Design Advisory #7: CAS-DA7-2004

Don't Touch THAT Damper!

Dampers are strategically placed in exhaust systems to provide balancing and/or shut off functions. Make adjustments to the wrong one and you can effectively ruin the design of an entire system. Several methods for the proper balancing of exhaust systems are discussed in "Chapter 6: Analyzing Exhaust Duct Systems" of McGill AirFlow's *Duct System Design Guide*. The balancing methods shown in that chapter equal those discussed about supply systems in Chapter 3 of the *Guide*. However, you'll soon see how much more difficult exhaust systems are to balance.

Supply systems represent a divergence of energy at each junction that is easily balanced by using up the excess through increased pressure loss associated with inefficient fittings, duct downsizing, and obstructions to the airflow. Exhaust systems represent a convergence of energy at each junction. Any physical change to one branch of a given junction not only affects the energy distribution in the other branches of that junction, but also those of other junctions within the system making them much more difficult to balance.

> Most often shut-off dampers or slide gates should be placed at the branch duct connection to the equipment for easy access by the user. System balancing dampers (sometimes gates or cut-offs) should be located where the branch duct connects to the trunk duct where it is well out of the reach of system users. "DO NOT TOUCH" signage should be incorporated where necessary.

Balancing exhaust systems appears to be an impossible task, but with available duct system design software the several iterations needed to balance the system properly can easily be achieved. Dampers are the most prevalent means of balancing supply or exhaust systems. They are often used along with cut-offs and slide gates as a shut-off device. The duct downsizing and low-efficiency fittings previously mentioned in Chapter 3 are generally used for return systems but not exhaust systems. Why? Because standard exhaust system design practices, especially those for systems conveying particulates, dictate that low-efficiency fittings not be used due to the increased potential for abrasive wear and formation of blockages in the system. Downsizing ductwork increases velocity and thus the potential for wear.

Many exhaust system layouts are rerouted, and the downsized branch run to a piece of equipment at one location will generally not work well once the equipment is relocated. Branch ducts should be sized to match equipment inlet collar connections established for proper carrying velocities.

Dampers offer a great degree of flexibility in the design and balancing of exhaust systems. They also have their drawbacks. Dampers, by design, are an obstruction to the airflow in the ductwork. Abrasive particulates can erode damper components to the point where the unit becomes useless and shuts down the system. Some particulates have the propensity to adhere quickly to themselves and form blockages around even the most minimal obstructions to the airflow (e.g., the leading edge of slip couplings). In these cases, duct downsizing becomes the best solution for balancing the system. Changes to layout will simply require new duct sizing for proper balancing.

Chapter 6 concludes our coverage of airflow design fundamentals in the *Duct System Design Guide*. Greater coverage of this subject can be found in the *Industrial Ventilation Manual* published by the American Conference of Governmental Industrial Hygienists and several chapters in various ASHRAE handbooks as referenced throughout the *Design Guide*. The next *Design Guide* advisory will feature information about acoustics in duct systems.

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CHAPTER 6: Analyzing Exhaust Duct Systems

6.1 Fitting Selection

If a system is exhausting abrasive particulate, the designer must address accelerated abrasive actions on the fitting walls caused by the angular impact of the particulate. These actions are best controlled by the proper choice of fitting types. Fittings that provide a gradual change in airflow direction limit the additional abrasion due to angular impact. The following recommendations apply in these situations:

1. If the particulate is not abrasive, use elbows that have a minimum centerline radius of 1.5 x diameter. If the particulate is abrasive, use elbows that have a minimum centerline radius of 2.5 x diameter. If the amount of abrasive particulate warrants it, consider using a flat back elbow with a wear plate that is replaceable. It should still that have a minimum centerline radius of 2.5 x diameter.

2. For abrasive material, converging-flow fitting branches should enter the main at no greater than a 30E angle. If there is any type of particulate or fumes being exhausted, converging-flow fitting branches should never enter the main at greater than a 45E angle. Converging-flow fitting branches entering the main at a 90E angle are not recommended and should never be used except in low pressure return air systems.

3. Tapered body converging-flow fittings should be considered when particulate fallout will create a hazardous situation or will promote plugging in the duct system.

6.2 Balancing the System

In the previous chapter, system pressure requirements were determined for an exhaust duct system (**Sample Problem 5-4**). This determination of system pressure also illustrated the excess pressure or imbalance in the system. If a system has branches with excess pressure, the flow through the nondesign branches will be greater than the design amount and the flow in the design will be less than the design amount. Airflow will travel to the path with least resistance. To get the correct flow through each branch, the system needs to be balanced. There are several ways to balance a system to obtain the correct airflow.

6.2.1 Using Dampers

One way to balance a system is to add dampers in the nondesign legs. Dampers restrict the flow and cause the pressure in each nondesign branch to increase to the point of excess pressure and thereby balance the system. However a damper can hinder the flow of certain materials through the duct system, causing erosion, particulate buildup, or other negative effects. **Section 3.2.1** discusses balancing supply systems using dampers. Similar concepts apply to return and exhaust systems.

6.2.2 Using Corrected Volume Flow Rate

The other method of balancing is *correcting the air volume flow rate.* Airflow in nondesign legs is increased until that branch or sections excess pressure is zero (thus also becoming a design leg) by matching the systems pressure requirement. **Equation 6.1** can be used to estimate the amount of airflow necessary to balance a section:

$$cfm_{corrected} = cfm_{inlet}\sqrt{\frac{|Fan_{inlet}\ SP|}{|Fan_{inlet}\ SP| - |Excess\ SP|}} \qquad \textbf{Equation 6.1}$$

where:

$cfm_{corrected}$	=	Corrected volume flow rate (*cfm*)
cfm_{inlet}	=	Inlet volume flow rate (*cfm*)
$Fan_{inlet}SP$	=	Inlet to fan static pressure (*inches wg*)
Excess SP	=	Excess static pressure in branch (*inches wg*)

Sample Problem 6-1 illustrates how **Equation 6.1** is used.

Sample Problem 6-1

What is the corrected volume flow rate for section 1 of Sample Problem 5-4? This section's inlet airflow volume is 10,500 cfm, the fan inlet static pressure is -8.85 inches wg, and the excess pressure is 1.13 inches wg.

Answer: From **Equation 6.1**:

$$cfm_{corrected} = 10{,}500\sqrt{\frac{|-8.85|}{|-8.85|-|1.13|}}$$

$$= 11{,}242\ cfm$$

[All analysis are done assuming standard conditions.] Enter the corrected cfm for better balancing in the branch. The corrected cfm is greater than the original by about 16.6 percent. This increase in volume flow rate will change the design and should be accounted for in the design. This method of balancing requires exact layouts, and changes in the system are not recommended since they will upset the balance. Particulate or dust accumulation should not occur if the design conditions are maintained. **Table 6.1** summarizes the results of the increased airflow. The system is more balanced but it is not significant at this point. With the redesign there is only 0.96 *inches wg* of excess pressure in section 1 compared to 1.13 *inches wg* without the extra airflow. We would need even more air, so another iteration would need to be done to further increase the airflow. Already though we have increased the fan inlet total pressure requirement to -8.42 *inches wg* from the -7.78 *inches wg* without the additional airflow and the airflow has increased by 742 *cfm* as well. Both of these directly affect the fan power requirements. Therefore, increasing the airflow may eventually balance the system, but the cost of the increased power consumption may not be worth it.

6.2.3 Using Smaller Duct Sizes and Less Efficient Fittings

Probably the most economical way to balance an exhaust system is similar to what we used in **Section 3.2.3**. By using smaller duct sizes in the nondesign legs of the system, we increase the friction rate and typically increase the dynamic losses of any fittings because of the higher velocities encountered. To do this, increment the duct size smaller until it creates a new design leg, then keep the previous size. The result will be a better-balanced system, with the benefit of smaller sizes that are less expensive and easier to install***; at no additional operating cost***. The system will operate with the same airflow volume and total pressure requirements as in the original design.

Sample Problem 6-2

Resize section 1 of Sample Problem 5-4 to produce a balanced system.

Answer: Three iterations were required to balance the system from Sample Problem 5-4. Again all calculations are done at standard conditions.

The first iteration begins with decreasing the size in section 1 of the system to 20 *inches* from 21 *inches*. This reduced the excess pressure in section 1 to 0.71 *inches wg*. Note that the fan inlet pressure requirement (the negative of the highest total cumulative *DTP*) actually decreased slightly to -7.71 *inches wg* from -7.78 *inches wg* in Sample Problem 5-4. That is because when we change the size (or airflow) of one of the branches of a converging flow fitting, the loss coefficient of both branches is changed. For this first iteration with the smaller size in section 1, the straight-through loss coefficient, C_s, increased from -0.07 to 0.01 while the branch coefficient, C_b, decreased from 0.11 to 0.05. A summary of these results is given in **Table 6-2a**.

For the second iteration, the size in section 1 was decreased to 19 *inches*. This reduced the excess pressure in section 1 to 0.13 *inches wg*. Again, the fan inlet total pressure requirement decrease slightly, this time to -7.60 *inches wg*. For this iteration, the straight-through loss coefficient, C_s, increased to 0.07 while the branch coefficient, C_b, decreased -0.05. A summary of these results is given in **Table 6-2b**.

For the third iteration, the size in section 1 was decreased to 18 *inches*. This changed the design leg to sections 1-3 and caused the excess pressure to be 0.80 *inches wg*, but in section 2. The fan inlet total pressure requirement increased to 8.22 *inches wg*, which is more than the original design. For this iteration, the straight-through loss coefficient, C_s, increased to 0.17 while the branch coefficient, C_b, decreased -0.22. A summary of these results is given in **Table 6-2c.**

Because the third iteration changed the design leg (and increased the system's pressure requirements), we should revert back to the second iteration where a 19 *inches* diameter duct produced the best balancing while not increasing the operation cost. The duct system cost would also be reduced, as 80 *feet* of 19-*inch* ductwork would cost less than 80 *feet* of 21-*inch* ductwork.

Table 6.1
Redesign of Sample Problem 5-4, Increased Airflow in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size (inches)	Volume Flow Rate (cfm)	Velocity (fpm)	VP (in wg)	Loss Coeff.	DTP (in wg)	DSP (in wg)	DP (in wg)	DP (in wg)	Loss Coeff.	DTP (in wg)	DSP (in wg)	Loss Coeff.	DP (in wg)	DTP (in wg)	DSP (in wg)	DTP (in wg)	DSP (in wg)	DTP (in wg)
1	21	11242	4674	1.36	0.25	0.34	1.70	0.94	n/a	-0.06	-0.08	-0.25			1.20	2.39	7.46	8.66	0.96
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	0.11	0.12	0.23			2.16	3.35	8.42	9.62	n/a
3	24	13742	4374	1.19	n/a	n/a	n/a	0.69	n/a				4.67	5.57	6.26	6.26	6.26	6.26	n/a

Table 6.2a - Iteration 1
Redesign of Sample Problem 5-4, Decreased Size in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size (inches)	Volume Flow Rate (cfm)	Velocity (fpm)	VP (in wg)	Loss Coeff.	DTP (in wg)	DSP (in wg)	DP (in wg)	DP (in wg)	Loss Coeff.	DTP (in wg)	DSP (in wg)	Loss Coeff.	DP (in wg)	DTP (in wg)	DSP (in wg)	DTP (in wg)	DSP (in wg)	(in wg)
1	20	10500	4813	1.44	0.25	0.36	1.81	1.01	n/a	0.01	0.01	-0.36			1.39	2.45	7.01	8.07	0.71
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	0.05	0.05	0.04			2.09	3.16	7.71	8.78	n/a
3	24	13000	4138	1.07	n/a	n/a	n/a	0.62	n/a				4.67	5.00	5.62	5.62	5.62	5.62	n/a

Table 6.2b - Iteration 2
Redesign of Sample Problem 5-4, Decreased Size in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size	Volume Flow Rate	Velocity	VP	Loss	ΔTP	ΔSP	ΔP	ΔP	Loss	ΔTP	ΔSP	Loss	ΔP	ΔTP	ΔSP	ΔTP	ΔSP	
	(inches)	*(cfm)*	*(fpm)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*
1	19	10500	5333	1.77	0.25	0.36	2.22	1.29	n/a	0.07	0.12	-0.58			1.86	2.93	7.48	8.55	0.13
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	-0.05	-0.05	-0.07			1.98	3.05	7.60	8.67	n/a
3	24	13000	4138	1.07	n/a	n/a	n/a	0.62	n/a				4.67	5.00	5.62	5.62	5.62	5.62	n/a

Table 6.2c - Iteration 3
Redesign of Sample Problem 5-4, Decreased Size in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size	Volume Flow Rate	Velocity	VP	Loss	ΔTP	ΔSP	ΔP	ΔP	Loss	ΔTP	ΔSP	Loss	ΔP	ΔTP	ΔSP	ΔTP	ΔSP	
	(inches)	*(cfm)*	*(fpm)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*
1	18	10500	5942	2.20	0.25	0.55	2.75	1.67	n/a	0.17	0.37	-0.76			2.60	3.66	8.22	9.28	n/a
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	-0.22	-0.24	-0.25			1.80	2.87	7.42	8.49	0.80
3	24	13000	4138	1.07	n/a	n/a	n/a	0.62	n/a				4.67	5.00	5.62	5.62	5.62	5.62	n/a

6.3 Specifying and Selecting a Fan

The information needed to specify and select a fan was shown in **Sample Problem 5-4**. Fan manufacturers catalog various information about the performance of their fans. This performance is based on the fan laws and tests run by the manufacturer. Fan data from the manufacturer is normally in terms of cfm and static pressure; however, some manufacturers use total pressure in their catalog. For more information about fans, fan testing, and performance, see AMCA publications (**Appendix A.9.6**).

From **Sample Problem 5-4**, the required fan total pressure is:

$$Fan\ TP = \mathrm{TP_{out}} - \mathrm{TP_{in}} = 1.07 - (-7.78) = \underline{8.85\ inches\ wg}$$

From a fan manufacturer's data at standard conditions, the fan needs to be sized for 8.85 *inches wg* total pressure at a volume airflow rate of 13,000 *cfm*, the required fan static pressure is:

$$Fan\ SP = SP_{\mathrm{out}} - SP_{\mathrm{in}} - VP_{\mathrm{in}} = 0 - (-8.85) - 1.07 = \underline{7.78\ inches\ wg}$$

From a fan manufacturer's data at standard conditions, the fan needs to be sized for 7.78 *inches wg* static pressure at 13,000 *cfm*.

Corrections for nonstandard conditions are expressed by the following:

$$Fan\ TP_{actual} = Fan\ TP \times Density\ correction\ factor$$
$$Fan\ SP_{actual} = Fan\ SP \times Density\ correction\ factor$$

For the conditions of **Sample Problem 5-4**, a nonstandard temperature of 400EF, barometric pressure of 24.90 inches Hg, and an elevation of 5,000 feet above sea level, the density correction factor from **Appendix A.1.5** is 0.51.

$$Fan\ TP_{actual} = 8.85(0.51) = \underline{4.51\ inches\ wg}$$
$$Fan\ SP_{actual} = 7.78(0.51) = \underline{3.97\ inches\ wg}$$

In summary, for sizing purposes, a fan capable of providing 13,000 cfm at 7.78 inch wg static pressure for standard conditions is needed for the system in **Sample Problem 5-4**. When the temperature is 400EF, the pressure is 24.90 inch Hg, and the elevation is 5,000 feet, the system will actually operate at a static pressure of 3.97 *inches wg* with the specified fan.

Balancing the system with dampers should not change the fan requirements as a damper in section 1 would be adjusted so the pressure requirements for the section 1-3 path are the same as those for the section 1-2 path, which is the design leg. Balancing the system by increasing airflow in one of the branches could significantly increase the cost (both first and operating) of the fan as both the airflow and fan inlet total pressure requirements increase. Using smaller sizes of duct to increase pressure loss in nondesign legs to balance the system, as was shown in **Sample Problem 6-2**, should have minimal affect on the fan requirements. For this system design the fan total pressure requirements would be:

$$Fan\ TP = \mathrm{TP_{out}} - \mathrm{TP_{in}} = 1.07 - (-7.60) = \underline{8.67\ inches\ wg}$$
$$Fan\ SP = SP_{\mathrm{out}} - SP_{\mathrm{in}} - VP_{\mathrm{in}} = 0 - (-8.67) - 1.07 = \underline{7.60\ inches\ wg}$$

Correcting for nonstandard conditions results in:

$$Fan\ TP_{actual} = 8.67(0.51) = \underline{4.42\ inches\ wg}$$
$$Fan\ SP_{actual} = 7.60(0.51) = \underline{3.88\ inches\ wg}$$

Therefore the same fan would probably be selected as in the unbalanced system design.

Investigate other considerations concerning fans, such as fan orientation and system effect prior to the selection. In exhaust systems, temperature, corrosion, erosion, and expansion may also affect system design and fan selection. Consult fan manufacturers for more specific application and selection information.

6.4 System Considerations

When analyzing exhaust systems, there are special considerations that are not found in systems such as those serving office buildings or commercial shopping centers. With the high temperatures, corrosive atmospheres, erosion, and various materials being transported, an industrial duct system must be able to resist those influences. Duct system performance from the standpoint of structural integrity, economic fitting selection, and available duct materials are just a few of the items for consideration. Reinforcement recommendations for spiral duct are located in **Appendix A.2.3**.