

C.1. Introduction

The information in this appendix illustrates calculation techniques that can be used to evaluate the energy and resource savings associated with system improvements. The techniques illustrated provide relatively fast, conservative savings calculations. However, the results of these calculations will only be as good as the inputs and assumptions made in performing them. Thus, PIER, LBNL, and PECI assume no responsibility for how they are applied beyond the teaching purposes of this guide. The information in this Appendix supports practical use of the calculations during the commissioning process.

These calculations can be used in the following ways:

- To quickly evaluate the potential for savings to guide the commissioning or retro-commissioning process.
- To give the Owner cost savings information regarding a particular modification.
- To provide useful information for troubleshooting purposes.
- As the basis for more detailed analysis where more exact results are required.

It is always good to understand the degree of certainty associated with the technique you are using and the factors that can affect that degree of certainty. The examples of each calculation include discussions of the limitations of the calculation technique and the assumptions underlying the equations that are used. These issues should be carefully assessed in light of your project's specific needs.

The calculations will tend to be focused on one component of a system, but the result can often be extrapolated to other similar components in the project. What seem to be small savings for one component can quickly expand to large savings in the context of entire project. The case study in Section [C.2.7](#) provides an example for savings from static pressure reduction.

Most energy savings that are achieved will persist for the life of the system if properly supported by training and ongoing monitoring. Unless there are significant changes in the way the system is used and operated, the potential to achieve the savings predicted after the initial implementation will persist year after year.

In most cases, the savings can be leveraged to pay for the work necessary to make the improvement. This is often presented in terms of simple payback; if you invest \$4,500 in a measure that will save \$1,000 per year, then the simple payback is the implementation cost of \$4,500 divided by the annual savings of \$1,000 or 4.5 years. This approach is a straightforward way to make a conservation decision. However, it probably understates the financial benefits associated with the avoided costs, like the interest that could be earned on the money that is now available for investment elsewhere. Utility rate fluctuations are also not addressed by simple payback calculations, but are real operating costs.

Economic calculations such as a net present value or return on investment (ROI) are examples of techniques that can be employed to help make more informed decisions. While a detailed discussion of these topics is beyond our scope, applying more complex cost-benefit evaluation techniques may be warranted depending on the proposed modification.

C.2. Fan Energy Savings Due to Static Pressure Reduction

C.2.1. Overview

Reducing the static pressure that a fan must overcome will reduce the horsepower required by the fan to move a given volume of air. Over time, this reduction in horsepower results in energy savings. Determining the magnitude of the energy savings available through a reduction in static pressure can provide valuable insight for making changes such as a revised duct fitting during the design phase. If the duct geometry was already installed, knowing the energy savings that could be achieved by an improved geometry will allow the cost effectiveness of a variety of options to be evaluated including:

- Removal and replacement of the fitting.
- Reduction in system supply air temperature to reduce the air flow requirement through the fitting, thereby yielding a net energy savings (trading refrigeration energy for fan energy).
- Shifting load to a different system to reduce the flow requirements through the fitting.

Calculations can also be used as a troubleshooting tool. For example, the balancer's data may reveal that the fan system will not be able to achieve design capacity without a larger motor and electrical service. In this case, strategies could be used to reduce the static pressure so that the existing motor can handle the load.

Reducing static pressure has immediate and long-term benefits. In the immediate time frame, it may reveal a less costly way to achieve the design intent than what would be associated with a motor replacement, which could get into problems with the electrical service to the motor and the speed and pressure limits associated with the fan class. In the long term, the system will use less energy to achieve the design intent.

To many, it may just seem easier to put in the larger motor. But hopefully, after reviewing the information contained in this appendix, it will become clear that the calculations and effort involved in a little bit of analysis are not complex and can yield significant benefits in terms of first cost and ongoing operating cost savings.

C.2.2. Equations

The equations that will be discussed determine the fan energy by evaluating the mass flow rate in the system (derived from the cubic feet per minute of air that the system is moving) and the kinetic energy imparted to that mass flow rate (derived from the system static pressure).

There are two main steps in the evaluation. The first step is the calculation of the horsepower savings. The second step is calculating energy savings due to this fan horsepower reduction. In some cases, calculating fan horsepower reduction is the only step necessary because it may provide enough information for a decision all by itself. For example, if you wish to understand if a pressure reduction will bring the system's horsepower requirements back into the range of a motor's capability, then the energy savings will not be as important as the potential for reducing horsepower.

C.2.2.1. Horsepower

The equation used to calculate the horsepower savings associated with a reduction in static pressure is as follows. Inputs for the variables associated with the calculation can be obtained from a variety of sources.

Equation C.1: Horsepower Associated with a Reduction in Static Pressure

$$\text{Horsepower} = \left(\frac{Q \times P_{\text{saved}}}{6,356 \times \eta_{\text{FanStatic}} \times \eta_{\text{Motor}}} \right)$$

Where :

Horsepower = The horsepower saved at the terminals of the motor (includes motor efficiency losses)

Q = The flow rate which is experiencing the reduction in static pressure in cubic feet per minute (cfm)

P_{saved} = The static pressure reduction associated with the improvement in inches water column (in.w.c.)

6,356 = A units conversion constant

$\eta_{\text{FanStatic}}$ = The fan static efficiency

η_{Motor} = The motor efficiency

Determining Flow

The result generated from the equation will only be as good as the information used to generate it. Flow can be obtained from a field measurement, the testing and balancing report, the equipment submittal, or information on the construction documents. For the current project conditions, a field measurement is often the best input. But it is often difficult to obtain this information if the commissioning provider does not have the equipment or time to perform a traverse of the system. The following discussion focuses on various ways to determine air flow.

Balance Reports: Information from the balance report can often be used and adjusted based on more easily observed parameters like a coil pressure drop, motor amperage, or speed. For example, if the balance report indicated that when the system was balanced at the design flow of 10,000 cfm, the pressure drop through the wet cooling coil was 1.25 in.w.c., then you might check the cooling coil pressure drop at the current operating condition and see if it is near what the balance report documents. If it is, then the system is probably moving about the same amount of air as it was when it was balanced.

If the cooling coil pressure drop does not match the balance report value, then you can use the balance report data to project the current flow rate based on the fan laws.^[1] Some caution must be used with using the fan laws because they only apply to a system in one configuration.

- If someone repositioned a major balancing damper on a constant volume system after it was balanced, that changes the system curve and the fan law relationships cannot be used to project the fan's performance in the new configuration. If the change was relatively minor, then you can probably assume you would be "close enough" but you should view the result with a little less confidence than if you knew that the system was in the exact same configuration as the day it was tested.
- Filter loading can also change the system curve.
- A VAV fan system is constantly changing its system curve. To use the fan laws to project the system's performance based on the balance report data, you would have to return the system to the configuration it was in when it was balanced.

Coil Pressure Drop: The coil pressure drop itself can be used to project the flow rate since pressure drop across it will correlate with the flow. The condition of the cooling coil can have an impact on the pressure drop. The most common modifier is if the cooling coil was wet when the reading was taken because it was dehumidifying. If it was dry, then the pressure drop at a given flow rate will be less than when it was wet. Equipment shop drawings will often document both the wet and dry pressure drop for the cooling coil. The cleanliness of any coil will also affect its pressure drop. If filter maintenance has been good, then it is fairly likely that the coil cleanliness has not changed much from when it was first installed, but a visual inspection will quickly verify this.

You could also take a pressure drop across some other component, like a heating coil, as a cross-check. Most heating coils are shallower than cooling coils and therefore, have less pressure drop to measure. But, they can provide an easily obtained cross check against the information from the cooling coil.

Motor amperage: As an indicator of flow, motor amperage should be used with caution since it is not a direct indicator of horsepower due to the variation in efficiency and power factor associated with different motor load conditions (See [Figure C.1](#) for an example). Motor kW is a little better since it eliminates one of these variables, but it is a more complicated field measurement to make since it requires a kW meter with volt and amperage inputs from several phases compared to a simple clamp-on amperage reading. In addition, the fan could be using the same horsepower as when it was balanced but moving a different amount of air if the fan speed was adjusted to accommodate a system modification for which there is not a balancing record. The fan speed could also have been inadvertently modified during a belt change if the fan or motor are equipped with a variable pitch sheave.

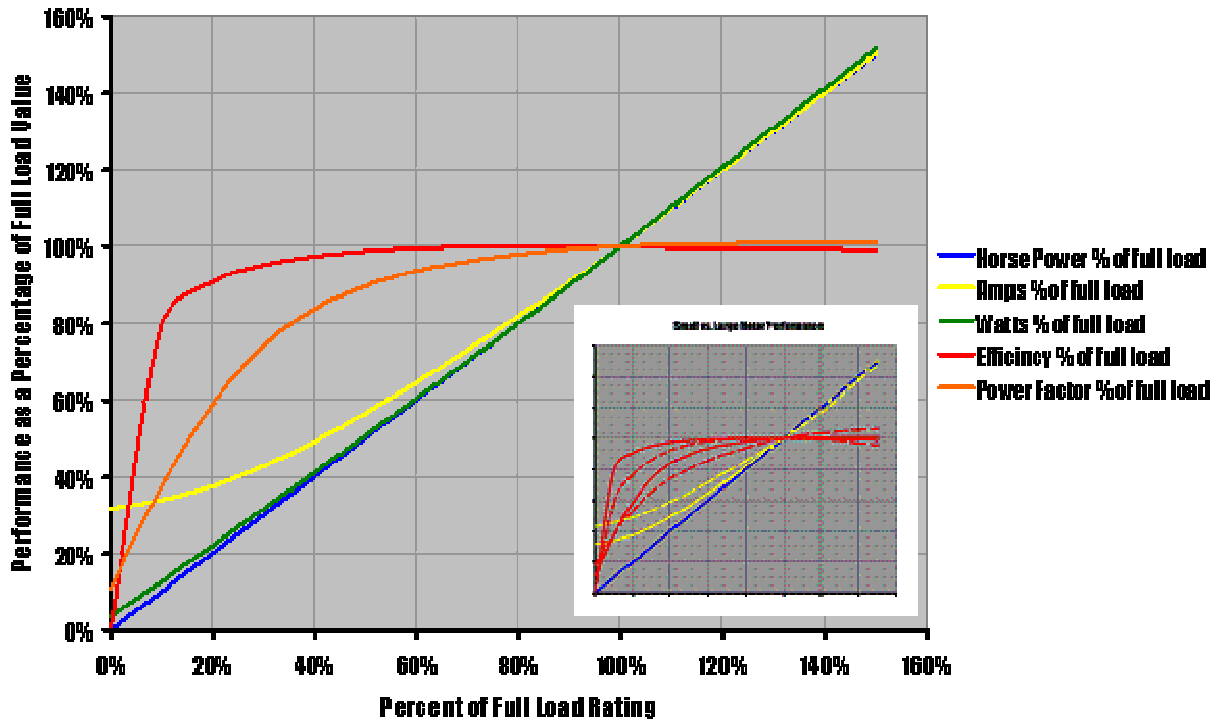
Filter Pressure Drop: Although filter pressure drops are easy to obtain using the indicators typically provided, they generally are bad indicators of flow because the filter pressure drop can vary drastically both with loading and with make and model.^[2] Even if the balance report says “clean filters” and you are looking at clean filters, a bit of skepticism is advisable unless you know that the filters you are looking at are the exact make and model that was in the system when the balancer did their measurements.

Fan Static Pressure: Fan static pressure is also a less desirable measure of performance than some of the other indicators. True fan static pressure is difficult to measure in the field for a variety of reasons including the need to correct for the velocity pressure entering the fan and the turbulent conditions that exist in the fan discharge. In addition, the fan can produce the same static pressure at a different flow rate if its speed has been changed.

Figure C.1: Motor amperage, efficiency, power factor, and kW as a function of the actual shaft load for a 25 horsepower energy efficient motor

Notice how the motor amps and watts do not follow the motor load as the power factor and efficiency drop off towards the low load conditions. Amperage is more misleading than watts as an indicator of load. These effects also vary with motor size. Generally, smaller motors exhibit poorer part load performance and correlation between amps and actual shaft load, with some small motors drawing 40-50% or more of their rated amps with no load on the shaft. The inset graph compares a 1 hp (dashed lines) and a 25 hp (solid lines) motor. (Note that the Y axis is in terms of percent of full load value to allow multiple parameters to be compared on the same axis. This means an efficiency or power factor point of 80% means 80% of the full load efficiency or power factor value for the motor, not 80% efficiency or power factor.)

Three Phase 25 Horse Power Premium Efficiency Motor Performance



Determining Static Pressure

Static pressure can be obtained by direct measurement in situations where you are trying to understand the energy penalty associated with a loss. Static pressure savings for different equipment configurations (for example, airfoil blade vs. flat plate blade dampers) can be determined from manufacturer supplied performance data. Static pressure savings for different fitting geometries can be estimated by calculating the loss for each fitting design based on information available from ASHRAE, SMACNA or manufacturers specializing in prefabricated fittings.

The Units Conversion Constant

The units conversion constant is simply a factor used to allow the result to be expressed in terms of horsepower based on the inputs of flow rate, head, and efficiency. If you examine the numerator of the equation, you will notice that if you multiply the volumetric flow rate (cubic feet per minute) by the appropriate factors (like density), you can convert it to mass flow rate (pounds per hour). You will also notice that, by using the appropriate multiplication factors you could convert the pressure or head in terms of inches water column to feet water column. If you multiply pounds per hour by feet, you get foot-pounds per hour, which can be converted to horsepower by the appropriate conversion factor. The units conversion constant 6356 is simply what you get when you multiply all of the conversion constants together (0.0765 pound of air per cubic foot of air times 1 foot per 12 inches times 60 minutes per hour, etc., etc., etc.).

In this case, the conversion constant includes a factor for converting cubic feet of air to pounds of air based on air at standard conditions. That means that when we calculate something with this equation and the air is at anything other than standard conditions then the result will not be exactly correct because

temperature, barometric pressure, and humidity content will affect the density of the air. In this particular case, the equation will provide reasonably accurate results for any of the temperature and pressure conditions typically encountered in HVAC systems at elevations between sea level and three or four thousand feet. For higher elevations and higher air temperatures, it will be necessary to either adjust the units conversion factor or view the result with a slightly more conservative eye.

Fan Efficiency

Fan efficiency is a dimensionless variable that takes into account the mechanical efficiency with which the fan wheel is able to take air and move it to a higher pressure level. Not all of the energy that goes into the fan shaft will end up in the air stream. Things that work against efficiency include bearing losses, aerodynamic losses and friction through the fan, leakage from the high pressure side back to the low pressure side, and the efficiency of the velocity to pressure and pressure to velocity conversions that occur as the air moves through the fan. Larger fans will generally be more efficient than smaller fans for any particular type of fan. Efficiencies can also vary with design (airfoil centrifugal, vane axial, propeller) as well as the operating point on the fan's curve and manufacturer. Table 1 of *Chapter 18* of the *2000 ASHRAE Systems and Equipment Handbook* provides a good comparison of the typical efficiency and other performance characteristics of various fan designs. This table can be used as a source for the efficiency term in our equation if specific information is not available from the manufacturer. As a general rule of thumb, efficiencies can vary from around 55% for a propeller or small centrifugal fan up to about 85% for a large double width, double inlet airfoil centrifugal fan.

Motor Efficiency

Motor efficiency is also a constant that takes into account the losses associated with converting electrical energy into rotating shaft energy, the function of the motor. Things working against motor efficiency are losses due to the resistance of the copper to electrical current flow, imperfections in the iron and other magnetic elements that result in imperfections in the magnetic field, fan losses associated with the inefficiencies of the cooling fan, aerodynamic effects associated with the rotor spinning in air, and bearing losses. As can be seen from [Figure C.1](#), efficiency will vary with motor size and motor load. It will also vary by manufacturer as a function of motor construction and the quality of the materials used.

Horsepower

Horsepower (the result of the calculation) can be converted to kilowatts (kW) for calculating the electrical energy savings, as will be illustrated in the next section. Horsepower and kW are power terms; i.e. the rate of doing work. Horsepower-hours and kilowatt-hours (kWh) are energy terms.

C.2.2.2. Energy

Converting the horsepower obtained from [Equation C.1](#) into annual energy savings is a fairly straightforward process. Horsepower is an expression of a rate of performing work; i.e. the rate at which energy is being used. If you multiply the rate at which energy is being used by the time over which the use occurs, the result is the energy consumption.

In our case, we would like to know the savings in terms of electrical energy and electrical cost savings, so it is desirable to turn the horsepower term into kW, a simple units conversion. If we choose the number of hours per year that the system operates as the period of which the savings occurs, we will obtain the annual energy savings for the proposed modification. Multiplying this accumulated annual energy savings by the current electrical rate will provide the annual dollars saved. These calculations are illustrated in [Equation C.2](#).

Equation C.2: A, B, and C - Projecting Energy Savings from a Horsepower Reduction

$$kW = \text{Horsepower} \times 0.746 \left(\frac{\text{kilowatts}}{\text{horsepower}} \right) \text{(A)}$$

$$kWh_{\text{Annual}} = kW \times \text{Hours}_{\text{Annual}} \text{(B)}$$

$$\text{\$}_{\text{Annual}} = kWh_{\text{Annual}} \times \text{Utility Electrical Rate} \text{(C)}$$

Where :

kW = The kW saved at the terminals of the motor (includes motor efficiency losses)

Hours_{Annual} = The annual number of operating hours for the system in which the change will be made

0.746 = A units conversion constant

kWh_{Annual} = The annual electrical energy savings

\\$_{Annual} = Annual cost savings

Utility Electrical Rate = The utilities net price for electrical energy in terms of \$ per kWh

C.2.2.3. Static Pressure at Different Flow Conditions

For constant volume systems, [Equation C.1](#) and [Equation C.2](#) will allow the savings associated with a static pressure reduction to be estimated with a reasonable degree of accuracy. For variable air volume systems, the problem becomes a bit more complicated. As the fan volume is changed, all of the terms in [Equation C.1](#) will vary with the flow variation and load variations associated with it. Thus, an exact solution would require the evaluation of equations [Equation C.1](#) and [Equation C.2](#) at each incremental flow condition. This is a significant undertaking that either requires a lot of time for manual calculations or developing a model of the system in a computer based building simulation program.

Since we are simply trying to understand the approximate savings and we need to make our evaluation fairly quickly, we need to come up with a technique that will take the variation in the variables into account but not require extensive calculations. Towards that end, the following paragraphs will examine each of the variables in [Equation C.1](#) in light of what occurs to them as the flow in the system varies.

- **Fan efficiency** will vary with load, as can be seen from the information presented in the ASHRAE Systems Manual reference. But, if the flow variation is accomplished via a reduction in speed, as is the case with most current technology systems, the variation in this parameter as the load drops off will be relatively minor, especially if the system only turns down to 40% or 50% of peak flow.^[3]
- **Motor efficiency** will also vary with load as can be seen from the information presented in [Figure C.1](#). But, we can also see from the figure that the variation is relatively minor until the load on the motor drops below 40 or 50%. Since the power required by a fan varies as the cube of the flow rate^[4], a 50% flow reduction results in the fan only requiring 12.5% of the horsepower it required at full flow if nothing else was changed^[5]. As a result, the decaying motor efficiency is applied to a relatively small power level and the impact is still fairly insignificant in the big picture unless the system spends a lot of time at the very low load condition. This cubic relationship is not exactly true for a VAV system since it is actually operating on a set of system curves created by the modulation of the terminal units and the fan laws can only be applied to a fixed system. Systems that vary their speed to reduce flow tend to mimic the effect of following a fixed system curve and will achieve a result very close to that predicted by simply assuming a fixed system curve and that the fan affinity laws apply. For systems that use other approaches like inlet vanes, blade pitch variation, or simply pushing the fan up the curve, the fixed system curve assumption is not as good, but it is a valid indicator of a trend that is useful in evaluating what will happen as the load on the system drops.

- **Static pressure** drops through various components in the system will vary roughly with the square of the flow. [Equation C.3](#) illustrates this. In reality, the exponent in the equation is not exactly “2” but for the purposes of most HVAC calculations, rounding it to “2” will provide satisfactory results.

Thus, if the flow rate is reduced by 50%, the pressure drop through any given component that experiences the flow reduction will be reduced to 25% of the value associated with the full flow condition. The square law relationship makes this a fairly powerful effect in terms of our evaluation since our calculation is targeted at evaluating energy savings associated with a static pressure reduction. Thus, we probably need to consider the impact of this relationship on our result, especially if the system spends a significant number of hours at a reduced flow condition. Otherwise we could overestimate the energy savings. Note that this effect does not impact the projected *horsepower* reduction at design flow. It only affects the accumulated energy savings achieved by this horsepower reduction over time.

Equation C.3: The Square Law

$$P_{New} = P_{Old} \times \left(\frac{Q_{New}}{Q_{Old}} \right)^2$$

Where :

P_{New} = The static pressure at the new flow condition to be evaluated.

P_{Old} = The static pressure at the old or original flow condition, usually the design condition.

Q_{New} = The flow at the new flow condition to be evaluated.

Q_{Old} = The flow at the old or original flow condition, usually the design condition.

There are a variety of ways that reduced static pressure can be taken into account in a simplified calculation scenario.

- 1 One of the easier approaches is to assume that the net affect of the flow variation is as if the system operated continuously at some reduced percentage of its design capacity. For instance, the average fan flow could be derived from the fan’s daily load profile. The static pressure requirement at this reduced percentage can be evaluated using the square law. Then the energy savings calculation can be done as if the system were a constant volume system that saw this modified pressure loss continuously.
- 2 A variation on option 1 is to bracket the savings by evaluating the fan system operation at two different reduced percentages of its design capacity. For instance, you might reflect back on your field experience as a commissioning agent and realize that many of the drives on the office building VAV systems that you have worked on seem to spend a lot of time running at 65 to 80% of maximum speed. So, you could decide to evaluate the option you are proposing at two points; one as if the fan ran continuously at 65% speed, and one as if the fan ran continuously at 80% speed and assume that the truth lies somewhere in between.
- 3 A more rigorous approach is to evaluate the fan at a variety of load conditions based on a load profile of some sort. In other words, if the load profile indicates that the flow will vary from a maximum of 100% to a minimum of 40%, you would evaluate the performance in even increments between 40 and 100%, perhaps every 10%. This would involve the following steps for each increment:
 - Calculate the static pressure at the reduced flow rate based on the square law using [Equation C.3](#).

- Calculate the horsepower at the reduced static pressure and flow rate based on [Equation C.1](#).
- Calculate the energy required at the reduced flow rate and static pressure based on the results of the horsepower calculation and the hours per year that the fan spends at that condition based on the daily load profile and the number of days per year. [Equation C.2 B](#) and C would be used.
- Add up the total energy savings and multiply it by the electrical rate to determine the annual cost savings. A spreadsheet makes this process less arduous than it sounds, but it will take some time and is only as good as the accuracy of the inputs. If you are making a lot of assumptions, then you may be better off using one of the less rigorous techniques and simply being more conservative in how you use the result in your recommendations.

The discussion of the factors that change in [Equation C.1](#) for VAV systems made reference to the system's load profile. Determining the actual load profile for the system would, at first glance, seem to be one of the more complex and difficult tasks. However, there is actually quite a bit of information available in this regard for most common HVAC applications. In addition, the project engineering records, and perhaps even trend data you have collected as a commissioning provider on other projects will contain information that can help you develop load profile information. In addition, many projects are required to document their load calculations or energy consumption calculations in the information submitted for review by approval authorities.

Even if this information is not documented, the engineer of record should have some form of a load calculation in their files for the project. Many firms use computer programs to calculate loads and energy consumption patterns, as seen in [Figure C.2](#). The flow values were developed by solving the sensible heat equation ($\text{Cooling Load in Btu/hr} = 1.08 \times \text{Flow Rate in cfm} \times \text{Difference between the supply temperature and space temperature in degrees F}$) for flow using the space load and typical supply and space temperatures. This spreadsheet is included in the link below:



[Link to go to a spreadsheet that will help you create a load profile.](#)

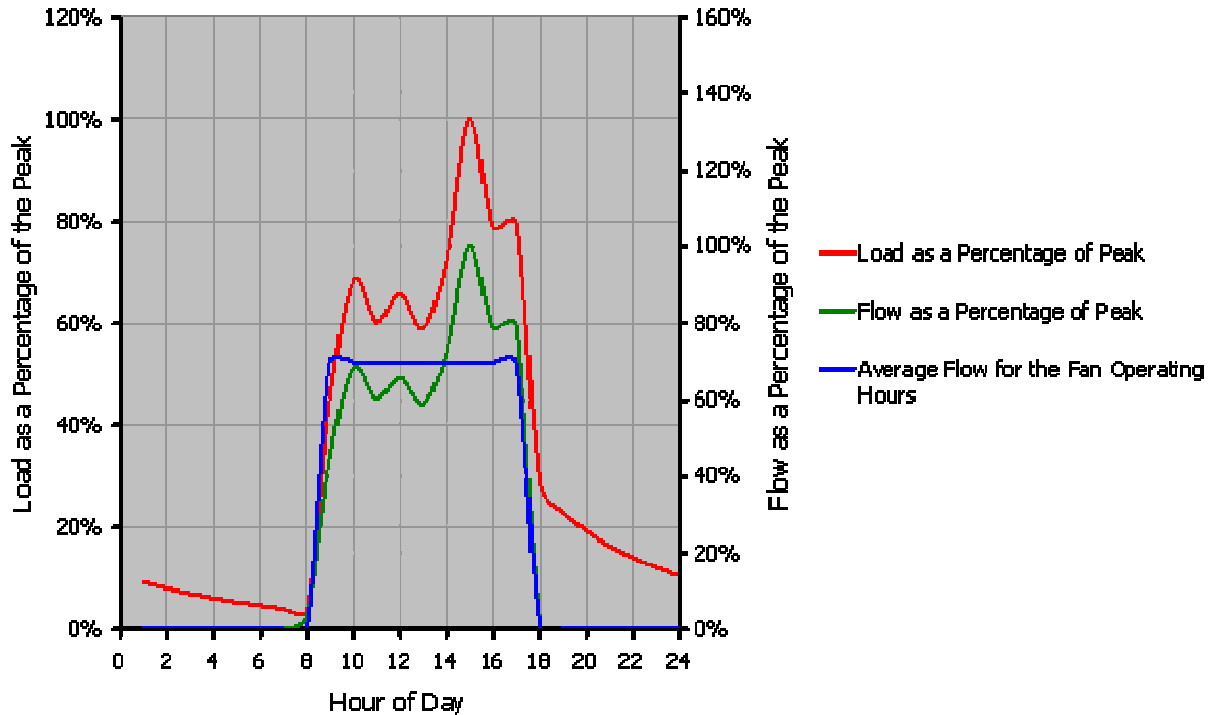
This sort of profile information can also be obtained from the *ASHRAE Handbook of Applications*. The chapter on *Commercial and Public Buildings* contains a table with load profiles, typical design criteria, filtration requirements, and other useful information for a variety of common buildings. Similar information can often be derived for less common building types based on the information presented in other chapters.

For the purposes of the static pressure reduction energy savings evaluation, the exact magnitude is not as important as the shape of the curve and average values. Expressing the information in terms of percentage of the peak value provides a frame of reference that can be easily transferred from project to project.

Figure C.2: Typical Interior Office Area Load Profile

The shape of this profile is fairly typical for a system serving interior loads in an office area with 9 to 5 occupancy. The peak will vary with the actual loads and minor variations in the shape will occur as a function of work or break schedules. In this example the average flow rate for the occupied hours is 80% of the maximum. Experience has shown that the average flow value for this type of occupancy will vary from 65% to 85% of the maximum.

Typical Interior Office Air Handling System Load Profile



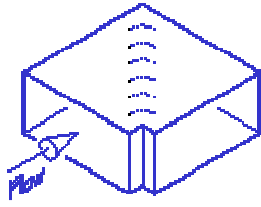
C.2.2.4. Duct Fitting Pressure Losses

Subtle differences in the way a duct fitting is fabricated can make significant differences in the pressure losses associated with the fitting. For example, the *ASHRAE Duct Fitting Loss Coefficient Tables* documents 5 different turning vane designs with loss coefficients that vary from a low of 0.11 to a high of 0.43. Many people would be surprised to learn that there even were 5 different turning vane designs let alone that the pressure drops associated with them could vary by a factor of 4. The physical differences between the different designs are minor details like the length of the trailing edge and the spacing and thickness of the vanes.

There are two equations associated with evaluating the pressure loss through a duct fitting. The first is used to evaluate the loss as shown in [Equation C.4](#).

Equation C.4: Duct Fitting Pressure Loss

This equation states that the loss through a fitting is a function of some experimentally determined loss coefficient and the velocity pressure associated with the velocity of the air flowing through the fitting. Since the loss is expressed in terms of velocity pressure, [Equation C.5](#) is used to convert the duct velocity to its corresponding velocity pressure.



$$\Delta p_{\text{fitting}} = C_o P_{\text{velocity}}$$

Where :

$\Delta p_{\text{fitting}}$ = Fitting pressure loss in inches water column

C_o = Local loss coefficient

P_{velocity} = Velocity pressure in inches water column, usually based on entering velocity

Equation C.5: Velocity Pressure as a Function of Velocity

$$V = 4,005 \sqrt{P_{\text{velocity}}}$$

Therefore :

$$P_{\text{velocity}} = \left(\frac{V}{4,005} \right)^2$$

V = Velocity in feet per minute

4,005 = A units conversion constant

P_{velocity} = Velocity pressure in inches water column

Notice that the velocity pressure is a function of the square of the velocity. That makes it a powerful relationship in HVAC systems. If you double the velocity through a fitting (i.e. you double the flow since flow and velocity are directly related), you will increase the pressure loss through the fitting by a factor of four. The magnitude of the loss will be a function of the loss coefficient, with a smaller coefficient being better (lower loss, lower energy, better efficiency) than a large one.

Fitting loss coefficients are available from a variety of sources. One of the most extensive compilations is contained in the *ASHRAE Duct Fitting Loss Coefficient Tables*, available from the ASHRAE bookstore. Many of the fittings in the Tables are also included in the *ASHRAE Handbook of Fundamentals*. Using these equations to evaluate the loss through a fitting is demonstrated in a subsequent example.

C.2.3. Additional Savings

When a static pressure reduction is implemented, there are other savings that occur in addition to the static pressure reduction savings. In most instances, the changes will be relatively minor and will simply add a safety factor to the projected savings. Knowing these additional savings can be useful when they may become significant, either due to the size of the system or the nature of the energy source or process.

C.2.3.1. Fan Heat Reduction Savings

When you reduce the fan energy that goes into a system, you reduce the fan heat that goes into the system (see the side bar if you are not familiar with fan heat). When a static pressure reduction reduces the fan energy it also reduces the fan heat, which reduces the cooling load and saves energy.

If you reduce the cooling load on the system, then you reduce the cooling load on the central plant, which saves both cooling energy and the energy to power the auxiliary systems that are associated with the plant. If the plant is simply an air-cooled condensing unit, then the condensing fans will probably run a little less. If the plant is a central chilled water plant, then distribution pumping cooling tower fan energy will be reduced.

Fan Heat:

Because the fan is doing work on the air, some of the work shows up as a temperature rise across the fan. You may have noticed how the tire pump gets warm when you pump up your bicycle tire, which is the same phenomenon. If the fan system is developing any static pressure, you can measure this temperature rise in the field with a thermometer capable of detecting half degree changes (use the same thermometer to take the inlet and outlet temperature readings so that calibration errors between thermometers do not mask the effect). If you know the flow rate in the system, you will discover that if you plug the fan heat into the sensible heat rise equation (sensible heat = 1.08 times the flow rate times the temperature difference) and convert the result to horsepower, it will correlate well with the fan's brake horsepower requirement for the current operating condition. This temperature rise will occur in a system with the motor out of the air stream. If the motor is in the air stream, then the motor efficiency losses also contribute to the temperature rise that you are measuring.

If the static pressure reduction results in a measurable reduction in the fan heat, then it will cause a measurable drop in the discharge temperature of the air handling unit, all other things being equal. This can have several effects if no other action is taken.

- For VAV systems, the system flow rate will tend to decrease if nothing else changes because the air has additional capacity to cool due to the lower supply temperature. Depending on what is controlling the system supply temperature (fan discharge or coil discharge) it will probably be desirable to adjust the setpoint of the controller by an amount corresponding to the reduction in discharge temperature associated with the reduction in fan heat.
- For constant volume reheat systems, the reheat load will go up if nothing else changes because of the additional cooling capacity represented by the lower supply temperature. If the system is a true constant volume reheat system operating the reheat coil for the purposes of humidity control, (as is often the case in clean rooms and hospital surgeries) then the reheat energy will go up while the fan energy goes down. This occurs because the fan heat was providing some of the reheat energy. In most cases, there will still be a net improvement in energy use unless the reheat function is being provided by an electric reheat coil, in which case it will probably just balance out the fan energy savings.

In summary,

- 1** If the fan heat was supplying reheat that was necessary anyway, then the net savings associated with the static pressure reduction will be reduced by the additional cost associated with the reheat process.
- 2** If the cooling coil discharge temperature can be raised and still maintain the necessary humidity level, then reducing the fan heat simply eliminates an unnecessary reheat load. In this case, all of the energy savings associated with the fan static pressure reduction will be realized.

For the purposes of our discussion, what matters is to know that these issues exist and to account for them appropriately in the recommendations that come out of the analysis.

C.2.3.2. Resource Savings

In many cases, measures that save energy will also save other resources. For example, when reducing static pressure by changing filter designs from a conventional design to an extended surface area design, additional savings can be realized due to longer change-out cycles (see Chapter 4: Filtration).

- 1** Costs could be reduced because the filters will probably last longer and require replacement less frequently.
- 2** Labor will be conserved because the man-hours required to change filters will be reduced due to the longer change-out cycle.
- 3** Disposal costs may be reduced because there will be fewer filters to break down and haul off to the land fill. In some locations, this will be a direct reduction in the cost of waste disposal for the site. Even if there is no cost savings, reducing waste makes the building more sustainable.

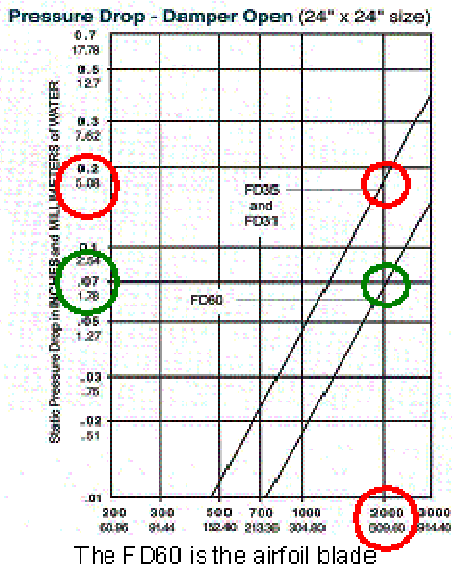
Recognizing other significant benefits besides energy savings is important because they may provoke additional interest and economic incentive to move a project forward.

C.2.4. Example 1: Airfoil Blade Smoke Isolation Dampers

Consider the following situation. You are involved in a design review for a project that is modifying a 20,000 cfm constant volume air handling system in a hospital serving a portion of its laboratory facility. At the completion of the first phase, the modified system will operate for about 10 hours a day, Monday through Friday and for about 5 hours on Saturdays to serve the newly remodeled laboratory area. After about a year, the system will operate 24 hours per day, 7 days per week. In reviewing the documents you notice that the smoke isolation dampers located in the unit's main supply duct have been specified with a conventional flat plate design. Since the velocity through the duct at that particular area is in the 2,000 fpm range, you begin to wonder if there might be a static pressure savings associated with using an airfoil blade design. Consulting the catalog for one of the manufacturer's reveals that a savings in the range of 0.10 inches w.c. can be achieved by switching to the airfoil blades. So you decide to do a calculation to determine what this translates to in terms of horsepower and energy to see if there might be a viable payback for the incremental cost difference associated with using the airfoil blade design instead of the conventional design.

Figure C.3: Comparative Damper Pressure Drops

Performance Data



C.2.4.1. Horsepower Reduction Calculation

The horsepower calculation is accomplished using [Equation C.1](#) as follows:

Assumptions:

- Damper performance information is based on the manufacturers data as shown in [Figure C.3](#).
- Pressure drop through the conventional damper = 0.065 inches w.c. @ 2,000 fpm.
- Pressure drop through the airfoil blade damper = 0.170 inches w.c. @ 2,000 fpm.
- Savings associated with airfoil blades = 0.105 inches w.c.
- Air handling unit performance is based on the value for the modified system as documented on the drawings you are reviewing.
- System flow rate = 20,000 cfm.

You don't have specific information on the fan or motor. But, you know that the existing fan is a Double Width, Double Inlet (DWDI) airfoil type centrifugal fan and decide to assume a nominal efficiency based on past experience and information presented in the *ASHRAE Equipment and Systems Handbook* chapter on fans. You also know that as a part of the retrofit, the existing fan is to be provided with a new, premium efficiency motor, so you assume a motor efficiency based on some catalog information that you have for premium efficiency motors.

Calculation:

$$\text{Horsepower} = \left(\frac{Q \times P_{\text{Saved}}}{6,356 \times \eta_{\text{Fan Static}} \times \eta_{\text{Motor}}} \right)$$

Flow rate = $Q = 20,000$ cfm

Static pressure reduction = $P_{\text{saved}} = 0.11$ in.w.c.

Assumed fan efficiency = $\eta_{\text{fan}} = 75\%$

Assumed motor efficiency = $\eta_{motor} = 89\%$
Fan horse power reduction = **Horsepower** = 0.5 hp

The 0.5 hp savings does not seem like much in terms of magnitude, but you know that it can add up over time. Since this system will run 24 hours per day in a year, you decide to proceed with an energy calculation to see what sort of savings you might achieve both in the immediate future as well as a year out. [Equation C.2](#) converts your horsepower figure to kW to allow you to proceed with the energy calculation.

$$kW = \text{Horsepower} \times 0.746 \left(\frac{\text{kilowatts}}{\text{horsepower}} \right)$$

Power reduction = **Horsepower** = 0.5 hp. as calculated above
 $kW = 0.37 \text{ kW}$

C.2.4.2. Constant Volume System Energy Savings

Since the system is a constant volume system, the energy savings calculation is relatively straightforward. In this particular instance, there are two conditions that we are concerned with; the savings during the first year of operation when the system will run on a schedule, and the savings for subsequent years when the system will begin to run 24 hours per day. [Equation C.2](#) B and C are used for the calculation.

Assumptions:

Constant volume system operation is assumed. The current system operating hours are assumed to be 2,860 hours/year based on information from the facilities engineering department indicating that the lab will be open Monday through Friday from 7am through 5pm and Saturday from 7am through noon. Assuming that the fan will run for all hours of lab operation is reasonable and conservative since it will probably start earlier to ensure that the spaces are comfortable by the time the facility is occupied. It will also run occasionally during the unoccupied hours as necessary to accommodate the night set back and set up functions.

The operating hours after the first year, when more functions are moved to the new lab from other locations are assumed to be 8,760 hours per year; i.e. round the clock operation.

The current electrical rate is assumed to be \$0.090 per kWh. This was obtained from raw utility bill data by taking the most recent months charges and dividing them by the consumption associated with them. The resulting electrical rate includes demand charges as well as energy charges. If the proposed modification was going to significantly alter the building's peak demand, then it might be necessary to evaluate the impact on demand separately from the impact on energy. In our case, the impact on demand will not be significant.

The kW savings per hour at design flow are assumed to be as calculated previously; i.e. 0.37 kW per hour.

Calculation:

$$kWh_{, \text{Annual}} = kW \times \text{Hours}_{\text{Annual}}$$

$$\text{\$}_{\text{Annual}} = kWh_{, \text{Annual}} \times \text{Utility Electrical Rate}$$

Power reduction associated with the reduction in static pressure = $kW = 0.37 \text{ kW}$ as calculated previously.
Operating hours = **Hours** = the assumed hours of operation as discussed above.

Annual energy savings at the current operating schedule = $kWh_{Annual} = 1,062$ kWh.

Annual cost savings at the current operating schedule = $\$_{Annual} = \96 per year.

Annual energy savings at the future operating schedule = $kWh_{Annual} = 3,253$ kWh

Annual cost savings at the future operating schedule = $\$_{Annual} = \293 per year

This calculation is for a 10 square foot damper, nominally 24 inches high by 60 inches wide. Based on pricing from a major manufacturer, its cost is probably in the range of \$800 - \$900 for a conventional design. Going to an airfoil blade design will add 15% to 20% to this cost, or, \$120 to \$180. So, the payback for going to the more efficient damper would be less than two years when the system was operating on a schedule. In a year or so, when the system is placed in a 24 hour per day operating cycle, the payback drops to less than a year.

There are several important points concerning the point at which the change to an airfoil damper is made.

- 1 While not a huge number in the context of the overall utility budget, the savings generated are enough to generate a quick payback if the change is made before the dampers are purchased.
- 2 It is important to catch items like this during design review or shop drawing review, and prior to damper fabrication or installation. If the damper was already fabricated but not installed, then the modification still may be viable in light of the proposed 24 hour per day operation. But, the payback would be much less attractive; probably somewhere in the range of 2 or 3 years because a new damper would have to be purchased and the one that had already been fabricated would be restocked (unlikely) or scrapped (more likely and not particularly sustainable).
- 3 If the damper had been installed, either in new construction or, if you were looking at this problem in a retro-fit application on a retrocommissioning project, then the replacing the damper may be difficult to sell because the savings would need to support the purchase and installation of a new damper, not just the incremental cost difference. The installed cost could easily be 2 to 4 times the damper cost, especially in an existing situation where access may be difficult and require disruption to tenants and removal and replacement of finishes.

In situations where more than the incremental first costs are incurred, the simple payback might stretch out into the 8 to 10 year range. For a long term Owner with an energy conscious attitude, this still might be viable, but would take second place to other projects with better paybacks. As will be illustrated in a subsequent calculation example, there may be other more pressing concerns driving the need to replace the damper. In those instances, the energy savings may simply be a way to make the replacement less painful to swallow.

C.2.4.3. Variable Volume Energy Savings Calculation

The system we were considering was a constant volume system. But, per the discussions in Section [C.2.2.3](#), we know that flow variation could have a significant impact on our results. To illustrate this as well as the technique to evaluate it, assume that the system we have been studying operated as a VAV system serving interior office spaces with a load profile similar to what is depicted in [Figure C.2](#).

In many cases, the first step may be to see what the horsepower savings is at design flow to allow us to understand if any further effort is justified. This is the same calculation we did in our previous example as discussed in Section [C.2.4.1 Horsepower Reduction Calculation](#).

Assuming that we decide that additional investigation is merited, the next step is evaluate the energy savings on a VAV system, which is different than for a constant volume system. The key decision to make in this step is deciding how to reflect the system's load profile in the annual energy savings calculation. A variety of techniques were discussed in Section D.1.2.3: Static Pressure at Different Flow Conditions. For the sake of time, we decide to use Option 1 and see where that leads us.

We will generate a load profile using some information we obtained from the engineer's hour-by-hour load calculations for one of the zones the unit serves, as illustrated in [Figure C.2](#). From this information we know that the average flow rate when the system is running will be about 70% of the peak design flow. So, our first step in the process is to use the square law ([Equation C.3](#)) to evaluate the pressure drop of the damper at the lower flow condition.

Assumptions:

Pressure drop through the conventional damper = 0.065 inches w.c. @ 2,000 fpm

Pressure drop through the airfoil blade damper = 0.170 inches w.c. @ 2,000 fpm

Savings associated with airfoil blades = 0.105 inches w.c.

System flow rate = 20,000 cfm

Average flow rate as a percentage of design = 70%
(from the load profile we generated with the engineer's data)

Flow rate to be used for the evaluation = 14,000 cfm

Calculation:

$$P_{New} = P_{Old} \times \left(\frac{Q_{New}}{Q_{Old}} \right)^2$$

Design flow rate = Q_{Old} = 20,000 cfm

Static pressure reduction at the design flow rate = P_{Old} = 0.11 in.w.c.

Reduced flow rate = Q_{New} = 14,000 cfm

Static pressure reduction at the reduced flow rate = P_{New} = 0.05 in.w.c.

After completing the calculation, we realize that we could have also evaluated this revised pressure loss at the lower flow rate using the damper manufacturers curves at the lower velocity associated with the lower flow rate (1,400 feet per minute vs. 2,000 feet per minute at design). When we look at the damper curves as a cross check, we discover that the values match.

Notice how the static pressure reduction is much lower than the reduction in flow due to the square law. The reduced static pressure saved will have a noticeable impact on our savings compared to a constant volume system.

The actual evaluation of the energy savings is identical to what was used before except the horsepower savings will be based on the reduced static pressure as shown below. Our assumptions with regard to flow and pressure are as indicated in the square law calculation. We will assume the same motor and fan efficiency that we used for the previous constant volume system calculation.

Calculation:

$$Horsepower = \left(\frac{Q \times P_{Saved}}{6,356 \times \eta_{FanStatic} \times \eta_{Motor}} \right)$$

Flow rate = Q = 18,000 cfm

Static pressure reduction = P_{saved} = 0.05 in.w.c.

Assumed fan efficiency = η_{fan} = 75%

Assumed motor efficiency = η_{motor} = 89%

Fan horsepower reduction = **Horsepower** = 0.17 hp. or 0.13 kW.

As expected, this is a much smaller reduction in horsepower than was associated with the constant volume system because it is based on the average flow rate the system will see, not the design flow rate. At design conditions, we will still see the horsepower reduction predicted for the system at design flow. But, since the system does not spend all of its hours operating at that condition, we need to evaluate the energy consumption based on a horsepower that is representative of the average system operating condition. As indicated previously, there are a variety of ways to do this with varying levels of rigor. We simply chose to use one of the simpler approaches to the problem.

To continue the energy savings calculation, we simply use the fan kW we just calculated for the variable volume application in [Equation C.2 B](#) and C for the two operating modes we are interested in (scheduled and unscheduled) as illustrated below.

Calculation:

$$kWh_{Annual} = kW \times Hours_{Annual}$$

$$\$_{Annual} = kWh_{Annual} \times Utility\ Electrical\ Rate$$

Power reduction associated with the reduction in static pressure = $kW = 0.13$ kW as calculated previously.

Operating hours = $Hours$ = the assumed hours of operation as discussed for the constant volume system.

Annual energy savings at the current operating schedule = $kWh_{Annual} = 364$ kWh.

Annual cost savings at the current operating schedule = $\$_{Annual} = \33 per year.

Annual energy savings at the future operating schedule = $kWh_{Annual} = 1,116$ kWh

Annual cost savings at the future operating schedule = $\$_{Annual} = \100 per year

So, while the savings are not nearly as significant as they were for the constant volume system, they still justify the incremental cost associated with going to the higher efficiency dampers *as long as the change can be made prior to fabrication and installation of the dampers*. The lower savings potentials magnify some of the contingencies discussed previously regarding making this change after the dampers are fabricated and installed or in a retrofit application.

C.2.4.4. Extrapolating the Savings

Even though the savings projected by the preceding calculations may appear to be relatively small to some, it is important to remember they were the savings associated with one component in a system. In many cases, there will be other locations in the system where these savings will be duplicated. There may be other similar systems on the project with similar savings opportunities.

For example, in many buildings, it would not be unusual for the duct leaving the air handling unit to turn and run down the corridor outside the equipment room, then, at some point, to exit the corridor to serve a load. When the duct exited the corridor, it would likely penetrate a fire and smoke separation similar to the one it penetrated leaving the mechanical room. Thus, there would be a second smoke isolation damper with a similar savings potential.

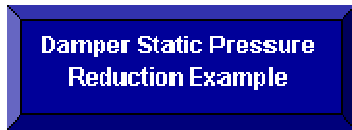
In addition, it is likely that the system would have a return air path that parallels the supply duct route. Even if the return system was a plenum return, the places where openings were provided to allow the air to pass into the corridor ceiling from the area served by the air handling unit, and to allow the air to pass from the corridor to the mechanical room would need to be protected to match the rating of the separation where the penetration occurred. Thus, there would be two additional smoke isolation dampers in the return system. Because the return velocities would be low relative to the supply flow, the damper areas would be significantly larger. Therefore, for the return smoke isolation damper calculation, the energy savings would be lower and the first cost would be higher, relative to the supply damper. The return velocities would be about half of the supply velocity, which would imply that the damper area would be twice that of the supply. Consequently, the first cost associated with the return damper would be twice as

much as that associated with the supply damper since the damper costs are directly related to the damper area. The operating cost savings associated with the return damper would be one quarter of that associated with the supply damper since the operating costs are related to the pressure drop and the pressure drop is a function of the square of the flow rate (see [Equation C.3](#) and the related discussions of the square law).

Thus by making a few assumptions and spending a few minutes reviewing the plans, the savings calculated for one component can be quickly compounded to reflect the savings and cost potentials for the system. The system wide savings can then be applied to similar systems on the project and also be used as a database for subsequent projects. See the case study in Section [C.2.7](#) for an example of a project where these techniques were used to extrapolate the savings from one calculation to an entire building to considerable advantage.

C.2.4.5. Sample Spreadsheet

The calculation of savings from using airfoil smoke dampers is actually simple to perform once you are familiar with the process. You may even find yourself making rough field estimates based on the equations with nothing but a calculator. To provide a basis for more formal calculations for reports and to automate the evaluation process, the following spreadsheet is provided.



[Link to go to a spreadsheet that performs the calculations described in Example 1.](#)

This spreadsheet can be used to gain a greater understanding of the calculation procedures and as a starting point for calculations on your projects.

C.2.5. Example 2: Meeting the Fan Horsepower Rating

Even though energy savings are important, they sometimes pale in comparison to a field problem where the failure of a system to perform could cost significant capital to correct. Consider the following scenario.

The 35,000 cfm VAV air handling system on your project is failing to deliver the required airflow. In an effort to correct this, the balancing contractor has increased the fan speed until the system's 25 hp motor is producing 28.75 hp by running in into it's service factor. After the adjustment, the contractor retested the system. Unfortunately, the results are not good. The test data indicates that the system is still short of capacity and that it will take 29.25 brake horsepower to achieve the required flow rate with the system as it exists. To provide for dirty filters, 30 or more brake horsepower is probably a more realistic assessment of the problem. The contractor checked the fan curve and determined that the fan will still be within its class rating at the proposed operating point and has started the construction paperwork process that will install a new motor on the system. This will be expensive because in addition to the motor, the variable speed drive serving the system will need to be replaced to handle the higher current requirements associated with the larger motor size. Accusations are already starting to fly, with the Owner making a strong statement regarding his unwillingness to pay for the change *and* his unwillingness to accept a system producing less than design performance. The designer indicated that the problem was no doubt the result of the contractor not complying with SMACNA duct construction standards referenced by the project specifications. The sheet metal contractor is staring at the one line duct installation drawings wondering exactly where in the duct system he should have used a different one of the myriad of duct fitting designs covered by the SMACNA standard to prevent the problem.

This is a difficult problem for everyone involved, and situations like this do come up in the field. Frequently, the problem is solved by throwing more energy at it in the form of higher fan speeds, more brake horsepower, and as suggested in this case by the balancing contractor, larger motors. All of these solutions penalize the system's energy efficiency for the life of the system. By solving the horsepower equation we have been discussing for static pressure, we may be able to gain a different perspective on the problem.

Specifically, it might be worth asking "how much static pressure would need to be eliminated from the system in order to reduce the horsepower requirement to a level that could be handled by the motor?" Once this static pressure was identified, analysis supplemented by focused pressure drop testing may reveal a relatively inexpensive modification that will provide the necessary level of performance without having to throw more power and energy at the problem. Such a solution could provide benefit for everyone involved by minimizing the expense associated with correcting the problem and improving the energy efficiency of the building for its life.

In our example, we need to achieve enough horsepower savings to allow the system to deliver design flow without overloading the motor. Strictly speaking, the motor is rated for 25 hp, and some would say it is in an overloaded state if the horsepower exceeds that value. On the other hand, most motors have a service factor rating and the motor can be safely operated at a load condition that exceeds its nameplate power rating by the amount of the service factor. Typically, service factors are in the range of 1.10 to 1.15. Running the motor at a power level in excess of its nameplate means that the motor will be drawing current in excess of its nameplate. The motor will tend to run hotter which can affect the service life of its windings.

As a starting point, solve [Equation C.1](#) for pressure difference and determine how much of a static pressure reduction is required to get the system back to where the motor can provide design flow running at its service factor limit.

Assumptions:

Equation C.6: Static Pressure Reduction Required to Achieve a Target Horse Power Reduction

Solving [Equation C.1](#) results in the following relationship:

$$P_{\text{Saved}} = \frac{\text{Horsepower} \times 6,356 \times \eta_{\text{FanStatic}} \times \eta_{\text{Motor}}}{Q}$$

Where:

Horsepower = The horsepower savings required at the motor terminals

Q = The system design flow rate in cubic feet per minute (cfm)

P_{Saved} = The static pressure reduction required to achieve the desired horsepower savings in inches water column (in. w. c.)

6,356 = A units conversion constant

η_{FanStatic} = The fan static efficiency

η_{Motor} = The motor efficiency

The horsepower requirement project to achieve design flow = 29.25 hp. This comes from the projection made by the balancer based on the current operating point via the fan affinity laws.

The horsepower available running the motor into its service factor = 28.75 hp. This is the maximum acceptable loading for the motor. If we can get below this, it would be good because it would provide a safety factor and some room for the filters to load. This value is a defined limit that will help us establish a target for eliminating pressure drop.

The minimum horsepower savings desired = 0.50 hp. The system flow rate = 35,000 cfm. This flow rate is the system's design requirement, established by the contract documents and design intent.

Calculation:

$$P_{\text{Saved}} = \frac{\text{Horsepower} \times 6,356 \times \eta_{\text{FanStatic}} \times \eta_{\text{Motor}}}{Q}$$

Flow rate = $Q = 35,000$ cfm

Desired horsepower reduction = $\text{Horsepower} = 0.50$ hp.

Assumed fan efficiency = $\eta_{\text{fan}} = 79\%$ (based on shop drawing information)

Assumed motor efficiency = $\eta_{\text{motor}} = 89\%$ (based on shop drawing information)

Required static pressure reduction = $P_{\text{saved}} = 0.06$ in.w.c.

Interestingly enough, it takes only a minor reduction (0.06 in.w.c.) in static pressure to bring the system performance back to a point where the existing motor it is adequate. In many instances, this revelation may throw a new light on the problem since it opens the door to a variety of options other than installing a larger motor and a bigger drive.

Methods to analyze and reduce system static pressure include:

- 1 Generate a field measurement-based pressure gradient diagram for the system and compare it to the one predicted by the engineer's calculations.** This type of functional testing can often be done in coordination with the balancing contractor. Given the relative simplicity of this effort, it would be desirable to undergo this process before a taking any other steps, regardless of the outcome of any other analysis of the problem. To make the fan selection, the engineer should have performed a static pressure calculation for the system, and, given the nature of the current problem, they should be willing to share the information with the rest of the team in an effort to solve the problem without incurring additional cost. Even in situations where there is not a readily available pressure gradient diagram, like a retrocommissioning process or a non-engineered system modification, generating the required information is not that difficult once you have been through the process a couple of times. The *ASHRAE Handbook of Fundamentals* can provide guidance in this process and also includes numerous fitting loss coefficients, duct friction charts, as well as examples of this type of calculation.

Once a calculated pressure gradient diagram has been generated, field testing can be used to compare the predicted performance with the actual performance by taking readings at various locations in the duct system. If the actual losses in a given section do not correspond well with the prediction, investigation is probably warranted to determine if there is some obstruction in the duct system that is causing the unexpected pressure loss. This investigation may require an internal inspection of the system, and an access panel may need to be cut. But, in the context of the cost to change a motor, the cost of adding an access door or two will be inconsequential. In many cases the inspection that the doors enable will reveal that the problem is due to a problem like missing turning vanes, a duct liner that has come loose and is caught on the duct reinforcing rods, or an elbow or a workman's jacket that was inadvertently left in the duct and now is lodged against some turning vanes. Even if no problems are revealed and a motor change is the only solution, having verified that there are no unwarranted, easily eliminated restrictions in the duct prior to undertaking this expense usually justifies the cost of the access panel.

The following links will provide some useful information with regard to this process.

Static Pressure Loss Spreadsheet

The static pressure loss calculation spread sheet and the pressure gradient diagram can be used to gain a greater understanding of the calculation procedures and as a starting point for your own calculations.

System Diagram

[Link to the system diagram associated with the pressure gradient diagram and static pressure loss calculation. \(PDF\)](#)

- 2 One place where it is often possible to eliminate static pressure from the system is at the filter bank. If the system has prefilters, it may be possible to simply run without them at the expense of having to change the final filters more frequently, assuming that the final filters are ahead of any of the unit's internal components. Switching from conventional filters to extended surface area filters for the final filters can mitigate this. Extended surface area filters can provide nearly twice the dust holding capacity of conventional designs and do it at much lower initial pressure drops, often 50% less. These filters are discussed in greater detail in Chapter 10: Filtration. (Even if the system doesn't have static pressure problems switching to these filters can provide energy and resource conservation benefits that may be desirable).
- 3 Another possible place to eliminate static pressure is at a major fire or smoke damper in the common supply duct. As can be seen from [Figure C.3](#) and the previous calculation example, switching to an airfoil blade design can often save a significant amount of static pressure. This approach has the disadvantage of requiring that the existing dampers be replaced, which will cost time and money and will probably require a system outage. But it may be more economical than a motor and drive replacement, especially if the sheet metal contractor is forced to do the work at his cost due to the way his contract is interpreted. In the bigger picture, this approach has the advantage of reducing the system energy requirement instead of increasing it. This provides a long-term benefit to the Owner, which can be a useful incentive for obtaining their approval and assistance in performing the work.
- 4 Another option similar to that described in item 3 is to modify a fitting in the duct system to improve the losses through it. Figures 18.1, and 18.3 through 18.5 in *Chapter 18 Distribution*, provide some examples of the savings potential associated with this option. The prime targets for this approach are fittings near the discharge of the fan and fittings in the large distribution ducts because the velocities in these ducts tend to be relatively high even though their friction rates are relatively low (see Table 18.2 and the related discussion in Chapter 18). Obviously, this approach also will require an outage and material and labor expenditures, but in some cases, these costs could be lower than changing the motor.

The biggest advantage for an approach that eliminates static pressure instead of adding motor horsepower is that it improves the efficiency of the HVAC system for the life of the system. In our case, assuming an average flow rate of 70% of design due to the impact of the load profile, the annual operating savings are in the range of \$35 to \$106 depending on the hours of operation. While this is not a lot of money, it's better than adding operating cost and represents a savings that will be realized every year of the system's life.

Horse Power Reduction Example

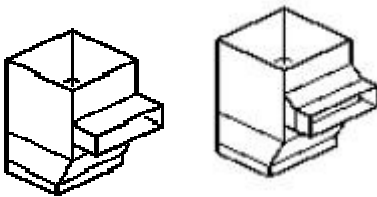
[Link to a spreadsheet that performs the calculations described in the preceding example.](#)

This spreadsheet can be used to gain a greater understanding of the calculation procedures and as a starting point for calculations on your projects.

C.2.6. Example 3: Measuring the Impact of Turning Vanes

Figure C.4 Subtle Differences in Fitting Design Make Big Differences in Performance

The difference between the design of these two fittings is very subtle. The one on the right has a 45° flare where the branch connects to the duct and the one on the left doesn't. Yet, at 2,000 fpm (a common duct riser velocity), the pressure loss through the one on the left could be 2 to 3 times as high as the one on the right. (See [Figure 11.1](#) in Chapter 11: Supplemental Information)



Translating the duct design information on the contract drawings to a real duct system that fits in the confines of the existing conditions presented by the building structure and architectural elements often requires interpretation and compromise on the part of the tradesmen installing the system. In addition, the hectic pace of construction can cause details to be missed or misinterpreted by less experienced field personnel. For duct systems and fittings, subtle differences in interpretation can often make big differences in the pressure loss associated with a fitting as can be seen from [Figure C.4](#) and the referenced figure in Chapter 13. On a one line duct drawing, where the riser may show up as a rectangle and the connection to it simply as a line touching the rectangle, there would be very little guidance for the field staff to tell them that they should use the better fitting. Even in a two line drawing, a note or an elevation detail would be required to really make things clear.

One line drawings also leave the need for turning vanes open to interpretation. SMACNA construction standards show how to build elbows in a variety of ways, including without vanes. The specification may require turning vanes, but many times the tradesmen actually hammering the duct together never see the specs and are working from the drawings, past experience, and some general instructions from their foremen. Consider the following example.

You are doing a walk-through of a section of your project where the ceiling is about to be installed. A week ago, when you were last on site, the sheet metal contractor had just been starting to run the duct through this area, which is the first portion of the system outside the mechanical room. Now, the trunk duct has been run through the area and is ready for extension to the next area scheduled for renovation. In addition, the distribution terminal equipment and ducts for the area have been run. As you walk around looking at the installation, you notice that the tell-tale line of screws across the corners of a mitered elbow that usually indicate that turning vanes are missing. You remember that the drawings for the system are one line drawings, so the vanes themselves would not actually show up on them. You start to wonder how big a problem this might be, so you run a few quick calculations using [Equation C.4](#) and [Equation C.5](#).

First you look at one of the small distribution ducts.

Assumptions:

You assume a flow rate of 350 cfm based on the information on the drawings. The 12 by 6 inch duct size is easy to find by simply measuring it. Since it is externally insulated, you don't have to worry about adjusting the dimensions for duct liner. These two pieces of information allow you to determine the velocity in the duct since the flow is equal to the area of the duct multiplied by the velocity through that area. In this case, the result is a velocity of 700 feet per minute.

The next piece of information you need is the loss coefficients for the different elbow designs (with and without turning vanes). But, having had problems with fittings before, you carry copies of the *ASHRAE Fundamentals Handbook* loss coefficient tables for some of the more common fittings with you in your clipboard for reference on just such an occasion. Referring to the tables, you determine that the best turning vane design in a mitered elbow will result in a loss coefficient of 0.11. For a mitered elbow with no turning vanes, the coefficient is an alarming 1.03, nearly 10 times that of the fitting with turning vanes. You now have everything you need to do the math.

Calculation:

Calculate the velocity pressure using [Equation C.5](#).

$$P_{velocity} = \left(\frac{V}{4,005} \right)^2$$

Velocity pressure = $p_{velocity}$ = 0.031 in.w.c.

Next calculate the loss for the two different elbow designs, with and without vanes, using [Equation C.4](#).

$$\Delta p_{fitting} = C_e p_{velocity}$$

Loss for the elbow with no turning vanes = $p_{Fitting}$ = 0.031 in.w.c.

Loss for the elbow with turning vanes = $p_{Fitting}$ = 0.003 in.w.c.

Difference (avoided pressure drop if vanes are installed) = 0.028 in.w.c.

You start to relax a little bit. The missing turning vanes in the small distribution duct will have a minimal impact on the performance of the system and probably will not matter. But, you decide to perform the same calculation for the elbow in the duct main that also appears to be missing turning vanes because the velocities are higher there. Using a similar procedure, you obtain the following results.

Flow rate = 20,000 cfm

Duct height = 24 inches

Duct width = 60 inches

Velocity = 2,000 fpm

Velocity pressure = 0.25 in.w.c.

Loss for the elbow with no turning vanes = 0.26 in.w.c.

Loss for the elbow with turning vanes = 0.03 in.w.c.

Difference (avoided pressure drop if vanes are installed) = 0.23 in.w.c.

The impact here is far more significant due to the higher velocities. Since larger ducts contain a much greater volume of air for a given perimeter, they have a much higher air handling capability at a given friction rate as compared to smaller ducts, see [Table 13.1](#) in *Chapter 13: Supplemental Information*.

That's the good news. The bad news is that it means the velocities are higher and the implications of a poor fitting design are much more significant. In our example, both ducts had been sized at the same

friction rate; 0.10 inches w.c. loss per 100 feet of duct, a common low velocity, low pressure system standard. But, the velocity in the larger duct was nearly three times that in the smaller duct, and the velocity pressure was nearly 9 (three squared) times that in the smaller duct due to the square relationship between velocity pressure and velocity.

Based on the results of your calculation, you decide to bring the subject up with the project team. The foreman checks into it reports that the vanes are not there. You suggest that the three of you have a discussion with the designer about the problem. The elbows in the low velocity distribution duct system will probably be O.K., especially since the area you were looking at was the first zone off of the main and will probably have more static than it needs and since vanes will be provided for the rest of the project. Asking the contractor to fix the problem in the main and letting the problem go unless the balancing contractor cannot get design flow in one of the branch ducts may represent an attractive compromise to everyone.

Had you not detected the problem, it may have never been picked up. While significant, the extra 0.23 in.w.c. of loss represented by the missing vanes in the elbow in the main may have been overcome by simply speeding up the fan when it failed to deliver the design performance. At 20,000 cfm, the extra static represents a little over one horsepower, and, if the brake horsepower requirement of the system was below the motor capability by that much, speeding up the motor would be a viable and quickly achieved solution to any design flow problems created by the missing vanes. If the system was a constant volume system and operated 10 hours per day, the annual energy penalty associated with speeding up the motor to make up for the missing vanes would be in the range of \$200 per year. If the system ran continuously, it would amount to over \$600 per year.

**Duct Fitting Pressure
Loss Example**

This spreadsheet performs the calculations described in the preceding example. The energy and horsepower calculations are also included.

This spreadsheet can be used to gain a greater understanding of the calculation procedures and as a starting point for calculations on your projects.

C.2.7. Case Study: Under 60 Minutes of Effort, Over \$60,000 Savings

As an example of how a small savings potential can compound into a large savings potential, consider the following example drawn from a high-rise project. A 20 story high-rise building has 16 air handling systems, all of which are located in a mechanical space above the 20th floor. Distribution duct work is contained in several large, fire and smoke rated mechanical chases that penetrate through the full height of the building, which also serve as return paths back to the air handling systems. All duct and return air penetrations entering and leaving these chases are protected with fire or combination fire and smoke dampers, depending on the chase rating at the location. The supply ducts for all of the systems carry significant volumes of air and are large with relatively high velocities despite the low friction rates (see [Table 11.2](#) in Chapter 11: Supplemental Information).

In the course of reviewing the shop drawings for the combination fire and smoke dampers on the project, the commissioning agent looked at the installed location for the dampers on the plans to see if the actuator locations shown on the shop drawings would allow sufficient space for the actuators to be maintained when the dampers were installed in the congested areas around the chase. As he looked at the drawings, he noticed that for all of the systems, a second smoke isolation damper was shown on the drawings,

typically within 10 to 30 feet of the combination fire/smoke damper installed at the chase. A little digging around in the construction documents revealed that the second damper was:

- From a smoke rating standpoint, identical to the damper in the ducts at the shaft wall penetration.
- Controlled to close any time the air handling unit it was associated with was off to comply with the NFPA 99A requirements for smoke isolation dampers on air handling systems over 15,000 cfm. The damper at the chase was required to maintain the integrity of the separation provided by the shaft wall. It was controlled to close any time there was a fire alarm and open if commanded to do so from the smoke management system control panel in the lobby.

Duct velocities in the vicinity of the dampers were in the 2,000 to 2,500 fpm range, so the damper represented a significant pressure drop, even with the airfoil blade design that was to be provided. Based on past experience and the wording of the code, the commissioning agent decided to investigate the following ideas:

- **Could the combination smoke and fire damper at the chase also provide the smoke isolation function?** If the control sequence associated with it were modified to include closing the damper when the air handling unit associated with it was off, the need for the second smoke damper currently in series with the combination fire and smoke damper would be eliminated.
- **What energy penalty is associated with the second damper in terms of pressure loss?**
- **What first cost penalty is associated with furnishing, installing and controlling the second damper?**

Since the commissioning agent had been involved in other projects where a smoke and fire damper associated with protecting a rated separation had also been used to provide the smoke isolation function for an air handling system, he thought that eliminating the second damper may represent a viable first cost and energy cost saving opportunity. To understand the potential savings associated with this, he took the following steps:

- 1 Using Equations [Equation C.1](#) and [Equation C.2](#) A, B, and C and the pressure drop for the dampers shown on the shop drawings, he determined that eliminating the damper from one of the typical systems could save around \$275 in operating cost due to the elimination of approximately 0.18 inches w.c. of static pressure from the system.
- 2 The extra dampers were worth \$75 to \$95 per square foot, depending on the exact configuration. Installation could easily double this cost when wiring, control, and commissioning were considered. Therefore, the installed cost for the potentially redundant damper could easily be in the \$150 to \$190 per square foot range.
- 3 The supply duct for the unit had a cross sectional area of nearly 16 square feet. Based on the information above, the damper probably represented an equipment cost of approximately \$1,200 to \$1,500 and an installed cost of double those figures.
- 4 On the return side, even though the return smoke isolation dampers at the shaft were associated with multiple units, and thus could not be considered redundant, each unit had its own return damper associated with its economizer cycle. Based on past experience, the commissioning agent knew that these dampers could be furnished with a smoke rating and could be used to provide the smoke isolation function as well as the economizer function. If that were possible on this project then there were first cost and energy savings available by eliminating the redundant return smoke isolation dampers.
- 5 To evaluate the savings potential, the commissioning agent used the following fairly simple assumptions:

- Since the return velocities were about half of the supply velocities, the return damper area would be about twice the supply damper area. Since the damper cost was generally related to the damper area, the savings in first cost for the return damper would be about twice that associated with the supply damper.
 - Since the return velocities were half of the supply velocities, the pressure losses through the return damper could be anticipated to be one quarter of those associated with the supply dampers, due to the “square law” (see [Equation C.3](#)). Since the operating costs are directly related to the pressure losses, the operating costs associated with the possibly redundant return dampers could be anticipated to be about one quarter of those associated with the supply damper.
- 6** Based on the preceding calculations, the net savings potential for the system the commissioning agent appeared to be:
- Operating cost savings in the range of \$340.
 - First cost savings in the range of \$7,500 to \$9,000.
- 7** Since there were 15 identical or nearly identical air handling systems, it was reasonable to assume that similar savings potential existed for all of them.

In other words, after about an hour of effort, the commissioning agent had identified the potential to save \$5,500 per year in operating cost and \$120,000 to \$140,000 in first cost. The operating cost savings will be realized every year of the system’s life cycle. In this case the systems had a 25-year design life cycle.. Thus, the accumulated operating savings had significance beyond the \$5,500 per year identified in the analysis.

The calculations performed were far from exact. What they did do was identify the potential for significant first and operating cost savings. Based on his analysis, the commissioning agent spent another 15 minutes to write an RFI (Request For Information) suggesting that perhaps, the redundant dampers could be eliminated and that this concept was worthy of additional evaluation. As a result, the topic was discussed at the next construction meeting. Since the project was experiencing some budgetary pressure, the first cost savings were very attractive to the Owner. The magnitude of the savings potential coupled with the fact that they could be evaluated from some fairly straightforward math and easily obtained costs (damper square footage times cost per square foot) made additional analysis unnecessary. It turned out that the code authority associated with the project happened to be present at the construction meeting. After studying the situation for a few minutes, the code authority agreed with the commissioning agent’s assessment. As a result, the next day, the general contractor initiated the paper work required to remove the dampers from the project. The \$275 in savings identified by the actual calculation performed by the commissioning agent had mushroomed into the potential for thousands of dollars in energy savings per year and tens of thousands of dollars in first cost

^[1] The fan laws or fan affinity laws are a set of relationships that can be applied to a fan system to predict new operating parameters based on existing operating parameters. They are discussed in detail in a variety of sources including the *ASHRAE Systems and Equipment Handbook* in the *Fans* chapter and the *NEBB Testing Adjusting Balancing Manual for Technicians under Air Systems*.

^[2] As a side note, the filter pressure drop indicators can often be used to take a measurement across some other component if you can come up with a little tubing and a drill to make some holes.

^[3] The referenced ASHRAE handbook also provides information on the effect of inlet vanes and blade pitch variation on fan performance if these techniques are employed to control flow rate in a system. Trane provides similar information in *Fans and Their Application in Air Conditioning* available from Trane’s Commercial Bookstore that can be accessed through the

Trane web site www.trane.com. Both of these resources could be used to evaluate the impact of other flow variation techniques on energy consumption if you find yourself dealing with such a system.

^[4] This is another one of the fan affinity laws. These laws were referenced previously in the discussion of the flow term in equation C.1. They are discussed in detail in a variety of sources including the *ASHRAE Systems and Equipment Handbook* in the *Fans* chapter and the *NEEB Testing and Balancing Manual* under *Air Systems*.

^[5] A 50% reduction is a reduction by 1/2. Thus, based on the fan laws, the power requirement would be reduced to $(1/2)^3$ or 1/8 or 12.5% of the requirement at 100% flow.
