Design Advisory #3: CAS-DA3-2003

Caveat Emptor—Product Substitutions Not Always a Value

Incorporating good duct system design practices before specifying equipment and controls will help ensure that system performance requirements are met effectively and efficiently.

Unfortunately, time constraints and evolving budget concerns many times force designers and contractors to compromise those better design practices and substitute products that worked in a similar application and apply them to the current project.

Product substitutions are not necessarily bad, provided the changes account for system needs as opposed to simply reducing the price of the components used. They can put a project back on schedule or within budget.

But buyer beware—they may also drastically undermine the performance of the system. Chapter #3: *Analyzing and Enhancing Supply Duct Systems* from McGill AirFlow's *Duct System Design Guide* (DSDG) provides guidelines for avoiding the pitfalls and optimizing the product substitution process.

Hundreds of times each year McGill AirFlow's duct system design expertise is called upon to analyze and recommend less expensive systems. Since product substitutions help bring projects within budget, our engineers work with the interested parties for a mutually beneficial solution that meets the system's performance needs at a lesser cost.

Many long-standing and proven product selection and manufacturing techniques allow for substitutions requiring few system redesigns. These include:

- substituting LO-LOSS™ or boot tap fittings for lateral plus 45° elbows or conical fittings
- shop-assembling taps to duct (manifolding) in lieu of installing full-body fittings and duct
- incorporating non-reinforced, multiple round duct instead of heavily reinforced flat oval or rectangular duct
- partially insulating versus completely insulating duct systems to meet acoustic criteria.

Product substitutions that cause system alterations can only be done accurately using the analyzing and enhancing methodologies detailed in Chapter #3, preferably in conjunction with industry-proven duct system design software.

Product substitutions and design alternatives do provide significant cost-saving options. The key to successful product substitution is requiring and monitoring supportive documentation that demonstrates the effectiveness of the substitutions to meet the performance requirements of the system.

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CHAPTER 3: Analyzing and Enhancing Supply Duct Systems

3.1 Analyzing a Preliminary Supply Design

After a duct system has been initially designed using one of the methods discussed in **Section 2.3**, it should be reviewed to see if improvements can be made. The first step in analyzing a system is to determine the total pressure required to operate it. The method of determining the system's total pressure requirements is shown in **Section 2.4.3** for the equal friction design method and **Section 2.5.3** for the static regain design method. In both cases, each path's total pressure requirement is calculated and the path with the highest total pressure requirement is the system's design leg or critical path. This is shown in **Table 2.7** of **Section 2.5** and repeated below in **Table 3.1**.

TABLE 3.1
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

By analyzing a system using total pressure, the designer can more easily see areas of inefficiency. High pressure losses in sections or paths can be lowered by increasing sizes or using more efficient fittings. For sections or paths where there is too much pressure (unbalanced), the designer can reduce duct sizes, use less efficient fittings, or put in balancing devices such as dampers or orifices.

3.2 Balancing Equal Friction Designs

The amount of balancing any path requires is simply the path's excess pressure. To balance a system, three methods are often employed for equal friction in non-design legs: balancing dampers, orifice plates, and enhanced equal friction design.

3.2.1 Balancing Dampers

Balancing dampers are the most common method of balancing equal friction designs because calculations to determine the damper setting are not usually required when the system is being designed. The damper setting is usually determined by measuring the air volume flow rate (*cfm*) in the field after installation. The damper is adjusted until the required air volume for the terminal device is correct. The required angle of the balancing damper can be determined, however, if the excess pressure is known and the relationship between the damper angle and the loss coefficient for the damper is known.

For example, **Table 3-2** from ASHRAE Fitting **CD9-1** lists loss coefficients for round butterfly dampers of type shown in **Figure 3.1**.

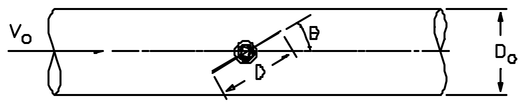

Figure 3.1
Round Butterfly Damper

Table 3.2
Round Butterfly Damper Loss Coefficients

D/D_o	Loss Coefficient *C* — Damper Angle *q*											
	0E	**10E**	**20E**	**30E**	**40E**	**50E**	**60E**	**70E**	**75E**	**80E**	**85E**	**90E**
0.5	0.19	0.27	0.37	0.49	0.61	0.74	0.86	0.96	0.99	1.02	1.04	1.04
0.6	0.19	0.32	0.48	0.69	0.94	1.21	1.48	1.72	1.82	1.89	1.93	2.00
0.7	0.19	0.37	0.64	1.01	1.51	2.12	2.81	3.46	3.73	3.94	4.08	6.00
0.8	0.19	0.45	0.87	1.55	2.60	4.13	6.14	8.38	9.40	10.30	10.80	15.00
0.9	0.19	0.54	1.22	2.51	4.97	9.57	17.80	---	---	---	---	---
1.0	0.19	0.67	1.76	4.38	11.20	---	---	---	---	---	---	---

The required loss coefficient is determined from **Equation 1.16a**:

$$C = \frac{\Delta TP}{VP}$$

where *DTP* is the additional pressure required to balance the path (i.e., the path's excess pressure). For the equal friction sample problem in **Section 2.4**, assuming the dampers will be put in the terminal sections, the required loss coefficient and corresponding damper angle required to balance the paths for a D/D_o = 1.0 are shown in **Table 3.3**.

Table 3.3
Sample System No. 1 - Damper Angles

Terminal	Path Excess Pressure (*inches wg*)	Velocity Pressure in Terminal Section (*inches wg*)	Required Loss Coefficient *C*	Required** Damper Angle *q*
A	0.67	0.11	6.09	33E
B	0.67	0.11	6.09	33E
C	0.18	0.14	1.29	16E
D	0.26	0.16	1.63	19E
E	0.12	0.11	1.09	14E
F	0.09	0.17	0.53	07E
G*	0.00	N/A	N/A	N/A

*   Design leg, no dampers required
** Values interpolated to nearest degree

3.2.2 Orifice Plates

The size of an orifice plate required to balance the paths can also be determined if the excess pressure, velocity pressure, and corresponding loss coefficient are known. By assuming the number of holes equals one, ASHRAE Fitting **CD6-2** gives the data shown in **Table 3.4** to determine the loss coefficient of orifice plates.

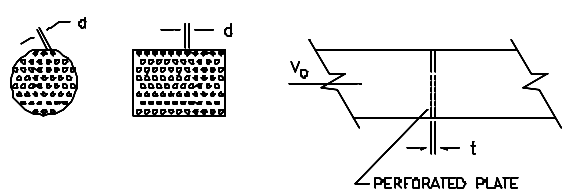

Figure 3.2
Orifice Plates

where (for *t/d*   0.015):

A_o = area of duct (ft^2)

A_{or} = orifice area = $p\,d^2/4$ (ft^2)

D = diameter of perforated hole (*inches*)

n = free area ratio of plate (dimensionless) = $S A_{or}/A_o$

t = plate thickness (*inches*)

Table 3.4
Orifice Plate Loss Coefficients

$\frac{t}{d}$	Loss Coefficient C — n 0.20	0.25	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00
0.015	51.50	30.00	18.20	8.25	4.00	2.00	0.97	0.42	0.13	0.00
0.200	48.00	28.00	17.40	7.70	3.75	1.87	0.91	0.40	0.13	0.01
0.400	46.00	26.50	16.60	7.40	3.60	1.80	0.88	0.39	0.13	0.01
0.600	42.00	24.00	15.00	6.60	3.20	1.60	0.80	0.36	0.13	0.01
0.800	34.00	19.60	12.20	5.50	2.70	1.34	0.66	0.31	0.12	0.02
1.000	31.00	17.80	11.10	5.00	2.40	1.20	0.61	0.29	0.11	0.02
1.400	28.40	16.40	10.30	4.60	2.25	1.15	0.58	0.28	0.11	0.03
2.000	27.40	15.80	9.90	4.40	2.20	1.13	0.58	0.28	0.12	0.04
4.000	27.70	16.20	10.00	4.60	2.25	1.20	0.64	0.35	0.16	0.08
6.000	28.50	16.60	10.50	4.80	2.42	1.32	0.70	0.40	0.21	0.12
8.000	30.00	17.20	11.10	5.10	2.58	1.45	0.80	0.45	0.25	0.16
10.000	31.00	18.20	11.50	5.40	2.80	1.57	0.89	0.53	0.32	0.20

For the equal friction sample, again assume the orifices will be put in terminal sections. The path excess pressures, terminal velocity pressures, and required loss coefficients are the same as those in **Section 3.2.1** for balancing with dampers. For orifice plate balancing, there is generally only one hole of diameter ***d***. The terminal diameters are in the range of 15 *inches* to 25 *inches*. An orifice thickness (t = **3** *inch*) is assumed. The required loss coefficient and corresponding orifice diameter to balance each terminal section are given in **Table 3.5**.

Table 3.5
Sample System No. 1 - Required Orifices

Terminal	Path Excess Pressure (*inches wg*)	Terminal Section Diameter (*inches wg*)	Required Loss Coefficient C	Required Open Area Assuming ***t/d*** = 0.015	Required Orifice Diameter ***d*** (*inches*)	Actual ***t/d***
A	0.67	15	6.09	0.45	8.9	0.028
B	0.67	15	6.09	0.45	8.9	0.028
C	0.18	17	1.29	0.67	13.9	0.018
D	0.26	19	1.63	0.64	15.2	0.016
E	0.12	15	1.09	0.69	12.5	0.020
F	0.09	20	0.53	0.78	14.6	0.017
G*	0.00	25	N/A	N/A	N/A	N/A

* design leg, no orifice plate required

3.2.3 Enhanced Equal Friction Design

Enhanced equal friction design incorporates equal friction duct sizing enhanced by total pressure duct balancing. The first step in using this method of balancing is to increase the friction loss per 100 *feet* in the non-design legs to a point where the path's pressure loss is equal to that of the design leg. This can be done systematically by decreasing duct diameters and determining the increased friction loss. The following shows the increase in friction loss and resulting decrease in sizes for non-design leg paths, required to balance the section. Note that decreasing branch sizes also affects the losses of the branch fitting and any in-line fittings (elbows). The following example in **Table 3.6** illustrates how decreasing duct size affects the overall pressure drop in terminals A and B of the equal friction sample problem in **Section 2.4.**

Given: V_c = 2,616 *fpm*, VP_c = 0.43 *inches wg* in Section 1.

Determine: Pressure increase in Sections 2 (Terminal A) and 3 (Terminal B) from a decrease in duct size.

Since the excess pressure in these sections is substantial (0.67 *inches wg*), begin downsizing with a diameter that will yield a velocity equal to upstream Section 1 or 2500 *fpm* maximum in order not to create excessive airflow noise in the branch feeding the outlet.

Table 3.6
Balancing by Duct Downsizing
Terminal A or B

Section Diameter (*inches*)	**Velocity** (*fpm*)	**Duct Loss** (*inches wg*)	**Conical Tee Loss Coeff.** (C_b)	**FITTING** ΔTP (*inches wg*)	**SECTION** ΔTP (*inches wg*)	**Path Total Pressure Required** (*inches wg*)	**Excess Pressure** (*inches wg*)
15	1,304	0.03	1.38	0.15	0.18	0.63	0.67
11	2,424	0.15	1.36	0.27	0.42	1.13	0.17
12	2,037	0.10	0.49	0.28	0.38	0.98	0.32

It should be noted that downsizing is an art and not a science. Proficiency in hand calculating is gained with experience and various combinations. The reduction in diameter from 15 to 11 inches results in an excess pressure of only 0.17 *inches wg* compared to 0.51 *inches wg* pressure. This is much better, but still not balanced against the design leg. Further resistance to flow from the fan to either Terminal A or B is required in order to be the same as from the fan to the end of the design leg which is Terminal G. Keep in mind that downsizing can sometimes cause the cumulative total pressure to exceed the required excess pressure. Exceeding the required excess pressure in a non-design leg should be avoided since it will change the design leg.

Some systems may have a branch connection to a terminal device (i.e., VAV box or diffuser), consisting of a shorter duct length that is generally presized. The presize is to ensure proper operation of the device. Assume the equipment manufacturer's restrictions require a 12-*inch* diameter inlet for proper operation. **Table 3.6** then shows the terminal device must use an excess pressure of 0.32 inches wg in order to balance this branch.

A second step to further balance an equal friction design is to incorporate less efficient fittings to use the excess pressure. Several other types of fittings are evaluated in place of the conical tee in **Table 3.7** in order to see how they balance the branch.

Table 3.7
Balancing by Fitting Substitution
Terminal A or B

Branch Fitting Type	Section Diameter (*inches*)	Velocity (*fpm*)	Duct Loss (*inches wg*)	Loss Coeff (C_b)	FITTING ΔTP (*inches wg*)	SECTION ΔTP (*inches wg*)	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
Conical	12	2,037	0.10	1.07	0.28	0.38	0.98	0.32
Straight	12	2,037	0.10	1.81	0.47	0.57	1.17	0.13
Shop-Fabricated Manifold	12	2,037	0.10	1.95	0.50	0.60	1.20	0.10
Field-Fabricated Manifold	12	2,037	0.10	2.42	0.63	0.73	1.33	N/A

Increased values of loss coefficient (C) result in less efficient fittings. Notice the improved balancing with the use of a less efficient fitting in place of a conical cross. However if the fitting is too inefficient, it can create a new design leg at a higher total pressure requirement as is the case with the Field Fabricated Manifold Cross. Less efficient fittings are generally less expensive but this should always be verified.

Efficient fittings are required only in the design leg of the system. These design leg fittings rarely account for more than 10 percent of the branch fittings of a system. Therefore, using less efficient fittings in non-design legs could result in substantial savings as well as help balance the system. Additionally, using smaller duct will reduce material and installation costs. The best sequence to reduce cost and balance the fittings is to first reduce the branch sizes to increase the friction rate and use as much excess pressure as possible, then use less efficient fittings (junctions and elbows) to further balance the system. If the path is still more than 10% unbalanced with the design leg, use dampers to finalize the balancing in the field. Orifice plates should be avoided since they do not offer the flexibility of field adjustment.

In **Table 3.8**, the other non-design branches are evaluated using the enhanced equal friction design for the sample problem. Again duct downsizing was limited to approximately 2,500 *fpm* for terminal outlet airflow and acoustical performance. Duct system acoustics will be discussed in **Chapter 8**.

Table 3.8
Sample System 1 - Enhanced Equal Friction Design Data Sheet

SECT	ITEM REF	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	0.01	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	0.26	2.90	0.17	20	0.03	(0.01)	0.02	0.31	0.34
2 and 3	SD5-24	Duct Str Cross	1,600	12	0.79	2,037	0.11	1.81	0.26	20	0.13	0.30	0.43	0.47	0.60
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	0.15 0.02	0.27	0.4 0.02	0.52
5	SD5-10 CD3-9	Duct Con Tee 90 Elbow	2,400	16	1.40	1,719	0.18	2.05 0.16	0.26	50	0.13	0.21 0.03	0.37	0.38 0.03	0.54
7	SD5-10	Duct Con Tee	3,200	19	1.97	1,625	0.16	1.60	0.19	27	0.05	0.07	0.12	0.26	0.31
7	SD5-10	Duct Con Tee	3,200	13	1.40	2,292	0.33	0.79	0.43	27	0.12	0.27	0.39	0.26	0.38
9	SD5-10	Duct Con Tee	1,600	15	1.23	1,304	0.11	3.22	0.17	24	0.04	0.13	0.14	0.34	0.38
9	SD5-9	Duct Str Tee	1,600	15	1.23	1,304	0.11	4.29	0.17	24	0.04	0.25	0.29	0.45	0.50
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	0.08 0.02	0.19	0.16 0.02	0.27

The rows that are shaded in **Table 3.8** are sections where smaller sizes and/or less efficient fittings were used in non-design legs to help balance the system. **Table 3.9** shows the before and after excess pressures to each of the terminals.

Table 3.9
Path Total Pressures for Sample System 1 Equal Friction Design

Terminal	Path (Sections)	Before Balancing		After Balancing	
		Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
A	1,2	0.79	0.51	1.17	0.13
B	1,3	0.79	0.51	1.17	0.13
C	1,4,5	1.12	0.18	1.18	0.12
D	1,4,6,7	1.04	0.26	1.26	0.04
E	1,4,6,8,9	1.18	0.12	1.28	0.02
F	1,4,6,8,10,11	1.21	0.09	1.21	0.09
G	1,4,6,8,10,12	1.30	0.00	1.30	0.00

Downsizing the duct and substituting a less efficient straight cross for the conical cross for the Terminal A and B sections 2 and 3, decreased the excess pressure from 0.51 *inches wg* to 0.13 *inches wg*. A rule of thumb is that if the excess pressure is less than 10 percent of the pressure required to operate the design, the path is considered well balanced. A designer may still want to consider dampers in these sections for fine tune adjustments since the excess pressure is just at 10 percent of the design total pressure. The other paths are fairly well balanced.

The enhanced equal friction design of **Sample System 1** proved to be quite beneficial. Overall, three duct sizes were reduced and less efficient (and less expensive) fittings were used in two places. Thus the redesigned system is balanced and will have a lower first cost of material without increasing the operating costs.

Downsizing alone improves the balancing the most by increasing the friction rate. Using less efficient fittings further improves balancing. However, in larger systems, often the substitution of less efficient fittings is more prominent in improving both balancing and reducing material cost. Again, the use of smaller ducts by virtue of the balancing process, results in lower first cost. In the case of using less efficient branch fittings, first cost is reduced further. Excess pressure reduction or better balancing means the system will deliver the designed airflow to the individual spaces or zones. The ductwork, by design, is less dependent on dampers and orifice plates. These devices only add to first cost and better balancing can eliminate them. Therefore, the enhanced equal friction design method can improve the systems balancing and reduce first cost.

3.3 Enhanced Static Regain Design

Enhanced static regain design is similar to the enhanced equal friction design just discussed in that it incorporates total pressure to balance the system. The difference, as the name implies, is that the duct downsizing and fitting changeouts are applied to static regain design instead of equal friction design. The comparative evaluation between static regain and equal friction in **Sample System 1** shows that if the friction loss in Section 1 of both designs are the same, the static regain design will have (1) lower overall design total pressure requirements, (2) larger duct sizes and, (3) significantly better balancing then the equal friction design. Enhanced static regain design is a more efficient design method than enhanced equal friction, especially when used with computer-aided duct design. Fewer duct downsizing and fitting selection considerations are required to balance the system because the static regain design is nearly balanced to begin with.

Sample System 2 in **Figure 3.3** uses the following design parameters to compare the difference between conventional equal friction and static regain designs with the enhanced static regain design. Although not shown, the enhanced equal friction design can exceed the performance/cost attributes of the static regain design.

Design parameters for **Sample System 2**:

1. No height restrictions for main trunk.
2. Height restriction of 12 *inches* for all other branches.
3. Required SP at outlets of 0.5 *inches wg*.
4. For equal friction and static regain design, all 90E tees are conical, all crosses are conical, and all 45E laterals are conical.
5. For enhanced static regain design, default fittings are substituted in non-design legs.

When height restrictions were not met by the design, flat oval duct was used. A friction loss factor of 0.10 *inches wg per100 ft* was used for the equal friction design, and the system static pressure for the static regain and enhanced static regain designs method was matched to this as closely as possible for an apples-to-apples evaluation. **Table 3.10** Comparison of System Operating Pressures and Sizes and **Table 3.11** Comparison of System Balancing show the results of this analysis. For equal overall pressure drop between designs, the enhanced static regain design yields smaller duct and fittings. Of the 59 sections, nearly 41 percent were reduced by the enhancing process, and the resultant average excess pressure of 0.06 *inches wg* was nearly 54 percent less than the equal friction method and 40 percent less than the static regain method. The enhanced static regain method also uses less expensive and less efficient fittings appropriate for the available pressure and this helps to further balance the system and reduce costs.

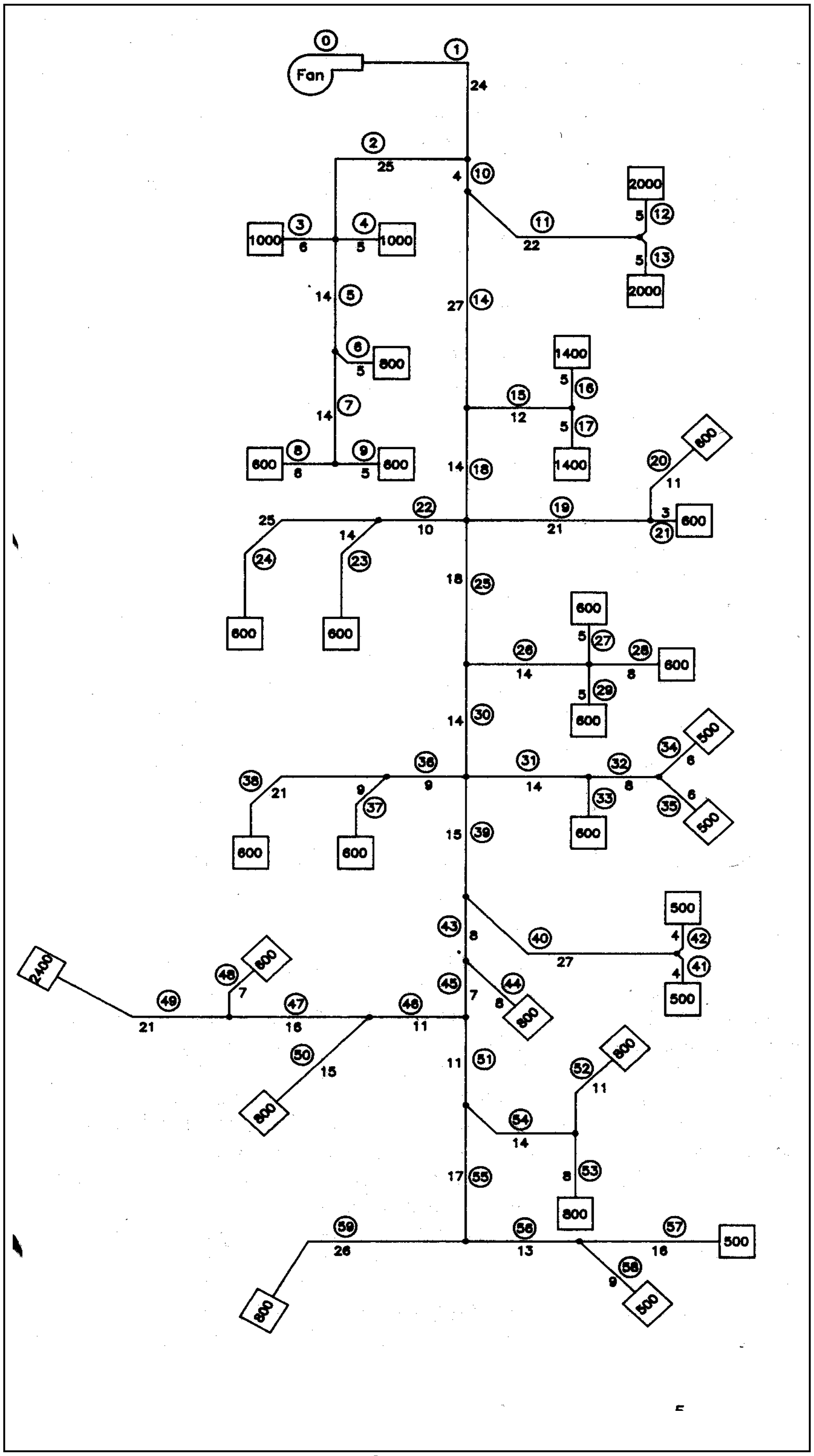

Figure 3.3
Sample System 2

Table 3.10
Sample System 2
Comparison of System Operating Pressures and Sizes

DESIGN METHOD	**EQUAL FRICTION**	**STATIC REGAIN**	**ENHANCED STATIC REGAIN**	
System Total Pressure (*inches wg*)	1.06	1.04	1.04	
System Static Pressure (*inches wg*)	0.78	0.76	0.76	**SECTION NUMBER**
Section Size: Round & Flat Oval (inches)	48	48	48	1
	12 x 45	12 x 45	12 x 45	2
	12 x 15	12	11	3
	12 x 15	12	11	4
	12 x 25	12 x 21	12 x 20	5
	12 x 14	11	10	6
	12 x 18	12 x 15	12 x 14	7
	12	9.5	10	8
	12	9.5	10	9
	46	46	46	10
	12 x 45	12 x 42	12 x 31	11
	12 x 25	12 x 21	12 x 18	12
	12 x 25	12 x 21	12 x 18	13
	43	45	45	14
	12 x 34	12 x 28	12 x 21	15
	12 x 20	12 x 15	12 x 15	16
	12 x 20	12 x 15	12 x 15	17
	40	44	44	18
	12 x 18	12 x 14	12	19
	12	9.5	9.5	20
	12	9.5	8.5	21
	12 x 18	12 x 14	11.5	22
	12	9.5	9	23
	12	9.5	9.5	24
	37	43	43	25
	12 x 25	12 x 20	12 x 42	26
	12	9.5	8	27
	12	9.5	8.5	28
	12	9.5	8	29
	36	42	42	30
	12 x 21	12 x 18	12 x 14	31
	12 x 15	12	11	32
	12	9.5	9.5	33
	11	8.5	8.5	34
	11	8.5	8.5	35
	12 x 18	12 x 14	11.5	36
	12	10.5	9	37
	12	9.5	9.5	38
	32	36	39	39
	12 x 15	12	11	40
	11	8.5	8.5	41
	11	8.5	8.5	42
	31	34	36	43
	12 x 14	11	9.5	44
	30	33	34	45
	12 x 42	12 x 37	12 x 34	46
	12 x 34	12 x 31	12 x 31	47
	12	9.5	10	48
	12 x 31	12 x 25	12 x 28	49
	12 x 14	11	10.5	50
	22	22	24	51
	12 x 14	11	11.5	52
	12	11	11.5	53
	12 x 21	12 x 18	12 x 18	54
	18	16	17	55
	12 x 15	12	12 x 14	56
	11	8.5	9	57
	11	8.5	9	58
	12 x 14	11	11.5	59
Number of Round Duct Sections	53	69	73	

* Friction loss factor for equal friction design was 0.10 *inches wg per 100 feet*.

Table 3.11
Sample System 2
Comparison of System Balancing

DESIGN METHOD	EQUAL FRICTION	STATIC REGAIN	ENHANCED STATIC REGAIN	TERMINAL SECTION
System Total Pressure (*inches wg*)	1.06	1.04	1.04	
System Static Pressure (*inches wg*)	0.78	0.76	0.76	
Path Excess Pressure (*inches wg*)	0.16	0.09	0.00	3
	0.16	0.09	0.00	4
	0.16	0.09	0.06	6
	0.16	0.01	0.10	7
	0.16	0.02	0.10	9
	0.24	0.19	0.11	12
	0.24	0.19	0.11	13
	0.18	0.12	0.11	16
	0.18	0.12	0.11	17
	0.15	0.07	0.02	20
	0.19	0.17	0.07	21
	0.18	0.13	0.03	23
	0.17	0.11	0.08	24
	0.12	0.12	0.00	27
	0.15	0.17	0.07	28
	0.12	0.12	0.00	29
	0.10	0.12	0.04	33
	0.12	0.12	0.09	34
	0.12	0.12	0.09	35
	0.13	0.14	0.06	37
	0.12	0.11	0.10	38
	0.12	0.13	0.05	41
	0.12	0.13	0.05	42
	0.13	0.20	0.08	44
	0.00	0.01	0.01	48
	0.00	0.03	0.01	49
	0.04	0.09	0.00	50
	0.06	0.05	0.09	52
	0.04	0.05	0.10	53
	0.09	0.00	0.08	57
	0.08	0.02	0.08	58
	0.07	0.03	0.09	59
Average Excess	0.13	0.10	0.06	
High Excess	0.24	0.20	0.11	
Average/System SP x 100	17%	13%	8%	
High/System SP x 100	31%	26%	14%	

Note: 0.00 (zero) excess pressure denotes a design leg path.

3.4 Fan Selection

Selection of the fan requires that all duct system resistance be evaluated. In general this resistance consists of the following:

1. Supply and return air duct and fitting losses.
2. In-line equipment losses (coils, filters, silencers, control dampers, terminal outlets, and VAV boxes.)
3. System inefficiencies (system effect) associated with improper fan inlet/outlet connections to the ductwork and inefficient ductwork layouts.
4. Nonstandard environmental conditions (temperature, humidity, and elevation).

System curves, which will be discussed in detail later in this chapter, determine overall flow resistance in the system for a given volume flow rate through the duct. Determination of supply air ductwork pressure losses is addressed fully in **Chapters 2 and 3**. Exhaust and/or return ductwork losses are addressed in detail in **Chapters 4 and 5.**

In-line equipment losses are easily obtained from the equipment manufacturers. Often data for air handlers incorporating a given fan type show an external static pressure (***ESP***) available for ductwork and components external to the unit. Internal losses can include coils, filters, and sometimes silencers. Air handler equipment schedules often list the total static pressure (***TSP***) value, which includes internal components and their associated losses in addition to the ***ESP*** available to overcome ductwork and duct component losses.

A fixed volume flow rate (*cfm*) through a fixed system layout and sizes results in a fixed total (***TP***) and static pressure loss (***SP***). Varying the volume flow rate will result in a change in the pressure loss as shown in **Equation 3.1** and **Figure 3.4.**

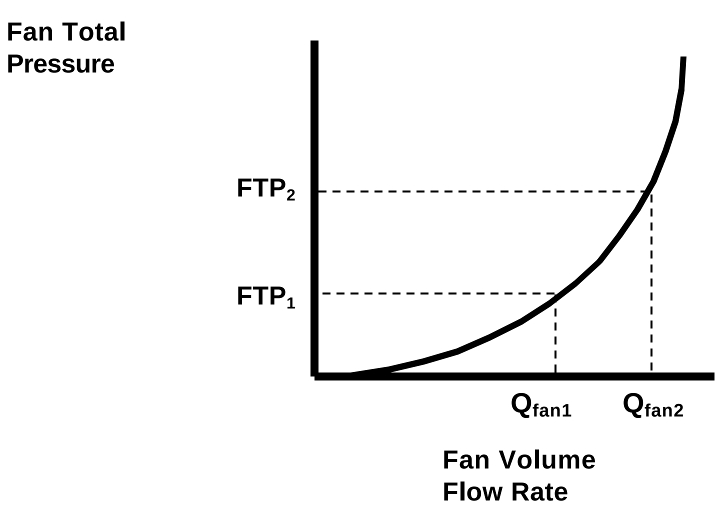

Figure 3.4
System Curve FTP versus Q

$$\frac{FTP_2}{FTP_1} = \left(\frac{Q_{fan2}}{Q_{fan1}}\right)^2$$ **Equation 3.1**

where:

FTP_1 and FTP_2	=	Fan total pressure requirements at Q_{fan1} and Q_{fan2} respectively (*inches wg)*
Q_{fan1} and Q_{fan2}	=	Volume flow rate requirements for systems 1 and 2 respectively (*cfm*)

Sample Problem 3-1

For a given duct system, the fan total pressure is 2.0 inches wg and the volume flow rate is 15,000 cfm. Determine the resultant total pressure and volume flow rate if the system volume flow is increased by 10 percent.

Answer: Rearranging **Equation 3.1** for FTP_2 and substituting the given values, the second condition can be determined. The volume flow rate for an increase of 10 percent is 16,500 *cfm* (1.1 x 15,000). Then

$$FTP_2 = FTP_1\left(\frac{Q_{fan2}}{Q_{fan1}}\right)^2 = 2.0\left(\frac{16{,}500}{15{,}000}\right)^2 = 2.42 \text{ inches wg}$$

Equation 3.1 shows that a 10 percent increase in volume flow rate will yield a 21 percent increase in static pressure for a given duct system. The duct system did not change in the above example problem. Trying to force more air through a given system with fixed duct sizes increases the total pressure requirement significantly. Having two sets of points will help define the system curve for the duct system. For more information on fan performance, see references in **Appendix A.9.2 and A.9.6**

The system curve is based on pressure losses for the supply, return and in-line components for a given volume flow rate. One other factor needs to be known in order to select a fan and that is the system effect factor (SEF).

3.4.1 System Effect Performance Deficiencies

The system effect factor (SEF) was developed to account for deficiencies in fan and system performance associated with improper flow conditions at the inlet and/or outlet of the fan. Fan equipment is normally rated with open inlets and a section of straight duct attached to the outlet. However, real-life installations often include improper outlet conditions, non-uniform inlet flow or swirl at the fan inlet. These conditions alter the aerodynamic characteristics of the fan and the full airflow potential is not realized. **Figure 3.5** gives a graphic presentation of how system effect causes deficient fan performance. Fan testing would be too expensive to model all the possible field conditions. Therefore, the duct designer needs to adjust the pressure calculation to account for these effects.

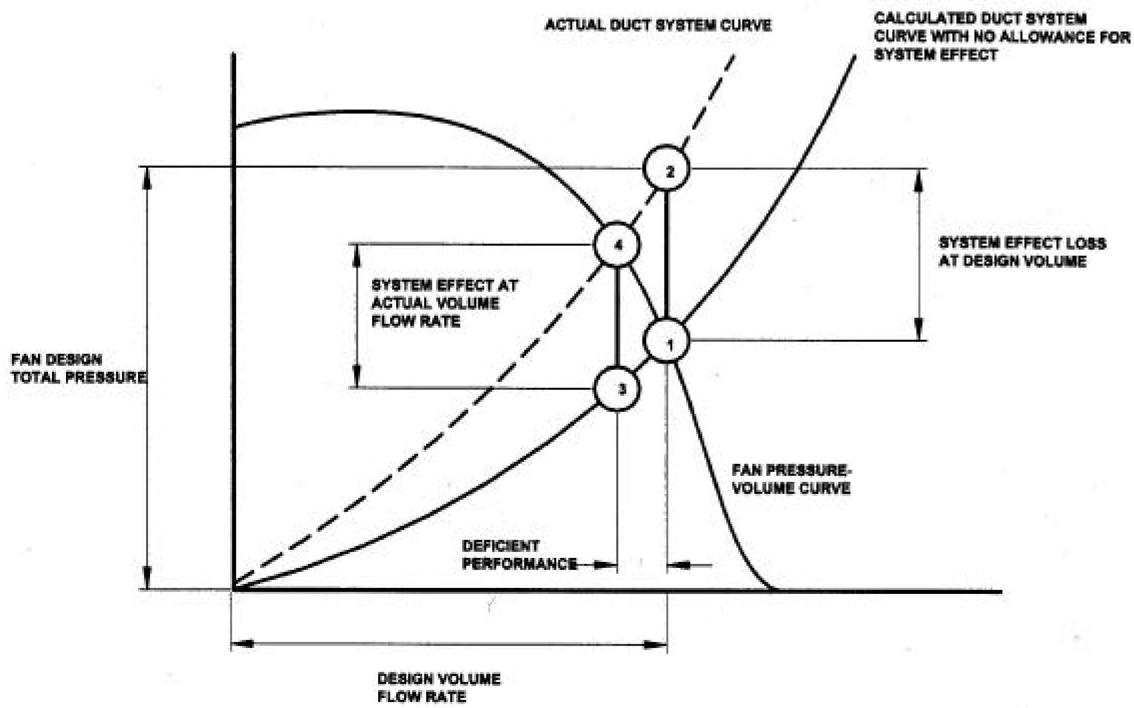

Figure 3.5
Duct System Effect

Point 1 in **Figure 3.5** represents the operating point on a system curve assuming no errors in calculating system resistance in ductwork and components. Proper fan selections require finding a fan performance curve, which passes through point 1. System effect causes added system resistance so a fan operating at a constant speed (rpm) with a higher system resistance will result in a volume flow rate deficiency shown by point 4. Achieving the design volume flow rate would require either a larger fan or increased brake horsepower (bhp) for the original fan. The actual operating point is point 2 on the actual system curve.

Further information on the description and calculation of system effect and system effect factors can be found in **Air Movement and Control Association (AMCA)** publication 201, **Fans and Systems and the ASHRAE 2000 HVAC Systems and Equipment Handbook** Chapter 18, Fans. Additional information for calculating fan inlet/outlet losses can be found **in ASHRAE 2001 Fundamentals Handbook**, Chapter 34, Duct Design (see Appendix A.9.2).

3.4.2 Duct Performance Deficiencies

As with fans, performance data for duct and fittings have much to do with flow conditions through these components based on their position in the system. Loss coefficient data for fittings are generally based on ideal flow conditions. Real-life layouts can often incorporate close-coupled fittings untested standard fittings, and customized fittings. Unlike fans, deficiencies in fittings' resistance are next to impossible to determine because of the variety of applications and factors influencing performance. Some of the more common arrangements have been tested.

These problems and problems associated with duct leakage all contribute to lessening the accuracy in determining a design system curve. Over the years, design engineers have established their own rules of thumb in developing safety factors to account for deficiencies. Many computer-aided duct designs which incorporate most of the available data have proven to be surprisingly accurate despite these deficiencies.

3.4.3 Fan Pressures

Fan selection is generally based on the fan total pressure or fan static pressure. These terms are exclusive to the fan industry and should not be used indiscriminately or confused with similar terms relating to duct system performance.

Fan Total Pressure (FTP) $= TP_{outlet} - TP_{inlet}$ **Equation 3.2**

$$= (SP_{outlet} + VP_{outlet}) - (SP_{inlet} + VP_{inlet})$$

Fan Static Pressure (FSP) $= FTP - VP_{outlet}$ **Equation 3.3**

$$= SP_{outlet} - SP_{inlet} - VP_{inlet}$$

The terms external static pressure (***ESP***) and total static pressure (***TSP***) often appear on air handling unit schedules along with various other performance factors associated with individual components that makeup the unit. The ***ESP*** includes all static pressure losses external to the equipment and, as a minimum, includes the static pressure losses associated with the supply and return air ductwork and any system effect. The **external static pressure loss** can also incorporate losses associated with filters, coils, and silencers for built-up air handling units. However, a majority of air handling units are packaged and include their own filters, coils, and silencers, leaving **external static pressures** for the interconnecting ductwork and any ductwork components (for example, VAV boxes, measuring stations, fire/smoke dampers, additional silencers, etc.). There is not a standard definition of external static pressure so make sure you check with the manufacture as to how much pressure is available for the ductwork system. Use manufacturers' instructions for selection of packaged air handling units. The type and size of fan to be selected depends on many factors and is discussed in detail in ASHRAE publications (**Appendix A.9.2**).

This discussion of fan selection is intended to ensure that all pressure losses are accounted for in determining the most accurate design operating parameters before selecting a fan performance curve. In **Figure 3.5**, point 2 on the system curve is the actual operating point of the system and the point through which the fan performance curve should intersect. If the original fan performance curve was used, the volume flow rate would be deficient by the difference between point 4 and point 2. Point 2, however, is on the actual system curve and needs to intersect a new fan performance curve (not shown on **Figure 3.5**) to supply an adequate volume of air. If we know where the actual system curve intersects the original fan performance curve we can determine the point on the new fan performance curve required using additional **fan laws**. *Remember though that the fan laws only apply to one system curve. You <u>can not</u> use the fan laws to determine the performance from one system curve to another.* That is, referring to **Figure 3.5**, knowing the Fan Total Pressure require for a given Volume Flow Rate such for point 4, you can determine point 2 operating parameters using the fan laws. However, you could not use the fan laws to determine the operating parameters of point from either points 1 or 3 because they are on a different system curve.

Equation 3.1 is a part of the **fan laws,** which govern the performance of fans and predict the fan performance at points of operation other than what was tested. The fan laws determine points of operation when changes are made in speed (rpm), volume flow rate (cfm), and brake horsepower (bhp), etc. The equations here assume that the fan wheel diameter and density ratios are unity. Here are the basic equations:

FAN LAWS

$$\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1}$$

Equation 3.4

where:

Q	=	Volume flow rate of airflow (*cfm*)
rpm	=	Fan speed in revolutions per minute

$$\frac{bhp_2}{bhp_1} = \left(\frac{Q_2}{Q_1}\right)^3$$

Equation 3.5

where:

bhp	=	brake horsepower

Sample Problem 3-2

For Sample Problem 3-1, it is know that the design volume represented by point 1 was to be 15,000 cfm at a total pressure of 2 inches wg. The same fan is to be used as represented by the fan catalog performance curve. The measured total pressure is 2.25 inches wg at 14,660 cfm. What total pressure must the fan reach to maintain the 15,000 cfm design volume flow rate?

Answer: The measured values represent point 4 in **Figure 3.5**. Use **Equation 3.1,** rearranged to solve for FTP at point 2 in a similar manner to what was done in **Sample Problem 3-1**:

$$FTP_2 = 2.25\left(\frac{15{,}000}{14{,}660}\right)^2 = 2.36 \ \textit{inches wg}$$

The fan total pressure must be increased 18 percent, from 2.00 to 2.36 inches wg, to attain the design volume flow rate that was deficient due to the system effect. This also represents an increase in power requirements of 18 percent.

3.5 Cost Optimization

Cost optimization in duct design involves minimizing the owning cost or the net present value of the sum of the initial cost of ductwork and equipment, and the operating cost that could go on for many years.

Here are factors to consider when calculating the net present value:

* Initial cost of material and equipment *Costs of maintenance

* Cost of installation
* Cost of field balancing
* Cost of energy
* Hours of operation
* Tax rates
* Insurance
* Cost of money
* Life of system

For simplification, the following assumptions are made in order to focus and compare the analysis:

1. The fan/motor drive combined efficiency is 75 percent.
2. The fan operates *50 weeks per year*, 7 *days per week*, 16 *hours per day* or 5,600 *hours* total *per year*.
3. Energy cost is $0.08*/kwh*.
4. The project life is 20 *years*.
5. The initial costs are of the duct material only.
6. The inflation rate is 3 percent per year.
7. Fan volume flow rate is constant.

In the following example, consideration is only given to the operating costs and initial cost of the ductwork. The systems annual operating cost can be expressed by **Equation 3.6**.

$$Cost/year = \left(\frac{Q_{fan}\ x\ FTP}{8{,}520\ x\ eff}\right)\ x\ \frac{Hours}{Year}\ x\ \frac{\$}{kwh} \qquad \textbf{Equation 3.6}$$

where:

Cost/year = System first year operating cost (*$*)

Q_{fan} = System volume flow rate (*cfm*)

FTP = System total operating pressure (*inches wg*)

$\frac{Hours}{Year}$ = Number of hours the system is in operation in one year

$\frac{\$}{kwh}$ = Cost of energy

eff = fan/motor drive combined efficiency

8,520 = a conversion factor to kwh (*kilowatt-hours*)

The object of the designer is to reduce or minimize the cost of owning the system. Looking at Equation **3.6**, there are several factors that can reduce the operating cost. The system volume flow rate (*cfm*) should be minimized by performing a thorough load analysis (refer to **Appendix A.9.2**). Variable air volume (VAV) systems can take into account variations in load requirements over periods of time to minimize airflow. Using efficient duct designs with low system effects by using proper connections at the fan can reduce the Fan Total Pressure requirements which has has a direct affect on operating cost. In the previous examples, it was shown that different design methods produce considerable differences in total pressure losses. These differences can result in significant operating cost savings. Selection of high efficiency fan/motor drive components for the system will also help to minimize operating cost.

Items in the equation that normally are out of the control of the designer are operating hours and the cost of energy. Normally, the owner dictates the number of hours the system will be operating annually. Cost of energy, unless it is being generated on site, is out of the control of the designer. Although there are certain times of the day when the cost of energy is lower, the owner may need to operate the duct system during that time. On the first cost side of the analysis, duct material costs are to be included. As was shown earlier, enhancing a duct system by balancing the airflow with smaller sizes and less efficient fittings can reduce first cost. Smaller duct and less efficient fittings are less expensive to purchase and to install. Ease of installation is not considered for this analysis, but it will be discussed later.

In order to calculate the owning cost or *net present value cost* (**NPVC**), the following equations and components are used.

Net present value cost is the sum of the first cost and the product of the present worth factor and the annual operating cost. The present worth factor is calculated from **Equation 3.7** as:

$$PWF = \left(\frac{(1 + IADR)^n - 1}{IADR\,(1 + IADR)^n} \right) \qquad \textbf{Equation 3.7}$$

where:

PWF	=	present worth factor
n	=	project life (*years*)
$IADR$	=	inflation adjusted discount rate (*percent*)

The inflation-adjusted discount rate is defined as:

$$IADR = \left(\frac{1 + ndr}{1 + ir} \right) - 1 \qquad \textbf{Equation 3.8}$$

where:

ndr	=	nominal discount rate (*percent*)
ir	=	Inflation rate (*percent*)

At a nominal discount rate of 20 percent and an inflation rate of 3 percent, the inflation-adjusted discount rate is:

$$IADR = \left(\frac{1+0.20}{1+0.03}\right) - 1 = 16.5 \textit{ percent}$$

The 3 percent inflation adjusted discount rate for discount rates of 30 percent and 5 percent are 26.2 percent and 1.9 percent respectively.

At an inflation adjusted discount rate of 16.5 percent, and a 20-*year* project life, **Equation 3.7** yields:

$$PWF = \left(\frac{(1+0.165)^{20} - 1}{0.165\,(1+0.165)^{20}}\right) = 5.78$$

The present worth factors for *IADR* = 26.2 percent and *IADR* = 1.9 percent are 3.78 and 16.51.

The net present value cost (***NPVC***) is determined by:

NPVC* = *(PWF x Annual Operating Cost) + First Cost **Equation 3.9**

The net present value cost for the *ENHANCED STATIC REGAIN* design of **Figure 3.3** with an initial operating cost of $1,954 and first of $10,580 is determined from **Equation 3.9** as:

= *(5.78 x $1,954) + $10,580* = *$21,874*

Table 3.12 compares the net present value costs for the three discount levels assuming a 3 percent annual inflation and a 20 year project life.

Table 3.12
System Cost Comparison

DESIGN METHOD	EQUAL FRICTION	STATIC REGAIN	ENHANCED STATIC REGAIN
System Total Pressure (*inches wg*)	1.06	1.04	1.04
System Static Pressure (*inches wg*)	0.78	0.76	0.76
First Cost	$13,548	$12,217	$10,580
Annual Operating Cost	$1,992	$1,954	$1,954
Net Present Value Cost at 30% Discount	$21,078	$19,603	$17,966
Net Present Value Cost at 20% Discount	$25,062	$23,511	$21,874
Net Present Value Cost at 5% Discount	$46,436	$44,478	$43,061

At system total pressure of 1.04 *inches wg*, the *ENHANCED STATIC REGAIN* design is 22 percent less expensive on a first cost basis than the equal friction design and, when using a 20 percent discount rate, and almost 13 percent less expensive on a net present value cost basis. The *ENHANCED STATIC REGAIN* design gives a lower material cost (first cost) at any system total pressure. Keep in mind that the lower first cost resulted because of the enhancement of duct and fittings. The smaller ducts and the less efficient fittings in non-design legs helped to reduce first cost.

Working under the assumption that the *ENHANCED STATIC REGAIN* design gives the lowest owning or net present value cost for any system, the owning cost may still not be minimized. Several design should be done (by computer) that vary the initial conditions, such as increasing or decreasing the initial velocity, so that the outcomes are different system total pressure requirements. The owning cost or net present value cost should be calculated for each and plotted as a function of the system total pressure requirement. The minimum value of this curve is the cost optimization point. For more information on Cost Optimization, see **Engineering Report No. 144, Computer-Aided Duct Design: Comparing the Methods**.