Design Advisory #2: CAS-DA2-2003

Excellent Duct Systems Require Design, Not Guesswork!

Duct System Design Guide: Chapter 2

HVAC system performance is only as good as it's weakest link. Unfortunately, duct systems are still a weak link in obtaining optimum HVAC performance.

While the majority of HVAC system equipment and controls are getting more complex and high-tech, many duct designs continue to spawn from a few spins of a duct calculator wheel.

Granted, these tools have gotten more sophisticated with the addition of dynamic fitting loss calculations. But these individual values, although calculated fairly accurately, cannot, without experienced use and additional hand calculations, result in overall balanced system performance.

Many designers rely, unfortunately, on the flexibility of variable air volume (VAV) control or other terminal boxes and associated controls to make their designs work. While this approach has proven to work quite well for several of the more popular low-pressure, low-velocity systems with simplified ductwork layouts, it has proven quite ineffective for medium- and high-pressure high-velocity systems, especially those systems having more complex ductwork layouts. Poor system performance often relates to system imbalances — air following the path of least resistance as opposed to flowing where it's needed.

Proper system air volume distribution requires proper dampening of non-design leg branches based on demand load conditions. Terminal box dampers that must close to balance the system may not have enough pressure to provide air to downstream ductwork. System imbalances at terminal boxes may result in inadequate system pressure to operate it. Proper system design is a balancing act!

Duct calculators are good for quick performance estimates based on budget duct sizing. But a proper duct design is one that takes into consideration the performance characteristics of all the components in the system.

Once heating and cooling loads are properly established, the next step should be the duct design. Air handling equipment, terminal boxes and controls cannot be adequately selected until the ductwork interconnecting them is designed correctly. This can only be accurately done using duct system design software. The key focus should be system design, not component sizing. McGill AirFlow's Duct System Design Service uses its proprietary UNI-DUCT® design software to accomplish this. This design service is available by contacting the McGill AirFlow location nearest you. Click here to access the Location Guide.

Take a good look at your next HVAC system project. What part of that system takes up the most area and most time to install? What component of an HVAC system is the most essential, next to the fan, in getting air where it's supposed to go? Why not spend some time using a proven duct system design program and be assured that air gets where it's supposed to go?

This ***Duct System Design Guide*** Chapter will give you greater appreciation for proper duct system design. Good duct system design is more complex than it first appears!

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CHAPTER 2: Designing Supply Duct Systems

2.1 Determination of Air Volume Requirements

Modern buildings are constructed to minimize unintentional exchanges of inside and outside air. The environment inside is expected to be fresh and the temperature and humidity to be nearly constant throughout the year. To accomplish this, it is necessary to introduce conditioned and circulated air to every occupied portion of the building.

The key parameter which quantifies the required amount of air movement for a given occupancy situation is the air volume flow rate. In the English system, the air volume flow rate is expressed in units of cubic feet per minute (*cfm*).

The first step in designing a duct system is determining the required air volume flow rate, and a wealth of information is currently available on this topic. Properly calculating air volume flow rate requirements is rather complex and involves a number of factors. A few of those factors are geographic location (expected temperature and sunlight conditions), building orientation, glass area (radiant heat gain), thermal conductivity of exterior building surfaces, interior heat sources, and desired or required percent of fresh air intake.

Computation of the air volume flow rate is not within the scope of this design notebook. It will be assumed throughout that these values are known and that the air location and air volume flow rates necessary at each terminal device have been established. **Appendix A.9.2** provides an excellent source for information concerning heating and cooling load calculations and the determination of air volume flow rate requirements, and there are many computer programs available to speed the calculations. **Appendix A.9.3** discusses room air distribution, including selection of terminal devices as well as proper location for both heating and cooling.

2.2 Location of Duct Runs

The second step in designing a duct system is laying it out. To do that it is necessary to determine the location of the air handling unit and the terminal devices. **Appendix A.9.3** provides guidelines for locating air diffusers and return grilles and information regarding coverage and air circulation. The air handler will ideally be located in an area which is remote from noise-critical spaces or which is acoustically insulated from them.

Next, taking into account all known architectural, mechanical, and electrical obstructions, a single-line sketch is made, connecting the air handling unit to each terminal or volume control box. Make the duct runs as straight as possible. Each turn will require additional pressure. Always try to maintain the largest volume flow in the straight-through direction, relegating the lesser flows to the branches. This is also a good time to locate and indicate fire and smoke dampers, access doors, and any other duct accessories that may be required.

Here are a few suggestions that may be helpful:

1. Pay particular attention to the available space for installation of duct. Generally, it will be installed between the roof or upper floor support girders and the suspended ceiling. This space is usually not large to start with, and it will be even more restrictive if other trades have already installed their equipment. If flat oval or rectangular duct is used, remember to allow two to four extra inches for the required reinforcement. Round duct under normal positive HVAC pressures (0 to 10 *inches wg*) does not require reinforcement, but you will need to allow for any flanged connections for large duct.

2. Remember that duct will be largest near the air handling unit. If there are space problems, this is the area where they will most likely occur. Try to locate ducts between the girders, if possible, instead of locating them below the girders as larger ducts may be used. Consider multiple runs of small duct in lieu of a single run of large duct. Often this approach minimizes reinforcement and improves acoustics.

3. Duct systems carry noise as well as air. The largest noise source will be the fan. Some noise is also generated by the moving air stream and will propagate down the duct run from the fan. Fan noise will normally be most pronounced near the air handler, and will dissipate as the distance from the fan increases. Remember that fan noise will be propagated through both supply and return ducts. High velocity, high pressure loss fittings, and/or components located in the airstream (tie rods, extractors, etc.) will introduce duct-generated noise. It is good practice to anticipate where the duct noise will be greatest and locate those sections over public areas such as hallways, lobbies, cafeterias, or restrooms. For more information see **Chapters 7, 8** and **9**.

4. Flexible duct is often used for final connections to or from volume control boxes and terminal units. This duct has a very high pressure loss per unit length, and is virtually transparent to noise. It should be avoided whenever possible, and in no case should it be installed in sections longer than 5 feet. Bends, turns, and sags should be avoided, as they will substantially increase the pressure loss and may choke off the air supply to the device they are serving.

2.3 Selection of a Design Method

In the first two design steps, the air volume flow rate for each terminal and the location and routing of the duct were identified. Now the designer must determine the size of the duct, and select the appropriate fittings. These decisions will have a pronounced impact on the cost and operation (efficiency) of the system.

Selection of a design method is not truly a step in the design process, but it is listed here because it provides an opportunity to discuss several options that are available to the designer. Often the design method employed will be the one with which the designer is most familiar or most comfortable. Too often, when asked why they design systems in a certain way, designers respond, "because we've always done it that way."

Before we discuss the mechanics of an actual duct design, let's look at several design methods to compare their features. Terms used below are explained in subsequent selections.

2.3.1 Equal Friction Design

- Probably the most popular design method.
- Quick and easy to use; many nomographs and calculators are available.
- A friction loss per unit length is selected for all duct; this value is usually in the range of 0.05 to 0.2 *inches wg per 100 feet* of duct length.
- All duct is sized using the known air volume flow rates and the selected friction loss.
- Fitting losses can be calculated, but more often are estimated.
- The critical path (maximum pressure requirement) is often chosen by inspection unless a computer program is used to do the calculation.

- System pressure is usually calculated by multiplying the critical path length, plus an allowance for fitting losses, by the design friction loss per unit length.

- System analysis and/or optimization is generally not performed unless a computer program is used.

- Balancing is usually required and is accomplished with dampers or orifice plates.

2.3.2 Constant Velocity Design

- Quick and easy to use; many nomographs and calculators are available.

- A velocity is selected, which will be maintained throughout the system.

- All duct is sized using the known air volume flow rates and the selected velocity.

- Fitting losses can be calculated but more often are estimated (equivalent lengths).

- The critical path (maximum pressure requirement) is chosen by inspection.

- System pressure is calculated by adding the individual pressure losses for each section of the critical path (determined by chart, calculator, or nomograph), plus an allowance for fitting losses.

- System analysis and/or optimization is generally not performed.

- Balancing is usually required and is accomplished with dampers or orifice plates.

2.3.3 Velocity Reduction Design

- Similar to constant velocity design, except that instead of selecting a single, constant velocity for all duct sections, the velocity is systematically reduced in each downstream section.

- This method has questionable application, since there is no valid reason for continually reducing downstream velocities; in fact, it is counterproductive in certain situations.

2.3.4 Static Regain Design

- Probably the most difficult and time-consuming design method, but generally produces the most efficient system (lowest operating pressure, well balanced).

- An initial duct size (or velocity) is selected.

- All duct is sized so that the pressure loss in any duct section is equal to the regain of pressure regain caused by reducing the velocity from the upstream section to the downstream section.

- Fitting losses must be calculated (they are used in sizing the duct).

- The critical path is chosen by inspection or, more often, by computer.

- System pressure is determined by adding the individual (calculated) pressure losses for all duct and fitting elements of the critical path.

- Static regain designs tend to be more self-balancing than other methods, but if balancing is required, it is usually accomplished with dampers or orifice plates.

2.3.5 Total Pressure Design

- Initially, the system is sized using one of the methods described above.
- The critical path(s) is determined.
- All non-critical legs, by definition, will have excess total pressure; if they are not redesigned or dampered, there will be a system imbalance.
- Duct sizes in the non-critical legs are reduced in size (velocities and pressures are increased) until there is no longer any excess total pressure in the section.
- Alternatively, or in addition, certain fitting types may be altered to reduce the excess total pressure (this generally involves the substitution of a less efficient and less costly fitting).
- Will always produce the lowest first-cost system for a given operating total pressure, and will self-balance the system and virtually eliminate dampering.
- The large number of iterations required by this method is best accomplished by use of a computer program.

2.3.6 Which Design Method?

For those with access to a computer and the necessary software, a total pressure design of a system initially designed by static regain will produce the most efficient, cost-effective system possible. Although the equal friction design is simple and straightforward, it generally will not result in the most efficient or cost-effective system. If systems are small or if the designer does not have access to a computer program, equal friction design with a low friction loss *per 100 feet* (0.05 *inches wg per100 feet* to 0.10 *inches wg per 100 feet*) will be most cost effective from a design time aspect.

In the following sections, both the equal friction and static regain design methods are explored. The total pressure method of design will be discussed in **Chapter 3**.

2.4 Equal Friction Design

2.4.1 Introduction

As mentioned earlier, equal friction is the most commonly used design method. In the following sections the duct is sized, the system pressure is determined, and the excess pressure is calculated. Duct sizing is the key part of this method of design since the other two steps are common to the other methods.

2.4.2 Duct Sizing

Equal friction design is based on the concept that the duct friction loss per unit length at any location in the system should always be the same. The first element of the design, therefore, is to select a friction loss per unit length for the system. Any value can be selected, but many designers prefer values between 0.05 to 0.20 *inches wg per 100 feet* of duct length. One method of choosing a friction loss per unit length is to determine an initial velocity and size the first section for this velocity. Then determine this section's friction loss rate from a friction loss chart or nomograph (duct calculator). The friction loss per unit length is then applied to other sections. Some suggested initial velocity values are given for round systems in **Table 2.1**. Alternatively, a friction loss per unit length is predetermined based on what the designer feels comfortable with, and then the friction loss chart or duct calculator is used to determine sizes.

Once the friction loss per unit length has been selected, one of the two parameters necessary to select duct sizes using the friction loss chart is fixed. (For a review of the use of friction loss charts, see **Section 1.4**.) Since it is assumed that the air volume flow rates for each terminal location are known, the air volume requirements can be determined for all duct sections in the system by applying the mass flow principles from **Section 1.2.2**.

Table 2.1
Suggested Initial Velocities in Round Systems

System Volume Flow Rate (*cfm*)	**Suggested Initial Velocity** (*fpm*)
0 to 15,000	1,500
15,000 to 30,000	2,500
30,000 to 70,000	3,000
70,000 to 100,000	3,500
above 100,000	4,000

Duct diameters are selected from the duct friction loss chart at the intersection of the appropriate unit friction loss per unit length line and the air volume flow rate line for the duct section in question.

Sample Problem 2-1

Use the equal friction design method to size ***Sample System 1*** *(**Figure 2.1**), with a friction loss of 0.20 inches wg per 100 feet. What is the diameter of each section? What would the diameters be if 0.10 inches wg per 100 feet had been selected as the design friction loss?*

Answer: First, determine the air volume flow rates for each duct in the system. These are shown in **Table 2.2**. Note that the system has been divided into numbered sections (in hexagons), which begin with a divided-flow fitting and end just before the next divided-flow fitting. The air volume flow rates for each section can be calculated by summing all downstream terminal requirements.

Table 2.2
Air Volume Flow Rates for Sample System 1

Section 1	20,600 *cfm*
Section 2, 3, 9	1,600 *cfm*
Section 4	17,400 *cfm*
Section 5	2,400 *cfm*
Section 6	15,000 *cfm*
Section 7	3,200 *cfm*
Section 8	11,800 *cfm*
Section 10	10,200 *cfm*
Section 11	3,600 *cfm*
Section 12	6,600 *cfm*

Next, calculate the duct sizes. For Section 1, refer to the duct friction loss chart in **Figure 2.2** and locate the intersection of the 0.20 *inches wg per 100 feet* friction loss line (horizontal axis), and the 20,600 *cfm* volume flow rate line (vertical axis). The 20,600 *cfm* line will have to be estimated. The intersection of these lines is approximately at the 38-*inch* duct diameter line. The friction loss chart also indicates that the air velocity in this section will be approximately 2,600 *fpm*.

Therefore, an air volume flow rate of 20,600 *cfm* will flow through a 38-*inch* diameter duct at a velocity of approximately 2,600 *fpm* and it will have a friction loss of about 0.20 *inches wg per 100 feet* of duct length.

Similarly, for Section 2 the intersection of the 0.20 *inches wg per 100 feet* friction loss line and the 1,600 *cfm* line is almost equidistant between the 14-*inch* and the 15-*inch* duct diameter lines. In this case, a 14.5-*inch* diameter duct would be the best choice. Some manufacturers will manufacture half sizes, but for simplicity this design will keep sizes in whole inch increments. Using a 14-*inch* duct to carry 1,600 *cfm* would actually require 0.23 *inches wg per 100 feet*, whereas a 15-*inch* duct would only require 0.17 *inches wg per 100 feet*. The larger duct size will be used. All remaining sections can be sized in a similar manner.

If the design were to be based on a friction loss of 0.10 *inches wg per 100 feet*, the procedure would be identical, except that the duct size would be selected at the intersection of the friction loss line representing 0.10 *inches wg per 100 feet* and the required air volume flow rate. In this case, Section 1 (20,600 *cfm*) would require a 44-*inch* diameter duct.

Figure 2.2 shows how these values are obtained from the friction loss chart. Dashed lines are shown on the chart for the 0.10 *inches wg per 100 feet* friction loss and the 0.20 *inches wg per 100 feet* friction loss. **Table 2.3** shows the duct sizes as determined by each design method.

Table 2.3
Equal Friction Duct Sizing for Sample System 1

Section	**Volume Flow Rate** (*cfm*)	**Duct Diameter** (*inches*)*	
		0.20 *inches wg per 100 feet*	0.10 *inches wg per 100 feet*
1	20,600	38	44
2,3,9	1,600	15	17
4	17,400	36	42
5	2,400	17	20
6	15,000	34	40
7	3,200	19	22
8	11,800	31	36
10	10,200	29	34
11	3,600	20	23
12	6,600	25	29

* Rounded to the nearest whole diameter

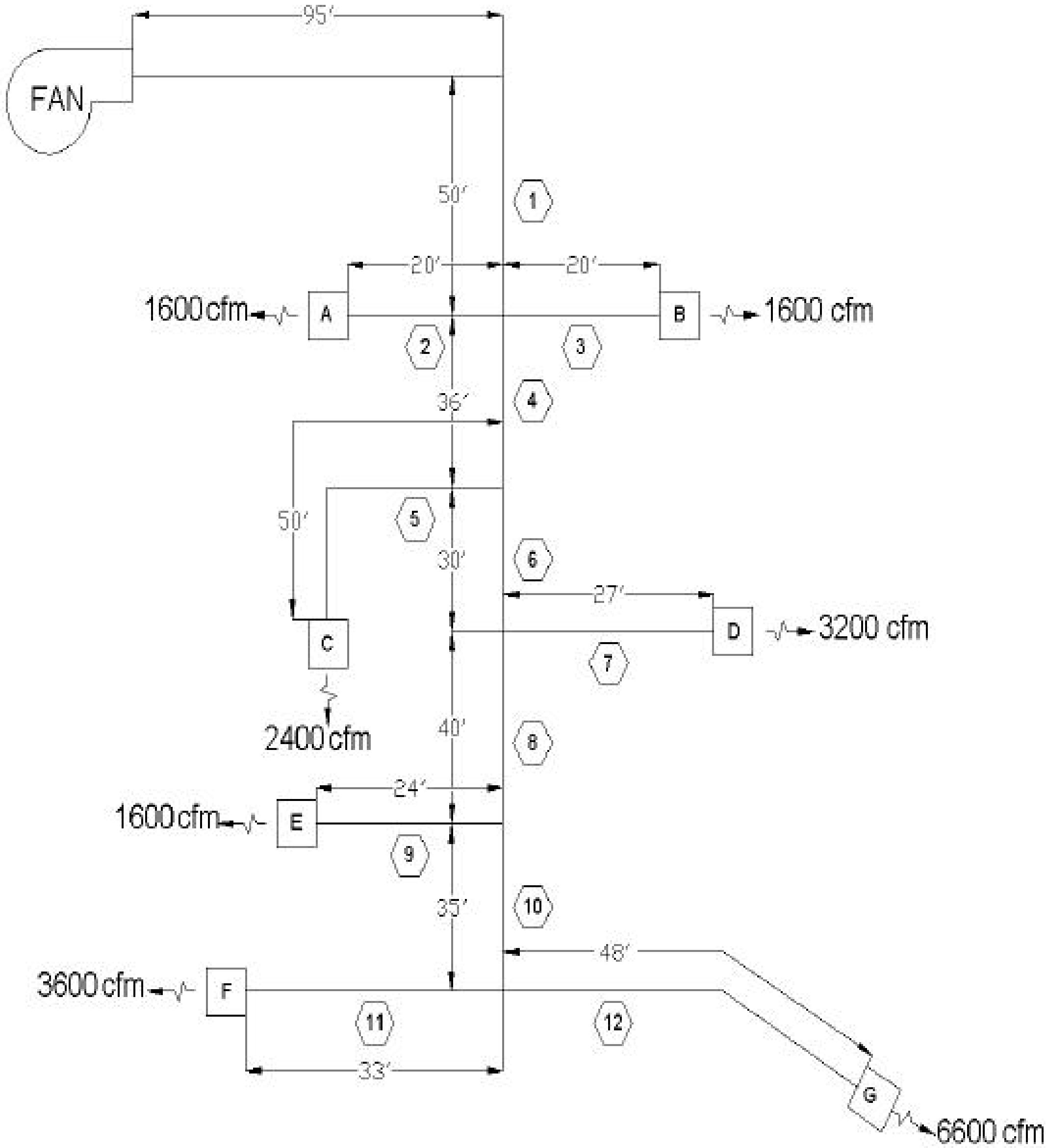

Figure 2.1
Sample System 1

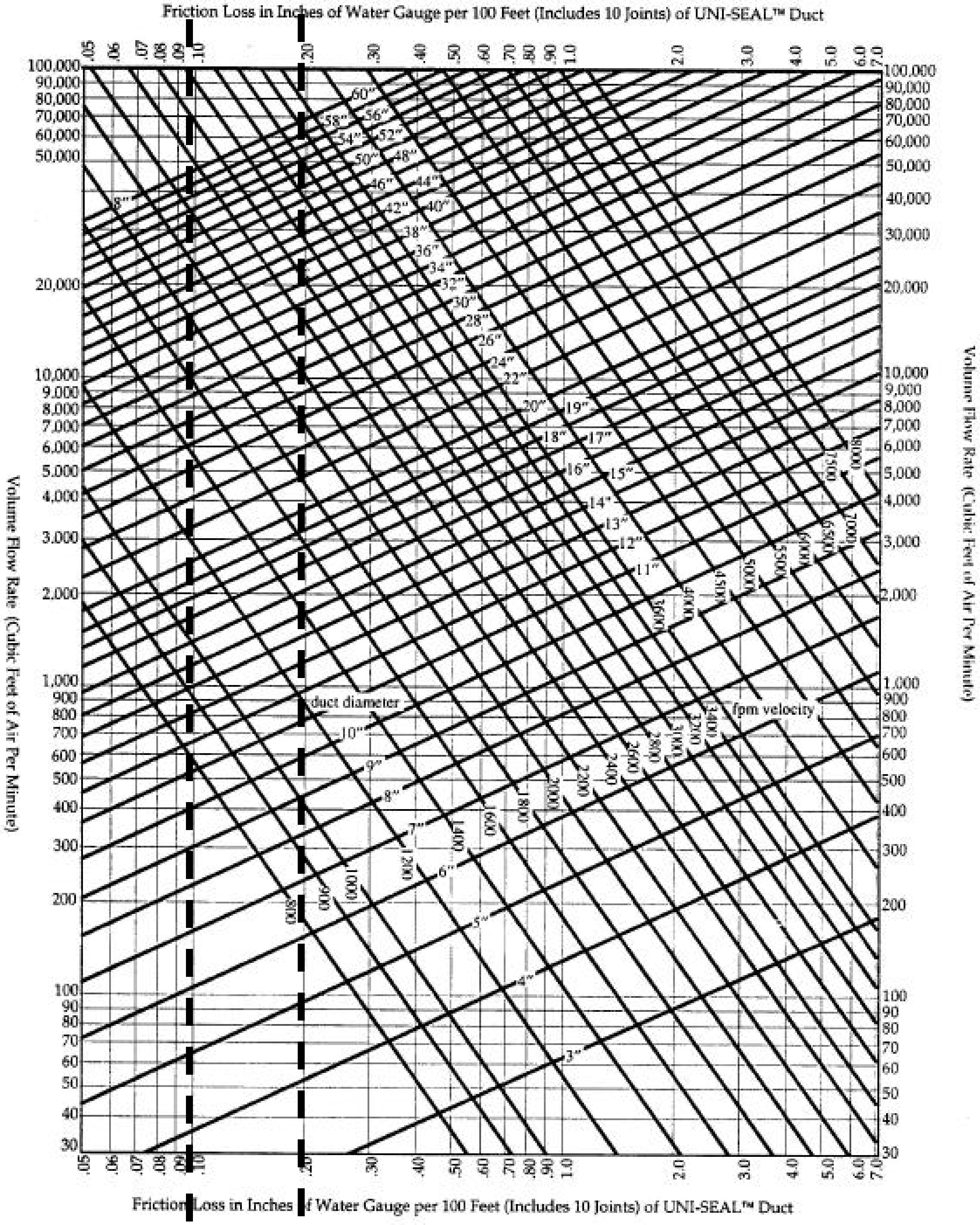

Figure 2.2
Friction Loss of Duct

From **Sample Problem 2-1**, a very important implication of the friction loss per unit length selection is apparent. When a higher (0.20 *inches wg per 100 feet*) friction loss per unit length is selected, the duct sizes in all sections are smaller than for the lower (0.10 *inches wg per 100 feet*) friction loss per unit length. Smaller duct sizes will result in a less expensive system to purchase and install. However, in subsequent sections, the selection of a higher friction loss per unit length will have negative consequences in terms of system pressure and operating costs and may result in substantially increased total costs.

2.4.3 Determination of System Pressure

Once the system air volume flow rate requirements are known, the duct can be sized. The designer knows that if the required volume of air flows through the ducts, it will do so at the unit pressure losses and velocities indicated by the friction loss chart. The next step is to make certain that the required volume flow rate of air will, in fact, flow through the ducts.

To supply the required volume flow rate in each duct section, the air handler must be capable of delivering the total system volume flow (20,600 *cfm* for **Sample System 1**). The air handler must supply this volume of air at a total pressure sufficient to overcome the resistance (frictional and dynamic) of the path having the highest pressure loss. This path is called the critical path or design leg, and the sum of all the total pressure losses from the air handler to the terminal of the critical path is called the system (total) pressure.

The next step in the design process is to identify the critical path and to determine its pressure loss.

System Pressure by Inspection

The simplest means of identifying the critical path and determining the system pressure is by inspection. Ideally, this should be done only by an experienced designer who has analyzed numerous systems and has a good feeling for the pressure losses associated with various system components.

It is often the case that the critical path is simply the longest path in a system. If frictional (duct) losses accounted for all the pressure losses, this would usually be the case; however, the dynamic (fitting) losses often are substantial and may cause one of the shorter paths to have a higher pressure loss.

In an equal friction design, all ducts should have the same pressure loss per unit length. Therefore, once the critical path is identified, the duct losses can be calculated by simply multiplying the design pressure loss (*inches wg per 100 feet*) by the total length of the critical path divided by 100 *feet*.

The critical path fitting losses should be calculated and added to the duct loss, although often only a very rough estimation is made. An acceptable compromise may be to ignore straight-through losses, which normally will be very small, but calculate and include the losses of any branches or abrupt-turn fittings that are in the critical path.

Sample Problem 2-2 will illustrate the inspection method for calculating the system pressure loss of **Sample System 1** (**Figure 2.1**).

Sample Problem 2-2

Determine the critical path by inspection and estimate the system total pressure loss for ***Sample System 1*** *(**Figure 2.1**), based on the 0.20 inches wg per 100 feet design. The fittings in Sections 1 and 5 are five-gore 90E elbows; all branch fittings are 90E conical tees; a bullhead tee with turning vanes is used at the Sections 10-11-12 junction; and there is a 45E, 3-gore elbow in Section 12.*

Answer: Because of the high volume flow rate requirement (6,600 *cfm*) and the distance from the fan, it is reasonable to assume that the critical path will be the duct run to Terminal G. The path includes Sections 1, 4, 6, 8, 10 and 12. The total center line length of these sections is 334 feet (95 + 50 + 36 + 30 + 40 + 35 + 48).

If the equal friction design is based on a friction loss of 0.20 *inches wg per 100 feet*, the total duct loss would be approximately:

334 feet H 0.20 *inches wg per 100 feet* = <u>0.67 *inches wg*</u>

Since bullhead tee losses are a function of volume flow rate and area ratios, it is necessary to know the volume flow rate and size of duct in Section 10 (common) and Section 12 (branch).

Q_b = 6,600 *cfm*	A_b = 3.41 *sq ft*	VP_b = 0.23 *inches wg*
Q_c = 10,200 *cfm*	A_c = 4.59 *sq ft*	VP_c = 0.31 *inches wg*
Q_b/Q_c = 0.65	A_b/A_c = 0.74	

Referring to **ASHRAE Fitting SD5-19**, the loss coefficient is approximately 0.67.

Substituting in **Equation 1.16b**, the total pressure loss is:

DTP_{c-b} = 0.23 x 0.67 = <u>0.15 *inches wg*</u>

In this case, the bullhead tee pressure loss is 22 percent of the duct loss and is therefore significant. It would also probably be wise to calculate the loss of the large elbow in Section 1. This is easily done by referring to **ASHRAE Fitting CD3-9**. At 2,616 *fpm*, a 38-*inch* diameter elbow will have a static pressure loss of approximately 0.05 *inches wg*.

($DTP = C$ x VP = 0.12 x 0.43 = 0.05 *inches wg*)

The loss for the 45E, 3-gore elbow in Section 12 is determined using **ASHRAE Fitting CD3-14** whereby DTP = 0.08 x 0.23 = 0.02 *inches wg*.

The total pressure loss of a path must also include the velocity pressure of the terminal section as well since this pressure will be lost when the air exits the section and must also be supplied by the fan. Therefore, a reasonable estimation of the pressure loss from the fan to Terminal G is:

0.67 + 0.15 + 0.05 + 0.02 + .23 = <u>1.12 *inches wg*</u>

If our critical path was selected correctly, this will be close to the system total pressure requirement, as determined by inspection.

System Pressure by Calculation

The best and most accurate way to determine system pressure losses is by calculation. There are many ways to approach these calculations, and many computer programs are now available to simplify the process. However, the longhand method is presented here since it will give the designer a better feel for the process.

In an equal friction design, the calculated pressure loss will differ from the estimated loss, as described in the preceding **Sample Problem 2-2**, in two ways: (1) the actual duct losses are used, and (2) all fitting losses are calculated and included.

Sample Problem 1 (**Figure 2.1**) will be used again for the analysis. **Table 2.4** is useful in collecting and analyzing the data necessary to properly calculate the system losses. It is grouped by section. Each section consists of duct and at least one branch fitting. The first section begins at the fan and ends just upstream of the first flow division; all subsequent sections begin with a flow division branch which is called the takeoff fitting, and ends just upstream of the subsequent downstream flow division. Elbows, constant area offsets, dampers, and other fittings that do not result in a change of area, shape, or air volume do not create new sections; their total pressure losses are included in the sections in which they occur.

For each section, the duct and fitting elements are listed, along with the following information:

1. Section velocity
2. Velocity pressure
3. Section velocity
4. Velocity pressure
5. Actual duct friction loss per 100 feet
6. Duct length
7. Actual duct loss
8. Fitting static pressure losses
9. Cumulative section static pressure loss
10. Fitting total pressure losses

Regarding 7, it is important to realize that, even in an equal friction design where the friction loss rate is supposedly constant, it will vary slightly from section to section. This is because the exact duct diameter required to move a given volume flow rate at the design pressure is seldom available.

Table 2.4
Sample System 1 - Equal Friction Design Data Sheet

SECT	ITEM REF[1]	ITEM	Q (cfm)	DIA. ($inches$)	AREA (ft^2)	V (fpm)	VP ($inches.\ wg$)	C	$\Delta P/100\ ft$ ($inches\ wg$)	L (ft)	ΔP DUCT ($inches\ wg$)	FITTING ΔSP ($inches\ wg$)	SECTION ΔSP ($inches\ wg$)	FITTING ΔTP ($inches\ wg$)	SECTION ΔTP ($inches\ wg$)
1	 CD3-9	Duct 90 Elbow	20,600	38	7.88	2,616	0.43	 0.12	0.20	145	0.29	 0.05	0.34	 0.05	0.34
4	 SD5-25	Duct Straight T/O	17,400	36	7.07	2,462	0.38	 0.14	0.19	36	0.07	 0.00	0.07	 0.05	0.12
6	 SD5-10	Duct Straight T/O	15,000	34	6.31	2,379	0.35	 0.13	0.19	30	0.06	 0.02	0.08	 0.05	0.11
8	 SD5-10	Duct Straight T/O	11,800	31	5.24	2,251	0.32	 0.13	0.19	40	0.08	 0.00	0.08	 0.04	0.12
10	 SD5-10	Duct Straight T/O	10,200	29	4.59	2,224	0.31	 0.13	0.20	35	0.07	 0.03	0.10	 0.04	0.11
2 and 3	 SD5-25	Duct Con Cross	1,600	15	1.23	1,304	0.11	 1.38	0.17	20	0.03	 (0.17)	(0.14)	 0.15	0.18
5	 SD5-10 CD3-9	Duct Con Tee 90 Elbow	2,400	17	1.58	1,523	0.14	 2.80 0.15	0.19	50	0.10	 0.17 0.02	 0.29	 0.40 0.02	0.52
7	 SD5-10	Duct Con Tee	3,200	19	1.97	1,625	0.16	 1.60	0.19	27	0.05	 0.08	0.13	 0.26	0.31
9	 SD5-10	Duct Con Tee	1,600	15	1.23	1,304	0.11	 3.22	0.17	24	0.04	 0.13	0.17	 0.34	0.38
11	 SD5-19	Duct Bullhead T/V	3,600	20	2.18	1,650	0.17	 1.03	0.18	33	0.06	 0.04	0.10	 0.18	0.24
12	 SD5-19 CD3-14	Duct Bullhead T/V 45 Elbow	6,600	25	3.41	1,936	0.23	 0.67 0.08	0.19	48	0.09	 0.28 0.02	 0.39	 0.16 0.02	0.27

1 – ITEM REF column lists the fitting designation as given by ASHRAE.

The velocities and pressure losses in the example above were read, as accurately as possible, from the friction loss chart in **Appendix A.4.1.1** or **Figure 2.2.** Nomographs, calculators or computers can also be used, and may provide slightly more accurate results. Note that the pressures attributable to the duct and all fittings within a section are totaled for each section. The **ASHRAE Duct Fitting Database** program was used to calculate the losses of fittings.

To determine the system pressure loss, it is again necessary to find the critical path. This time, however, there is no guesswork involved. Since actual pressure loss data is available for each section, the pressure requirement can be determined for all paths. The path with the highest pressure requirement (fan to terminal) is the critical path, and this pressure is the system pressure requirement. **Table 2.5** analyzes the terminals of **Sample System 1**. Remember that the velocity pressure in the terminal section must be added to determine the path's total pressure requirements.

Table 2.5
Path Total Pressures for Sample System 1 Equal Friction Design

Terminal	**Path (Sections)**	**Section Total Pressure Losses** (*inches wg*)	**Path Total Pressure Required** (*inches wg*)
A	1,2	0.34 + 0.18 + 0.11	0.63
B	1,3	0.34 + 0.18 + 0.11	0.63
C	1,4,5	0.34 + 0.12 + 0.52 + 0.14	1.12
D	1,4,6,7	0.34 + 0.12 + 0.11 + 0.31 + 0.16	1.04
E	1,4,6,8,9	0.34 + 0.12 + 0.11 + 0.12 + 0.38 + 0.11	1.18
F	1,4,6,8,10,11	0.34 + 0.12 + 0.11 + 0.12 + 0.11 + 0.24 + .17	1.21
G	1,4,6,8,10,12	0.34 + 0.12 + 0.11 + 0.12 + 0.11 + 0.27 + 0.23	1.30

From this analysis, we can see that the pressure required to deliver the design volume flow of air to Terminal G is greater than that required for any other path. The critical path, therefore, is from the fan to Terminal G, and the system total pressure requirement is 1.30 *inches wg*. To determine the system static pressure requirement, subtract the initial velocity pressure. In this case, the initial velocity was 2,616 *fpm* (*VP* = 0.43 *inches wg*), so the system static pressure is 0.87 *inches wg*.

The inspection method of **Sample Problem 2-2** provided the correct critical path but the pressure estimate (1.12 *inches wg*) was low by 0.18 *inches wg*. The main reason is because the straight-through losses were ignored. It was guesstimated that the path to Terminal G would be the critical path, which it was. However if the system pressure was estimated at 1.12 *inches wg* and the true pressure required at the fan is 1.30 *inches wg*, then the fan may be selected incorrectly to supply the lower pressure.

In analyzing real systems, it is important to note that there will be other losses, in addition to the duct and fitting losses discussed above. Probably most significant is the pressure loss associated with terminal devices. Typically, diffusers and registers have a pressure requirement ranging from 0.01 to 0.5 *inches wg*. If the system has volume control boxes, there must be sufficient residual pressure at the end of each path to power the box (or overcome the internal resistance of the device) and also to move the air through any low pressure or flexible duct on the downstream side of the box. Losses for the terminal devices should be added to the pressure requirements of each path prior to the determination of the critical path and system pressure.

2.4.4 Excess Pressure

In the preceding equal friction problem, the system operates at 1.30 *inches wg* to deliver air to the critical path (Terminal G). This is more than the pressure necessary to operate the other paths to Terminals A through F. If the system is turned on as designed, air will always follow the path of least resistance, and this will result in excess volume flow through the paths to Terminals A and B and progressively less air through the downstream terminals as the flow resistance increases. The critical path terminal will not get its required volume, while other terminals will get too much air.

Looking again at the pressure requirements for the system, the amount of excess pressure which is present at each terminal can be computed by calculating the difference between the system (critical path) pressure of 1.30 *inches wg* and the individual path pressures.

Table 2.6
Excess Pressure for Sample System 1 Equal Friction Design

Terminal	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
A	0.63	0.67
B	0.63	0.67
C	1.12	0.18
D	1.04	0.26
E	1.18	0.12
F	1.21	0.09
G	1.30	0.00

In order to balance the system, it is necessary to add resistance to each noncritical path, equal to the excess pressure of the path. In this way, each path essentially becomes a critical path and the design volume flow rate will be delivered to each terminal, as long as the air handler provides the total volume flow rate (20,600 *cfm*) at the correct initial velocity (2,616 *fpm*) and system total pressure (1.30 *inches wg*).

The simplest way to introduce resistance to a system is probably with the use of balancing dampers. These dampers can be adjusted to provide a pressure drop equal to the excess pressure for each flow path. Once adjusted, they should be permanently secured to avoid tampering, which will destroy the balance. Obviously, this method of balancing is appropriate only for inflexible constant-volume systems. Balancing is not as easily maintained in a variable air volume (VAV) system. Most often the balancing dampers in a VAV box are capable of handling excess pressure; however, the amount of excess pressure can affect VAV box performance.

Although the use of dampers is simple, it is not necessarily cost effective. **Chapter 3** will discuss an efficient means of balancing systems without the use of dampers.

2.5 Static Regain Design

2.5.1 Introduction

The key concept of static regain design is that the magnitude of the static pressure must be maintained as constant as possible throughout the system. At first this seems to be a contradiction, because static pressure is required to overcome duct friction, and duct friction is produced everywhere air flows over a duct surface. It seems only logical that the static pressure will progressively reduce as air flows from the fan to the terminals.

Remember, however, that static pressure and velocity pressure combine to yield total pressure, and it is the total pressure, not the static pressure, which is irretrievably lost as the air flows through the system. At any location where the duct area and/or volume flow rate change, there will usually be a change in air velocity. Velocity pressure will increase or decrease as the square of the velocity changes.

Referring to **Figure 1.3**, recall that as velocity pressure increases (Case II) there will always be a reduction of static pressure; however, a reduction in velocity pressure can be accompanied by a decrease (Case III) or an increase (Case I) in static pressure. Where static pressure increases, this phenomenon is called static regain, and is the basis of this design method. If the reduction in velocity pressure is greater than the total pressure loss, static regain will occur. If the total pressure loss is greater than the reduction in velocity pressure, then static regain can not occur. The following examples illustrate this concept:

1. In **Sample Problem 1-16**, a straight tee fitting produced a total pressure loss of 0.17 *inches wg*, upstream-to-branch. There was a reduction of 0.08 *inches wg* velocity pressure. This resulted in a static pressure loss of 0.09 *inches wg*. No static regain is possible in this situation.

2. **Sample Problem 1-17** substituted a conical tee and LO-LOSS™ tee under the same conditions. The LO-LOSS™ resulted in a total pressure loss of only 0.06 *inches wg*, upstream-to-branch. Since the velocity pressure was reduced by 0.08 inch wg., this condition resulted in a static pressure regain (increase) of 0.02 *inches wg* (which is written as a negative loss or -0.02 *inches wg*).

3. **Sample Problem 1-19** considered the straight-through leg of the straight tee (which would be the same for conical tee or LO-LOSS™ tee as well). Here the total pressure loss (upstream-to-downstream) was just 0.01 *inches wg*, but the velocity was reduced from 1,592 *fpm* to 955 *fpm*, resulting in a velocity pressure reduction of 0.10 *inches wg*. In this case there was a static pressure regain of 0.09 *inches wg*.

Static regain design uses the increase in static pressure, especially in straight-through legs of divided-flow fittings, to create a well-balanced system.

2.5.2 Duct Sizing

In static regain design, the first section (connecting to the fan or plenum) is sized for a desired velocity. This velocity may be any value, but the rules of thumb in **Table 2.1** are often used for round and flat oval ductwork. However, if silencers or rectangular ductwork are involved, velocities are generally held to 2,500 *fpm* maximum in order to minimize additional pressure drop.

To determine the optimum velocity, a cost analysis would have to be performed. This is demonstrated in **Section 3.5**.

All downstream duct is then sized so that the total pressure loss in each section is just equal to the reduction in velocity pressure such that the change in static pressure is zero from the beginning of one branch fitting or takeoff to the beginning of the next branch or takeoff fitting. Selecting a large duct size will result in greater regain in the fitting and less pressure loss in the duct and section=s fittings. By reducing the duct size, the static regain will be reduced (since the change in velocity pressure is reduced) and the duct and fitting pressure losses will be increased. When the velocity pressure reduction is about equal to the section=s duct and fitting losses, the section is correctly sized. This will result in the section static pressure loss being equal to approximately zero. Thus the goal of maintaining the same static pressure throughout the system is approached, which means the system will be balanced. However in many branch systems, it is almost impossible to decrease the velocity enough to offset the section=s losses without increasing the duct size above the upstream size. Therefore, when designing downstream sections, it is usual and practical to begin with the size of the section just upstream. The end result will be some imbalance in the system, which either must be dampered, or sections resized after the initial design to remove the excess pressure, which is the topic of **Chapter 3 Analyzing and Enhancing Positive Pressure Air Handling Systems**.

The following sample problem sizes the duct sections of **Sample System 1 (Figure 2.1)** using the Static Regain Design method.

Sample Problem 2-3

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*Refer to **Sample System 1** (**Figure 2.1**). Assume that Section 1 is sized at 38 inches and will carry 20,600 cfm. Use the static regain method to determine the size of Section 4.*

Answer: A 38-*inch* duct carrying 20,600 *cfm* will have a velocity of 2,616 *fpm*. First, consider keeping the downstream diameter (Section 4) the same size at 38 *inches*. Duct Section 4 is known to be 36 *feet* long and will carry 17,400 *cfm*. A 38-*inch* diameter duct at 17,400 *cfm* would have a velocity of 2,209 *fpm* and from the friction loss chart, a friction loss of 0.15 *inches wg per 100 feet*. The friction loss of the 36-*foot* duct section will be (36/100) H 0.15, or 0.05 *inches wg*. From ASHRAE Fitting **SD5-25**, at 2,209 *fpm* straight-through (downstream) and 2,616 *fpm* common (upstream), DTP_{c-s} = 0.04 *inches wg* across the straight-through portion of the cross fitting. The total section pressure loss is 0.09 *inches wg*. The velocity pressure drop, DVP_{c-s}, across the straight-through is 0.12 *inches wg*. Since the drop in velocity pressure exceeds the total pressure drop by 0.03 *inches wg*, this will be the static regain in Section 4. Since the static regain is greater than zero (in magnitude), we will try a smaller size. Try using 37-*inch* diameter duct.

A 37-*inch* diameter duct at 17,400 *cfm* will have a velocity of 2,330 *fpm*, and a friction loss of 0.17 *inches wg per 100 feet*. The friction loss of the 36-*foot* duct section will be (36/100) H 0.17, or 0.06 *inches wg*. DTP_{c-s} is 0.05 *inches wg* across the straight-through portion of the cross fitting. The total section pressure loss is 0.11 *inches wg*. The velocity pressure drop, DVP_{c-s}, across the straight-through is 0.09 *inches wg*. Since the drop in velocity pressure is 0.02 *inches wg* less than the total pressure drop, there is a static pressure loss of 0.02 *inches wg* and therefore no static regain.

To get a static regain (or pressure loss) equal to zero, we would need to use a duct size between 37 and 38 *inches* in diameter. This is usually not practical and either the 37- or 38-*inch* duct would be acceptable here. Use the size that either gives the closest to zero static regain. If the choices are about equal, use the size which results in the lowest cost, considering the cost of a reducer if it must be added to use a smaller size. For our example problem we will keep the 38-*inch* diameter for Section 4.

There are several important points to keep in mind when using static regain design:

1. The straight-through (downstream) duct sizes should generally not be made larger than common (upstream) duct sizes for the sole purpose of achieving a static regain. If it is not possible to achieve regain using a constant diameter duct (downstream the same size as upstream), then size the section the same as the upstream duct diameter and continue with the design, trying to achieve regain in the next downstream section.

2. When a section branches off the main trunk duct, it may not be possible to achieve regain with certain branch fittings. The straight tee in **Sample Problem 1-16** presents such an example. In these cases, substitution of a more efficient fitting may lower the total pressure loss enough that static regain is possible (see the LO-LOSS tee of **Sample Problem 1-17**). If not, use the largest duct size that is compatible with the space restrictions and minimum velocity criterion (see point 4, below).

3. Consider using a constant diameter duct until a reduction can be justified by less cost. Often a reducer and smaller size duct will cost more than keeping the same size if the reduction is only an *inch* or two and the size is large. The choice between the same size and next lower size should only occur if the static regain of the larger size is about equal in magnitude to the static pressure loss of the smaller size.

4. Select a minimum velocity, and do not use (larger) duct sizes that will cause the velocity to fall below this value. Especially on divided-flow fitting branches, a point may be reached where a larger duct size produces no measurable difference in the fitting pressure loss.

Table 2.7 is a worksheet that shows the iterations and results of a static regain analysis of **Sample System 1** (**Figure 2.1**). Section 1 is sized at 38 *inches*, the fittings are described in **Sample Problem 2-2**, and a minimum velocity of 900 *fpm* is to be maintained. Straight-through sections are treated first (Sections 4, 6, 8 and 10), followed by the branches (Sections 2, 3, 5, 7, 9, 11 and 12). An asterisk shows selected sizes. The **ASHRAE Duct Fitting Database** was used to determine the loss coefficients.

The following comments may be helpful in understanding how sizes were selected.

<u>Section 4</u>: See **Sample Problem 2-3**.

<u>Section 6</u>: The first size iteration is 38 *inches*, the upstream diameter. A slight overall static regain (DSP = -0.02 *inches wg*) is achieved, so a second iteration is made at 37 *inches*. This produces a section static pressure loss of 0.02 *inches wg*. Any duct size smaller than 37 *inches* will further increase the section static pressure loss. Therefore, the 38-*inch* duct is selected (see comment 3, above).

<u>Section 8</u>: The first iteration is again 38 *inches* which produces a section static regain (DSP = -0.04 *inches wg*). A 37-*inch* duct produces a slight static regain (DSP = -0.01 *inches wg*) , so we should not use 38-*inch* duct again since it would result in an even greater regain. A 36-*inch* duct produces a slight section static pressure loss (DSP = 0.01 *inches wg*), but this size is selected instead of the 37-*inch* duct (see comment 3, above).

<u>Section 10</u>: The upstream (Section 8) diameter of 36 *inches* is selected for the first iteration, and this result in a section static regain (DSP = -0.00 *inches wg* to two decimal places). Since this is what we want we will keep the 36-*inch* diameter.

<u>Sections 2 and 3</u>: These branches are identical, so they are considered together. Selecting the first iteration size for straight-through sections is a simple matter. It is usually the upstream section size or 1 or 2 *inches* smaller. Branches sizes are not as simple to select. In this case, the first iteration was selected at a size that would yield a velocity near the minimum. This resulted in a slight regain (DSP = -0.02 *inches wg*). A second iteration at 16-*inch* diameter resulted in a slight static pressure loss (DSP = 0.01 *inches wg*). This is closer to zero so this size is selected.

<u>Section 5, 7, 9 and 11</u>: The first iterations for these sections were also based on sizes the produced air flow close to the minimum velocity. In each of these cases there was already no static regain. Since we could not make the duct sizes larger without violated the minimum velocity constraint, these sizes were chosen.

<u>Section 12</u>: This section is a fairly long run and has a higher air volume flow rate than the other branches, but the first iteration was again based on the size that produced a velocity close to the minimum, which is 36 *inches*. This produced a section static pressure regain (DSP = -0.03 *inches wg*). A 34-*inch* duct resulted in a 0.03 *inches wg* section static pressure loss. Either 34-*inch* or 36-*inch* duct will be acceptable from a pressure loss standpoint, so a 34-*inch* diameter is selected to save in material cost.

There is more information given in **Table 2.7** than is actually necessary to perform the design calculations. This is done to show the relationships among static, velocity, and total pressure losses in fittings and sections.

For the bullhead tee, it is necessary to determine the loss coefficient from **ASHRAE Fitting SD5-19**. The static and total pressure losses can then be determined from **Equations 1.17b** and **1.17c**. The section static pressure loss (DSP_{sect}) is a summation of DP_{duct} (remember for duct, $DSP = DTP$) and DSP_{ftg}. For an ideal static regain design, this loss should be zero.

The section total pressure loss (DTP_{sect}) is important as a measure of total system energy consumption. The following equations will always apply, and can serve as a data check.

$$\boldsymbol{DTP_{sect} = DSP_{sect} + DVP_{sect}} \qquad \textbf{Equation 2.1}$$

$$\boldsymbol{DTP_{sect} = DP_{duct} + SDTP_{ftgs}} \qquad \textbf{Equation 2.2}$$

2.5.3 Determination of System Pressure

To determine the system pressure loss, it is necessary to find the critical path, as was done for the equal friction problem. Remember that the path with the highest pressure requirement (fan to terminal) is the critical path, and this pressure is the system pressure requirement. Therefore, it is necessary to identify the critical path and to determine its pressure loss. Determination of the total pressure losses for a system that has already been sized using the Static Regain design method is accomplished by adding the pressure losses for each section in a path from the fan to the terminal.

Table 2.8 represents the pressure loss analysis for **Sample System 1**, based on the static regain analysis of **Table 2.7**. The critical path is from the fan to Terminal C and requires 0.86 *inches wg* to deliver the required volume flow of air. The system static pressure requirement is 0.43 *inches wg, which* is determined by subtracting the initial velocity pressure of 0.43 *inches wg* from the system total pressure requirement.

2.5.4 Excess Pressure

The amount of excess pressure that is present at each terminal can be computed by calculating the difference between the system (critical path) pressure and the individual path pressures. Results of the evaluation of excess pressure for **Sample System 1** are shown in **Table 2.9**.

It is interesting to note that in the equal friction problem presented previously the terminals closest to the fan had the highest excess pressure and the terminals away from the fan had the highest path pressure requirements. However, static regain design creates the possibility of regain at each straight-through fitting, so sections remote from the fan (which have benefited from several straight-through regains) may actually have lower total pressure requirements than those sections closer to the fan. Also the static regain design required less overall pressure (0.86 *inches wg* compared to 1.30 *inches wg*) even though the first section sizes were the same for both. The static regain design was also much better balanced with excess pressures in the range of 0.05 *inches wg* to 0.13 *inches wg* compared to the equal friction design which has excess pressures in the range of 0.09 *inches wg* to 0.51 *inches wg*.

Excess pressures of 0.05 *inches wg* or less are considered negligible. Therefore, this static regain designed system is still not considered fully balanced and will require additional balancing dampers or balancing by use of VAV box dampers.

Table 2.7
Sample System 1 - Static Regain Design Data Sheet

SECT	ITEM REF	*Q* (*cfm*)	DIA. (*inches*)	AREA (ft^2)	*V* (*fpm*)	*VP* (*inches wg*)	*C*	Δ *P/100 ft* (*inches wg*)	*L* (*ft*)	Δ *P* DUCT (*inches wg*)	FITTING Δ*SP* (*inches wg*)	SECTION Δ*SP* (*inches wg*)	FITTING Δ*TP* (*inches wg*)	SECTION Δ*TP* (*inches wg*)
1*	CD3-9	20600	38	7.88	2616	0.43	0.12	0.20	145	0.29	0.05	0.34	0.05	0.34
4a*	SD5-25	17400	38	7.88	2209	0.30	0.14	0.15	36	0.05	-0.08	-0.03	0.04	0.09
4b	SD5-25	17400	37	7.47	2330	0.34	0.14	0.17	36	0.06	-0.04	0.02	0.05	0.11
6a*	SD5-10	15000	38	7.88	1905	0.23	0.13	0.11	30	0.03	-0.05	-0.02	0.03	0.06
6b	SD5-10	15000	37	7.47	2009	0.14	0.13	0.07	30	0.04	-0.02	0.02	0.03	0.07
8a	SD5-10	11800	38	7.88	1498	0.14	0.15	0.07	40	0.03	-0.07	-0.04	0.02	0.05
8b	SD5-10	11800	37	7.47	1580	0.16	0.15	0.08	40	0.03	-0.05	-0.01	0.02	0.06
8c*	SD5-10	11800	36	7.07	1669	0.17	0.14	0.09	40	0.04	-0.03	0.01	0.02	0.06
10*	SD5-10	10200	36	7.07	1443	0.13	0.13	0.07	35	0.03	-0.03	-0.00	0.02	0.05
2 and 3 a	SD5-25	1600	18	1.77	905	0.05	6.68	0.07	20	0.01	-0.03	-0.02	0.34	0.36
2 and 3 b*	SD5-25	1600	16	1.40	1146	0.08	4.01	0.12	20	0.02	-0.02	0.01	0.33	0.35
5*	SD5-10 CD3-9	2400	22	2.64	909	0.05	6.75 0.14	0.06	50	0.02	0.10 0.01	0.13	0.35 0.01	0.38
7*	SD5-10	3200	25	3.41	939	0.05	3.33	0.05	27	0.01	0.01	0.03	0.18	0.19
9*	SD5-10	1600	18	1.77	905	0.05	3.72	0.07	24	0.02	0.07	0.08	0.19	0.21
11*	SD5-19	3600	27	3.98	905	0.05	1.51	0.04	33	0.01	-0.00	0.01	0.08	0.09
12a	SD5-19 CD3-14	6600	36	7.07	934	0.05	1.19 0.07	0.03	48	0.02	-0.05 0.00	-0.03	0.03 0.00	0.05
12b*	SD5-19 CD3-14	6600	34	6.31	1047	0.07	1.00 0.07	0.04	48	0.01	0.01 0.00	0.03	0.07 0.00	0.08

Table 2.8
Path Total Pressure for Sample System 1 Static Regain Design

Terminal	**Path (Sections)**	**Section Total Pressure Losses** (*inches wg*)	**Path Total Pressure Required** (*inches wg*)
A	1,2	0.34 + 0.35 + 0.08	0.77
B	1,3	0.34 + 0.35 + 0.08	0.77
C	1,4,5	0.34 + 0.09 + 0.38 + 0.05	0.86
D	1,4,6,7	0.34 + 0.09 + 0.06 + 0.19 + 0.05	0.73
E	1,4,6,8,9	0.34 + 0.09 + 0.06 + 0.06 + 0.21 + 0.05	0.81
F	1,4,6,8,10,11	0.34 + 0.09 + 0.06 + 0.06 + 0.05 + 0.09 + 0.05	0.74
G	1,4,6,8,10,12	0.34 + 0.09 + 0.06 + 0.06 + 0.05 + 0.08 + 0.07	0.75

Table 2.9
Excess Pressure for Sample System 1 Static Regain Design

Terminal	**Path Total Pressure Required** (*inches wg*)	**Excess Pressure** (*inches wg*)
A	0.77	0.09
B	0.77	0.09
C	0.86	0.00
D	0.73	0.13
E	0.81	0.05
F	0.74	0.12
G	0.75	0.11