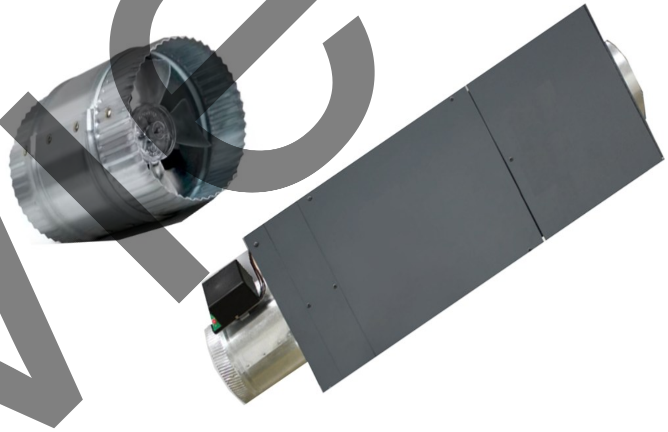


Figure 9-3: In line fan (left) fan with filter (right).

Dedicated outside air fans bring in a predetermined amount of airflow from the outside. The airstream entering the structure is usually filtered. Note: dedicated fans without filters bring the air into the return duct system, before the equipment's air filter, so the airstream is filtered before it ultimately passes through the system's heat exchangers. Figure 9-3 shows two types of dedicated outdoor air fans. The amount of air being brought into the structure must, as mentioned earlier, be verified to ensure that the design ventilation requirements are met. Generally, the equipment manufacturer will provide a guidelines and instruction for adjusting the airflow rate. However, like exhaust fans, the length and size of duct used can cause airflow rates to vary from the manufacturer's published table values. TAB technicians must verify that the actual airflow being delivered is correct.

Figure 9-4: Outside air dehumidifier.

In homes where high humidity is an issue, many contractors install a dehumidifier that is designed to bring in outside air at a set humidity. Figure 9-4 shows a typical outside air dehumidifier designed to be connected into the HVAC system's supply or return duct. Some of the air in the return duct is routed to the dehumidifier where it mixes with outside air. This mixed airstream then passes through the dehumidifier and is returned to the return duct, at a point that is closer to the air handling unit. Generally for TAB purposes it is a best practice to bring the air from the return duct and put the fan-powered mixed air back

At this point, the dehumidifier manufacturer's literature will need to be utilized to determine the amount of air delivered by the dehumidifier at a static pressure of 0.4 inWC. Let us assume that, based on the manufacturer's data, the dehumidifier delivers 215 CFM of air at 0.40 inWC.

Knowing that the dehumidifier's blower was initially set to bring in 60 CFM of outside air and, at a static pressure of 0.40 inWC, the dehumidifier is moving 215 CFM of mixed air, the amount of air being pulled from the return duct can be calculated as follows:

Amount of Air From Return Duct = Total Dehumidifier Mixed Aiflow – Dehumidifier's set Outside Air Airflow Rate
Amount of Air From Return Duct = 215 CFM – 60 CFM
Amount of Air From Return Duct = 155 CFM

Using the fan law formula the new HVAC System total CFM can be calculated:

$$\text{CFM}_{\text{TARGET VALUE}} = \text{CFM}_{\text{BEGINNING AS FOUND}} \div (\text{SP}_{\text{NEW VALUE}} \div \text{SP}_{\text{BEGINNING AS FOUND}})^{1/2}$$

Substituting the known values of:

- $\text{CFM}_{\text{BEGINNING AS FOUND}}$: 800 CFM
- $\text{SP}_{\text{NEW VALUE}}$: 0.40 inWC
- $\text{SP}_{\text{BEGINNING AS FOUND}}$: 0.35 inWC

We get the following:

$$\text{CFM}_{\text{TARGET VALUE}} = \text{CFM}_{\text{BEGINNING AS FOUND}} \div (\text{SP}_{\text{NEW VALUE}} \div \text{SP}_{\text{BEGINNING AS FOUND}})^{1/2}$$
$$\text{CFM}_{\text{TARGET VALUE}} = 800\ \text{CFM} \div (0.40 \div 0.35)^{1/2}$$
$$\text{CFM}_{\text{TARGET VALUE}} = 800\ \text{CFM} \div (1.143)^{1/2}$$
$$\text{CFM}_{\text{TARGET VALUE}} = 800\ \text{CFM} \div 1.069$$
$$\text{CFM}_{\text{TARGET VALUE}} = 748\ \text{CFM}$$

Important Note: The TAB technician needs to verify that this calculated value meets the air handler's minimum CFM requirement. If it is met, the system can be balanced.

This airflow value is the airflow rate passing across the HVAC equipment's heat exchanger. The total system airflow can now be calculated as the sum of the airflow rates attributed to the HVAC equipment and the dehumidifier as follows:

Total System Airflow Rate = Airflow Rate for HVAC Equipment + Airflow Rate for the Dehumidifier
Total System Airflow Rate = 748 CFM + 215 CFM
Total System Airflow Rate = 963 CFM

For most multi-zone systems, the first step for balancing is to make sure all of the zones are open and, if the system is equipped with a bypass duct, the damper in this duct must be closed, Figure 10-1. A bypass duct connects the supply duct to the return duct in a multi-zone system. Bypass dampers, Figure 10-2, are set to open when one or more zone damper closes and static pressure builds up in the supply duct. The opening of the bypass damper allows the excess air in the supply duct to flow directly back into the return duct, Figure 10-3. Bypass dampers are designed to be closed when all of the supply ducts are open. Balancing bypass ducts will be covered later in this chapter.

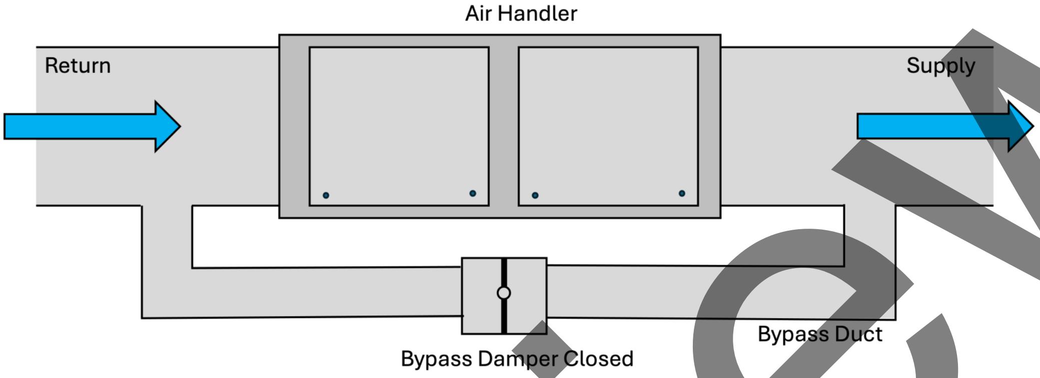


Figure 10-1: *A bypass duct connects the supply duct to the return duct. This bypass damper is in the closed position.*

Figure 10-2: *Manual hand damper (left) automated bypass damper (right).*

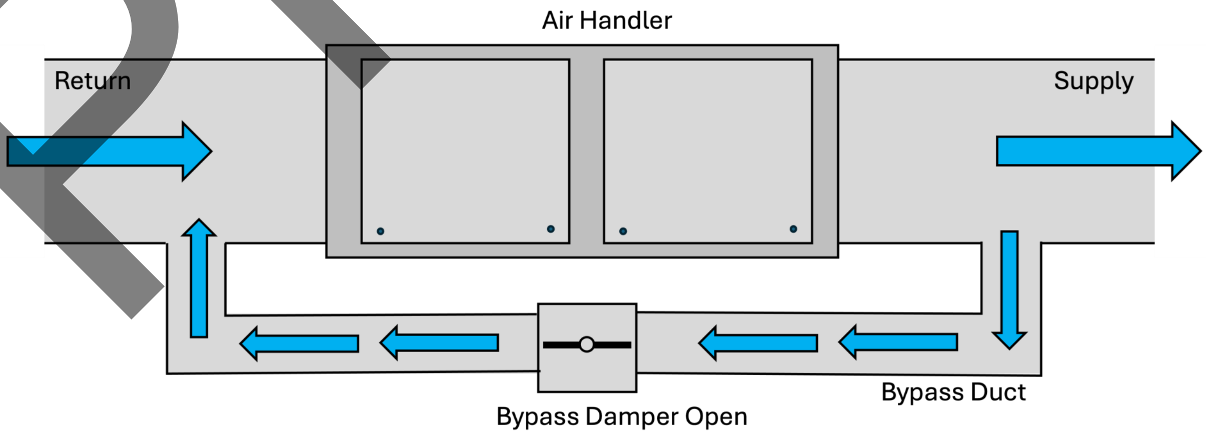


Figure 10-3: *When the bypass damper opens, air from the supply duct flows to the return duct.*

down the ESP and SP values for each step.

9. Place the system in the operational mode that requires the most CFM, and set the thermostat to keep the system operating during the balancing process.

Allow the system to stabilize by operating the equipment for at least 10 minutes and then continue the process as follows:

10. Shut down all of the zones except for the one with the least designed airflow. Note: Manual Zr provides guidance on how much bypass airflow is allowable. The smallest zone should be designed accordingly.
11. Open the bypass damper(s).
12. Re-measure the SP on the supply trunk.
13. Adjust the manual/hand damper on the bypass duct until the SP on the main trunk is back to the original value it had in the 1st test.
14. Lock down the manual damper on the bypass duct and make sure the SP is still equal to the original value. If the SP value is not equal to the original value, readjust the manual damper until it is, lock down the damper's position and mark it as well.

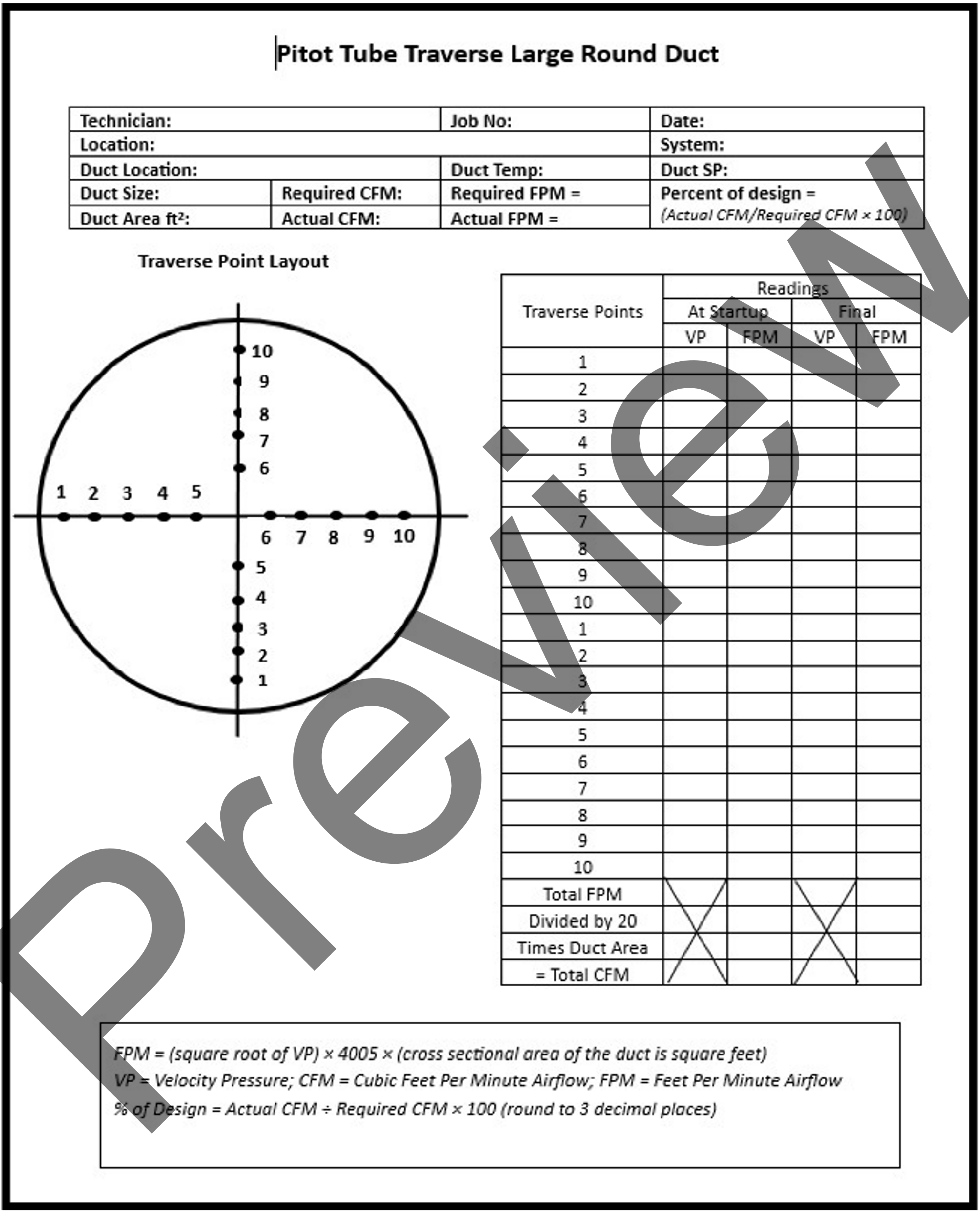
Before leaving, the zone thermostats/controls should be turned back to their normal operational setting, outside air dampers should be opened back up, and the ESP should be checked in both heating and cooling modes with only the smallest zone open to make sure the balanced airflow CFM falls within the OEM's minimum airflow requirements for the unit, and meets Manual Zr requirements.

As a final note, automatic bypass dampers that operate based internal programming based on Fan Law 2 [$SP_{final} = SP_{beginning} \div (CFM_{beginning} \div CFM_{final})^2$] are now available. To balance them, follow the manufacturer's directions. They usually require the SP to be set on the damper control with all of the dampers open. The damper then moves to maintain that SP as the zones open and close.

Conclusion

Improperly balanced multi-zone systems can cause a number of system problems that include premature equipment failure due to a lack of airflow under some of the operating conditions. Balancing multi-zone systems takes a little longer than balancing basic, single-zone HVAC systems, but, for comfort and energy savings, multi-zone systems are the best option. Properly balancing a multi-zone system makes sure the comfort and energy savings are realized by the homeowner. The Capacity Deployment zone system is easier to balance because there is no need for excess air options or balancing them.

Every diffuser and return grille in any multi-zone system needs to be balanced to the designed airflow. Two-speed multi-zone systems must be balanced while the system's blower operates at high speed. As noted in Chapter 8 of this manual, the airflow may not remain proportional when the unit's blower is operating at a lower speed. The exception to that is if the speed is related to maintaining a supply duct static pressure using fan law 2 related calculations and settings. If the duct's static pressure remains the same, the airflow through the open diffusers should also be the same. When you master balancing residential multi-zone systems you will become one of the best residential test and balance technicians in the world.



Pitot Tube Traverse Large Round Duct

Technician:		**Job No:**	**Date:**
Location:			**System:**
Duct Location:		**Duct Temp:**	**Duct SP:**
Duct Size:	**Required CFM:**	**Required FPM =**	**Percent of design =** *(Actual CFM/Required CFM × 100)*
Duct Area ft²:	**Actual CFM:**	**Actual FPM =**	

Traverse Point Layout

Traverse Points	Readings			
	At Startup		Final	
	VP	FPM	VP	FPM
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
Total FPM				
Divided by 20				
Times Duct Area				
= Total CFM				

FPM = (square root of VP) × 4005 × (cross sectional area of the duct is square feet)
VP = Velocity Pressure; CFM = Cubic Feet Per Minute Airflow; FPM = Feet Per Minute Airflow
% of Design = Actual CFM ÷ Required CFM × 100 (round to 3 decimal places)

Pitot tube measured lengths on next page

Diffusers 1 and 2

1A ____________________ $\dfrac{\textit{Diffuser 1 Measured CFM}}{\textit{Diffuser 1 Design CFM}}$

1B ____________________ $\dfrac{\textit{Diffuser 2 Measured CFM}}{\textit{Diffuser 2 Design CFM}}$

1C ____________________ $\dfrac{\textit{Line 1A + Line 1 B}}{2}$

1D ____________________ *Diffuser 1 Design CFM × Line 1C*

1E ____________________ *Diffuser 2 Design CFM × Line 1C*

Diffuser 3

2A ____________________ $\dfrac{\textit{Diffuser 3 Measured CFM}}{\textit{Diffuser 3 Design CFM}}$

2B ____________________ *Line 1C × 2*

2C ____________________ $\dfrac{\textit{Line 2A + Line 2B}}{3}$

2D ____________________ *Diffuser 3 Design CFM × Line 2C*

Diffuser 4

3A ____________________ $\dfrac{\textit{Diffuser 4 Measured CFM}}{\textit{Diffuser 4 Design CFM}}$

3B ____________________ *Line 2C × 3*

3C ____________________ $\dfrac{\textit{Line 3A + Line 3B}}{4}$

3D ____________________ *Diffuser 4 Design CFM × Line 3C*

Diffuser 5

4A ____________________ $\dfrac{\textit{Diffuser 5 Measured CFM}}{\textit{Diffuser 5 Design CFM}}$

4B ____________________ *Line 2C × 4*

4C ____________________ $\dfrac{\textit{Line 4A + Line 4B}}{5}$

4D ____________________ *Diffuser 5 Design CFM × Line 4C*

Air Constant Equations

BTUH (sensible) = 1.08 × CFM × [Dry-Bulb Difference ($T_1 - T_2$)]

BTUH (latent) = 0.68 × CFM × [Grains Difference ($GR_1 - Gr_2$)]

Fan Law 1

$(CFM_{beginning} \div CFM_{final}) = (RPM_{beginning} \div RPM_{final})$

$(CFM_{beginning} \div RPM_{beginning}) = (CFM_{final} \div RPM_{final})$

$RPM_{final} = RPM_{beginning} \div (CFM_{beginning} \div CFM_{final})$

Fan Law 2

$(CFM_{beginning} \div CFM_{final})^2 = (SP_{beginning} \div SP_{final})$

$SP_{final} = SP_{beginning} \div (CFM_{beginning} \div CFM_{final})^2$

Fan Law 3

$(CFM_{beginning} \div CFM_{final})^3 = (BHP_{beginning} \div BHP_{final})$

Formula for calculating residential single-phase BHP: BHP = (A × V × Eff. × PF) ÷ 745.7

Where: A = amperage; V = volts; Eff. = nameplate Motor efficiency; name plate motor power factor; and 745.7 is a constant conversion factor.

Equation Abbreviations

A = Measured Amperage

EFF = Efficiency

BHP = Break Horsepower

BTUH = British Thermal Units Per Hour

CFM = Cubic Feet Per Minute

GR = Grains of Water

PF = Power Factor

RPM = Revolutions Per Minute

SP = Static Pressure

T = Temperature

V = Volts

air conditioner	Equipment with the ability to absorb heat in occupied spaces and reject it outdoors.
air handler	Equipment with a heating element and/or cooling coil and other components contained in a cabinet or casing.
amps	A unit of electrical current (ampere; A).
balancing / air balancing	Adjusting an air conditioning system so that the right amount of air is delivered to the right places in the home to achieve the designed heating or cooling effect.
blower	A motorized air-moving device that produces a pressure difference in air to move it.
brake horsepower	The actual horsepower a motor is operating at.
BTU	British Thermal Unit, the measurement of heating and air conditioning capacity. A BTU is defined as the amount of heat that must be added to one pound of water to raise its temperature one degree Fahrenheit.
BTUh	British Thermal Units added or removed per hour.
coil	A heat transfer surface found in a heating or cooling system. Usually constructed of piping or tubing, with plates or fins.
condenser	The portion of an air-conditioning system that is responsible for rejecting heat from the system.
contractor	The person or entity responsible for performing the work and identified as such in an owner-contractor agreement.
damper	A device for adjusting the amount of air flowing through the duct.
diffuser	An air terminal device that has an adjustable damper for minor air pattern and volume control.
draft	The movement of air caused by excessive velocity from a supply diffuser or to a return grille.
duct	Conduits used to carry air.
duct airflow balance	Work associated with measuring and adjusting airflow rates at various points in an air distribution system to provide correct airflow delivery to rooms or spaces as prescribed during the design process.
ESP	External Static Pressure, measured in Inches of Water Column, inWC.
evaporator (heat exchanger)	The component of a refrigeration system in which the refrigerant absorbs heat and vaporizes.
fan	A device that produces a pressure difference in air to move it.

RESIDENTIAL AIRSIDE TESTING AND BALANCING

DONALD PRATHER AND EUGENE SILBERSTEIN, M.S., CMHE, BEAP

Residential Airside Testing and Balancing provides a practical, step-by-step guide to verifying and optimizing airflow performance in residential HVAC systems using modern tools and proven field methods. Beginning with airflow fundamentals, heat transfer, and system operation, the book builds a strong foundation before introducing the specialized instruments and measurement techniques required for accurate testing and diagnostics. Readers learn the science behind air movement including fan laws, air properties, and gas relationships, alongside hands-on procedures such as duct traversing, airflow measurement, and interpretation of design values derived from load calculations.

The text emphasizes real-world application, helping technicians understand not only how to measure airflow, but why proper testing and balancing is essential for comfort, efficiency, equipment longevity, and customer satisfaction. By combining theory with field-tested practices, this manual prepares technicians to confidently evaluate residential air distribution systems and achieve repeatable, verifiable results.

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