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design brief

ADVANCED VARIABLE-AIR-VOLUME (VAV) SYSTEMS

Summary

This Advanced VAV System Design Brief provides recommendations to help engineers improve the efficiency of large HVAC systems. It focuses on built-up variable-air-volume (VAV) systems in multistory office buildings.

The recommended measures promote efficient, practical designs that advance standard practice, achieve cost-effective energy savings, and can be implemented using current technology. Here are some of the key recommendations:

- Reduce design system static pressure

- Employ demand-based static pressure reset

- Use low-pressure plenum returns/relief fans

- Employ demand-based, supply temperature reset to reduce reheat energy and extend economizer effectiveness

- Design fan systems to turn down and stage efficiently

- Size terminal units to balance energy impacts of pressure drop and minimum airflow control

- Set terminal unit minimums as low as required for ventilation and use intelligent VAV box control schemes to prevent stratification

- Employ demand-based ventilation controls for high-density occupancies

- Design conference rooms to provide ventilation without excessive fan energy or reheat

Of all the recommendations in this Design Brief, VAV box control and supply air pressure reset often have the largest impact on system efficiency. Design engineers are encouraged to pay particular attention to these two issues.

Many large HVAC systems use significantly more energy than necessary. Design engineers can improve the efficiency and cost effectiveness of built-up VAV systems by following recommendations that emphasize integrated design and designing for the full range of system operation.

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Introduction

This Design Brief provides an authoritative new resource for heating, ventilation, and air-conditioning (HVAC) designers. It presents the most current recommendations on variable-air-volume (VAV) airside system design, and provides brand-new information on fan selection and modeling.

The recommended measures incorporate findings from a recent study of built-up VAV systems conducted for the California Energy Commission. That three-year study, which included field monitoring of five large office buildings in California, resulted in the publication of the *Advanced VAV System Design Guide*, which presents a more comprehensive version of the recommendations in this Design Brief.[1]

Both publications—this Design Brief and the full Design Guide—focus on built-up VAV systems in multistory office buildings in California or similar climates (California has 16 climate zones). But much of the information is useful for a wider range of system types, building types, and locations.

The recommendations in this Design Brief address airside system design, with an emphasis on getting the air distribution system components to work together in an integrated fashion. Of all the recommendations in this Design Brief, VAV box control (section 3) and supply air pressure reset (section 7) will have the largest impact on system performance. Design engineers are encouraged to pay particular attention to these two topics.
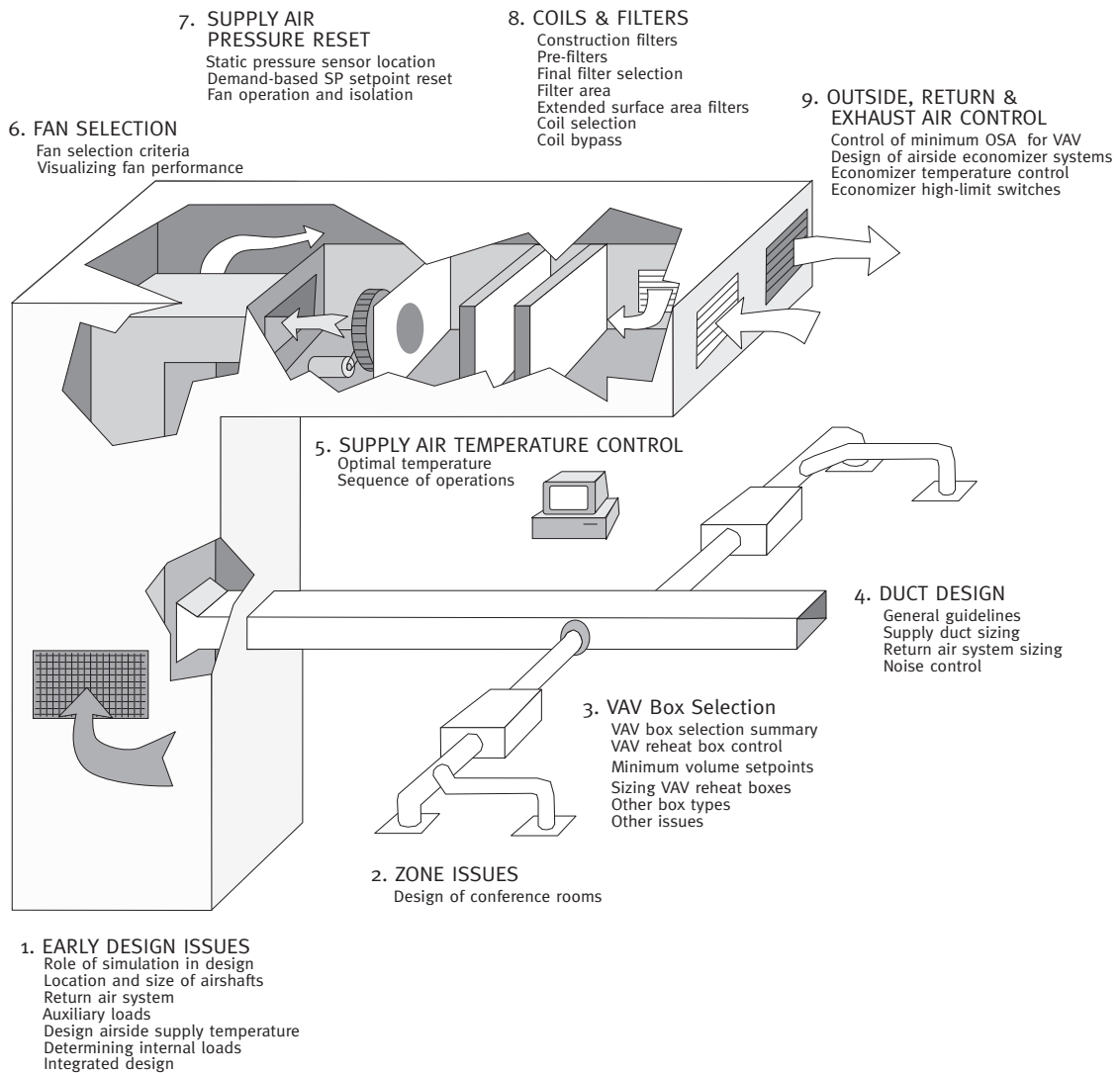
Figure 1 provides an overview of all the topics covered in this Design Brief.

Market Share

Over the next 10 years, about 30 million square feet per year of large office buildings will be constructed in California, equal to 20 percent of new construction in the state. Approximately one-half of those buildings will be served by VAV reheat systems. Therefore, the recommendations in this Design Brief will apply to roughly 150 million square feet of new buildings built in the 10-year period between 2003 and 2012. This equals roughly 10 percent of the total commercial construction forecast.

This Design Brief is organized around key design considerations and components that affect the performance of VAV systems.

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While chilled water systems account for only about 4 percent of the HVAC systems in commercial buildings, they account for as much as 45 percent of the statewide cooling capacity. And chilled water systems with VAV reheat—the type of system addressed by this Design Brief—are estimated to account for slightly more than 20 percent of all cooling capacity. Clearly, the performance of these systems has a tremendous ability to affect statewide energy use.

Following the practices in this Design Brief can lead to major improvements in system performance, energy efficiency, and occupant comfort.

These practices can lead to major improvements in system performance, energy efficiency and occupant comfort.

Significant fan and reheat energy savings are possible through the design strategies promoted in this Design Brief. The potential savings are illustrated in the graphs below, which present results of simulations conducted as part of the Advanced VAV System Design study mentioned above.[2] In this example, the "Standard" case is a reasonably efficient code-complying system, while the "Best" case includes a number of the improvements suggested in this Design Brief. The result of this simulation show that **fan energy drops by 50 to 60 percent, and reheat energy is reduced by 30 to 50 percent**.[3]

Most of the savings are due to the efficient "turndown" capability of the best practices design and the fact that HVAC systems operate at partial load nearly all the time. The most important measures are careful sizing of VAV boxes, minimizing VAV box supply airflow setpoints, and controlling VAV boxes using a "dual maximum" logic that allows lower airflows in the deadband mode, all of which are discussed in section 3, and supply air pressure reset control, which is discussed in section 7. Together these provide substantial fan and reheat savings because typical systems operate many hours at minimum—yet higher than necessary—airflow.

Extra time carefully spent in early design can save weeks of time later.

Figure 2: Fan and Reheat Energy Savings, San Francisco

Simulation results for a typical San Francisco office building, comparing standard VAV design practice to the best practices promoted in this Design Brief.
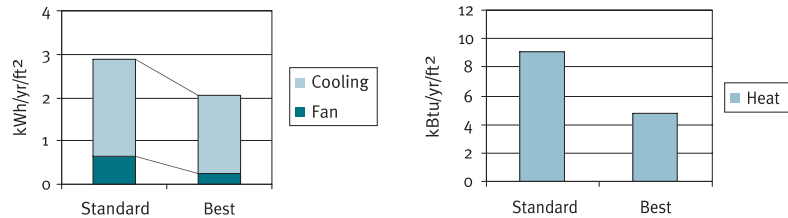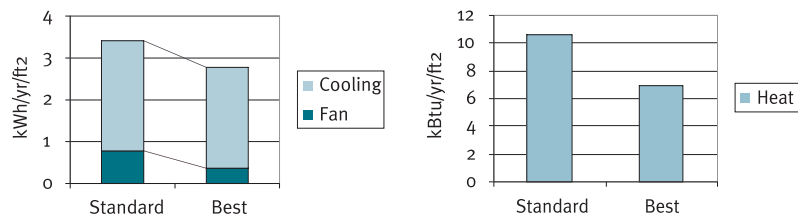

Figure 3: Fan and Reheat Energy Savings, Sacramento

Simulation results for a typical Sacramento office building, comparing standard VAV design practice to the best practices promoted in this Design Brief.

Early Design Issues

The old adage, "An ounce of prevention is worth a pound of cure," certainly holds true for the design of built-up VAV systems, which are complex custom assemblies. Extra time carefully spent in early design can save weeks of time later in the process. It can also help improve client relations and reduce construction and operating costs.

Below is a short list of early design issues that VAV system designers should pay particular attention to. For detailed information about early design decisions and integrated design, refer to the Advanced VAV System Design Guide and to the Energy Design Resources Design Brief, "Improving Mechanical System Energy Efficiency Through Architect and Engineer Coordination."[4]

Key Recommendations: Early Design Issues

Use simulation tools to understand the part-load performance and operating costs of system alternatives. Because built-up VAV reheat systems are so complex, it is very important to use building energy simulation tools to assess design alternatives. Many of today's powerful simulation tools are available for free or at a reasonable cost, and have simplified user interfaces that allow a complex model to be built in 15 minutes or less. Building energy simulation should be an integral part of design at all phases, from schematic design through acceptance and post-occupancy.

Size and locate the air shafts to reduce system static pressure and fan energy use. The location and size of air shafts can have a tremendous impact on the cost and efficiency of the mechanical systems as well as on architectural space planning and structural systems. Good design practices include using multiple air shafts for large floor plates (over about 15,000 ft^2), and placing the air shafts close to, but not directly under, the air-handling equipment for built-up systems.

Use return air plenums when possible because they reduce energy costs and first costs. Establish the return air system design very early in the design process, because it has a significant impact on many issues, including the cost and complexity of the mechanical system. In general, plenum returns have a very low pressure drop, allowing the use of either barometric relief or low-pressure relief fans.

Design the HVAC system to efficiently handle auxiliary loads that operate during off hours. Most buildings will require auxiliary cooling systems to serve loads that do not operate on the normal HVAC system schedule. It is important to evaluate the performance of the HVAC system when serving only these loads, which typically are a small percentage of the total building load.

Determine the supply air temperature at a fairly early stage in design. The supply air temperature needs to be determined relatively early in order to calculate airflow

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INTEGRATED DESIGN

Achieving optimal airside efficiency requires more than just selecting efficient equipment and control schemes. It also requires careful attention to early architectural design decisions, and a collaborative approach to design among all disciplines.

Teamwork is critical to the design of high-performance buildings. Decisions that are not traditionally within the purview of the mechanical designer nonetheless have a great impact on the cost, efficiency, and success of their design. For example, the glazing selected by the architect affects thermal loads and comfort. Use of high-performance glazing or shading devices can drastically reduce the size of the mechanical equipment and improve comfort.

An integrated design process can improve the comfort and productivity of the building occupants while at the same time reducing building operating costs. A high-performance building can be designed at little or no cost premium with annual energy savings of 20 to 50 percent compared to an average building, but this level of savings will require a high level of cooperation among design team members.

For more about the integrated energy design process, see the Energy Design Resources Design Brief, "Integrated Energy Design."

Monitored loads of an office building in Sacramento illustrate the importance of designing for efficient part-load operation. Figure 4, based on measured system airflow in a 955,000 ft², 25-story office building, shows that the HVAC system may operate at only one-half of the design airflow for the bulk of the time.

This is quite typical for an office building. The design airflow for a typical floor of the monitored building is 0.83 CFM/ft². During cool weather, the airflow doesn't exceed 0.4 CFM/ft², and in warm weather airflow is seldom greater than 0.5 CFM/ft². Figure 5 shows similar results for cooling delivered to that floor.

requirements and equipment size. In California office buildings, a temperature in the range of 52°F to 57°F results in a good balance between efficiencies of the chilled water plant and the air distribution system at peak cooling conditions. Size interior zones for 60°F or higher supply air temperature to allow for supply air temperature reset in mild and cold weather.

Avoid overly conservative estimates of lighting and plug loads. Given the steady downward trend in lighting power, traditional lighting load assumptions may no longer be valid. Many office spaces are now designed with less than 0.8 W/ft² of lighting. In addition, studies measuring plug loads show that
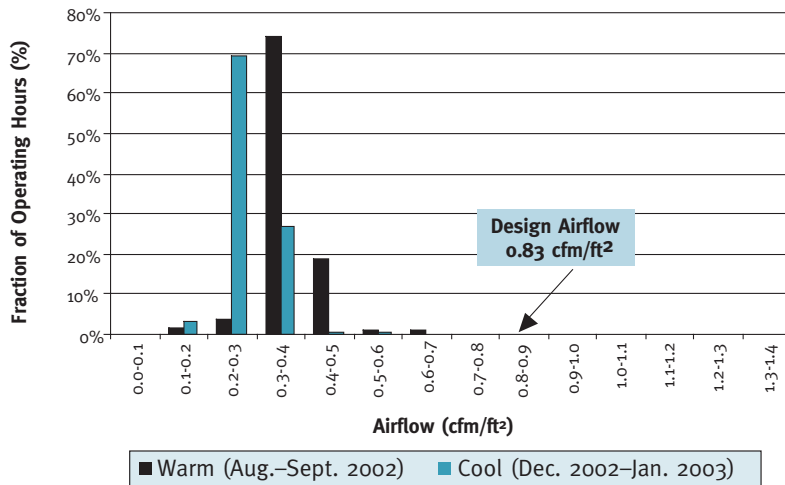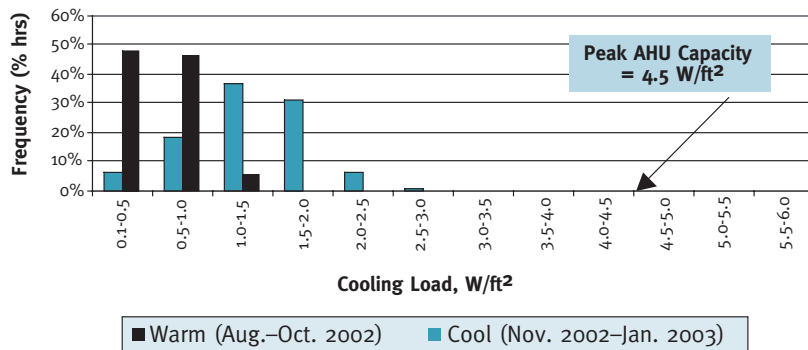
Figure 4: Measured System Airflow

Design Airflow
0.83 cfm/ft²

■ Warm (Aug.–Sept. 2002)   ■ Cool (Dec. 2002–Jan. 2003)

Figure 5: Measured Cooling Delivered By Air Handler

Peak AHU Capacity
= 4.5 W/ft²

■ Warm (Aug.–Oct. 2002)   ■ Cool (Nov. 2002–Jan. 2003)

actual loads are much lower than indicated by nameplate ratings and much lower than commonly used design values.

Zone Issues

Zone issues that must be addressed by the HVAC designer include thermal comfort, zoning and thermostats, use of CO_2 sensors for demand control ventilation, occupancy controls, window switches, and issues affecting the design of conference rooms. While all these topics are important, two deserve particularly close attention: demand control ventilation and conference room design.

Key Recommendations

Consider demand control ventilation in any space with expected occupancy load at or below 40 ft²/person. Title 24 currently requires demand ventilation controls in very dense occupancies (10 ft²/person or less). In the 2005 update to California's Title 24 Standard, demand ventilation controls using CO_2 sensors are required on all single-zone systems serving dense occupancies (40 ft²/person or less).

Although not required, almost any VAV reheat zone serving an expected occupant load denser than about 40 ft²/person can potentially benefit from CO_2 control.

For conference rooms, use either a VAV box with a CO_2 sensor to reset the zone minimum or a series fan-powered box with zero minimum airflow setpoint. Conference rooms, because of their variable occupancy and high ventilation requirements, can lead to excessive reheat if the minimum flow is not reset. For resetting the zone minimum, a VAV box with a CO_2 sensor uses less mechanical system energy than an occupant sensor.

Alternatively, use a series fan-powered VAV box with a zero minimum primary airflow setpoint. Because Title 24 allows transfer air to be used to meet ventilation requirements, minimum ventilation can be provided by the series fan supplying only plenum air, eliminating central air and reheat. This is the simplest option from a controls perspective and one of the most efficient.

VAV Box Selection and Controls

This section discusses three of the most important measures for obtaining significant fan and reheat energy savings: careful sizing of VAV boxes, minimizing VAV box minimum airflow setpoints, and controlling VAV boxes using a "dual maximum" logic that allows lower airflows in the deadband mode. These measures, together with supply air pressure reset control (discussed in section 7), provide substantial fan and reheat savings because typical systems operate many hours at minimum—yet higher than necessary—airflow.

This section gives guidance on selecting and controlling VAV boxes with hot water reheat; it applies only to VAV boxes with pressure-independent controls.

Key Recommendations for VAV Box Selection and Controls

For all except very noise-sensitive applications, select VAV boxes for a total (static plus velocity) pressure drop of 0.5 inches of water (in. W.C.). For most applications, this provides the optimum energy balance. For a given design airflow rate, more than one box size can meet the load, so the question is which size to use. Smaller VAV boxes will have a higher total pressure drop, increasing fan energy, and higher sound power levels. However, larger boxes cost more and are more limited in how low the minimum airflow setpoint can be set, which can increase fan energy and reheat energy under low load conditions. For most applications, boxes should be selected for a total pressure drop of about 0.5 in. W.C. This applies to all boxes regardless of how close or remote the boxes are to the fan.

Calculate the total pressure drop. The total pressure drop (ΔTP), which is equal to the static pressure drop (ΔSP) plus the velocity pressure drop (ΔVP), is the true indicator of the fan energy required to deliver the design airflow through the box. Unfortunately, manufacturers typically only list the static pressure drop, which is always lower than the total pressure drop.

To size boxes when ΔTP is not catalogued, calculate the velocity pressure drop using the following equation:

$$\Delta TP = \Delta SP + \Delta VP$$

$$= \Delta SP + \left[\left(\frac{v_{in}}{4005} \right)^2 - \left(\frac{v_{out}}{4005} \right)^2 \right]$$

The velocity (v) at the box inlet and outlet are calculated by dividing the airflow rate (CFM) by the inlet and outlet area (ft^2), which in turn is determined from the dimensions listed in the catalogs.

Use a "dual maximum" control logic, which allows for a very low minimum airflow rate during low-load periods.

It is common practice in VAV box control to use control logic with a single maximum airflow setpoint (see Figure 6). A more energy-efficient VAV box control logic uses a "dual maximum" strategy. In addition to a minimum airflow setpoint and a cooling maximum airflow setpoint, there is also a heating maximum airflow setpoint, hence the name "dual maximum" (Figure 7).

Figure 6: VAV Hot Water Reheat Box Control—Single Maximum

A single maximum airflow setpoint is common practice, but it is less energy efficient than a dual maximum strategy.
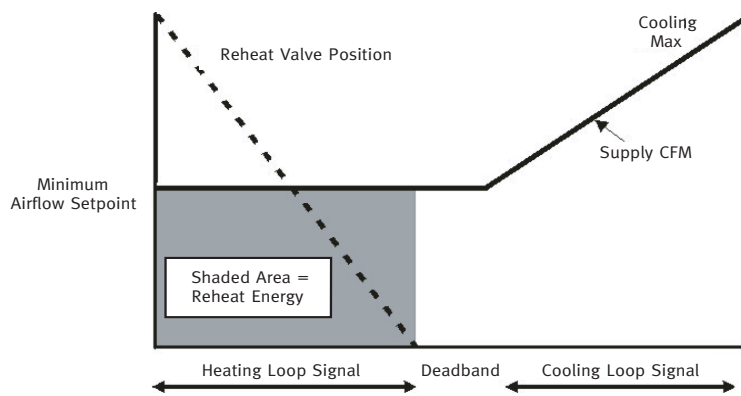
With the single maximum strategy, the minimum is typically set to the airflow required to meet the design heating load at a supply air temperature that is not too warm, e.g., ≤90°F. This flow is typically higher than the code ventilation requirement. With the dual maximum strategy, however, the minimum can typically be set to the ventilation requirement. This reduces both reheat energy and fan energy. By reducing the deadband minimum airflow rate, spaces are not overcooled when there is no cooling load and "pushed" into the heating mode. And by controlling the reheat valve to maintain discharge supply temperature rather than space temperature, supply air temperature can be limited so that stratification and short-circuiting of supply to return does not occur.

A diffuser designed for good mixing at design cooling conditions may "dump" at low flow. Dumping means that the air leaving the diffuser does not have sufficient velocity to hug the ceiling (known as the Coanda effect) and mix with the room air before reaching the occupied portion of the room. Instead, a jet of cold air descends into the occupied space, creating cold drafts and discomfort.

However, concern about dumping may be overblown. There are many buildings operating comfortably with lower than 30 percent airflow minimums. Researchers at U.C. Berkeley and Lawrence Berkeley National Laboratory performed several laboratory experiments with two types of perforated diffusers and two types of linear slot diffusers.[5] They measured air change effectiveness (using tracer gas) and thermal comfort (using thermal mannequins) in heating and cooling mode and at various flow rates (100, 50, and 25 percent turndown). They found that in nearly all cooling tests, thermal comfort was within the acceptable range and air change effectiveness was consistently at or above 1.0.

Figure 7: VAV Hot Water Reheat Box—Dual Maximum (Recommended Approach)

A dual maximum strategy has a heating maximum airflow setpoint in addition to a minimum airflow setpoint and a cooling maximum airflow setpoint. It is more energy efficient than the single maximum strategy.
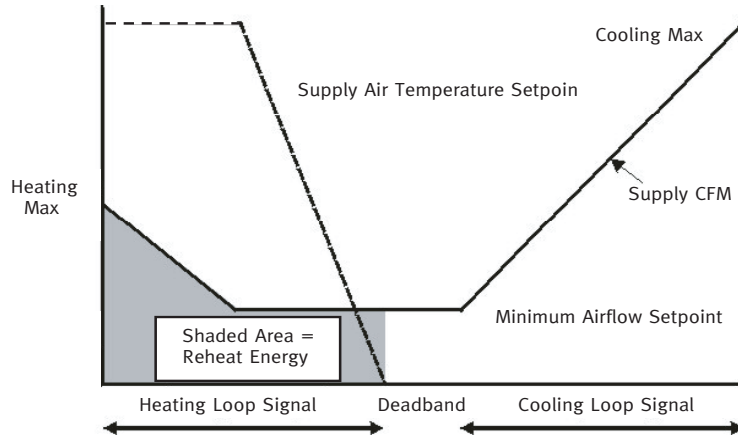
Set the minimum airflow setpoint to the larger of the lowest controllable airflow setpoint allowed by the box and the minimum ventilation requirement (often as low as 0.15 CFM/ft².) Title 24 places limits on both the lowest and highest allowable VAV box minimum airflow setpoints. With the dual maximum strategy described above, the minimum airflow setpoint need not be based on peak heating requirements. To minimize energy use while still complying with Title 24 ventilation requirements, the minimum airflow setpoint should be set to the greater of the minimum airflow at which the box can stably control the flow, or the ventilation requirement.

Duct Design

An ideal duct system would be designed to have the lowest lifecycle cost (LCC), perfectly balancing first costs with operating costs. However, it's impractical to rigorously optimize LCC because there are simply too many variables and unknown factors. Still, some general guidelines can help designers develop good designs that provide a reasonable, if not optimum, balance between first costs and operating costs.

Table 1. Summary of Sample Box Max and Min

Table 1 summarizes the turndowns for typical selection of VAV boxes. The column "Best turndown" is the ratio of the minimum cubic feet per minute (CFM) to the maximum CFM as if the box size were selected just at the maximum allowable flow rate. "Worst turndown" is the ratio of the minimum CFM for that box size to the maximum CFM of the next smaller box size as if the box had the smallest airflow in its size range.

These values for best and worst turndown represent the range of potential selections within a given box neck size and show that VAV box airflow can be controllable at 10 percent or lower airflow in most cases.

Nominal Size	Max CFM	Min CFM	Best Turndown	Worst Turndown
4	230	14	6%	n/a
5	333	23	7%	10%
6	425	33	8%	10%
7	580	44	8%	10%
8	675	58	9%	10%
9	930	73	8%	11%
10	1,100	91	8%	10%
12	1,560	130	8%	12%
14	2,130	177	8%	11%
16	2,730	232	8%	11%
Average				9%

Note: These values were developed using a controller/ sensor accuracy of 0.004 in. W.C.

Key Recommendations

Run ducts as straight as possible to reduce pressure drop, noise, and first costs. This is the most important rule. The straighter the duct system, the lower both energy and first costs will be. Try to keep the number of bends and turns to an absolute minimum.

Use standard length straight ducts and minimize the number of transitions and joints. Straight, standard length sheet metal ducts are relatively inexpensive, especially compared to the labor needed to connect pieces and seal joints. See Figure 8.

Use round spiral duct wherever it can fit within space constraints. Round duct is generally less expensive than oval and rectangular duct, especially when run in long, straight sections.

In the bottom design, four transitions are reduced to one, with each remaining duct section sized for multiples of the standard 5-foot rectangular duct length, which reduces both first costs and energy costs.
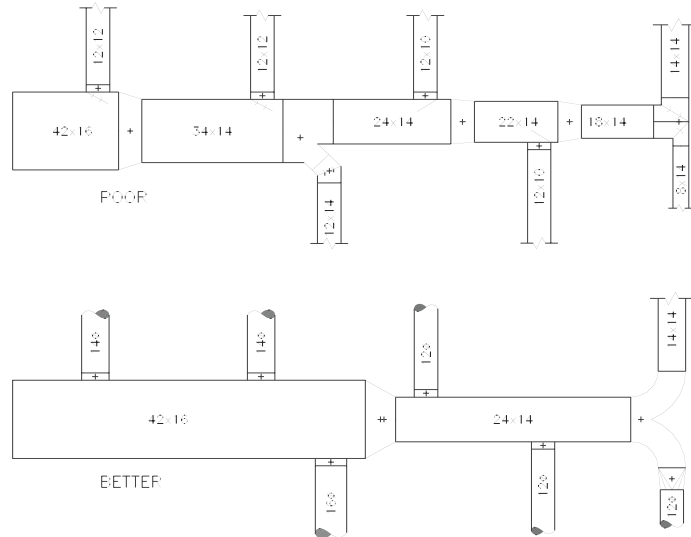
Flat oval duct is often the next best option when space does not allow the use of round duct. In general, limit the use of rectangular duct to situations where ducts must be acoustically lined, for duct sections containing many fittings, and for large plenums.

Use radius elbows rather than square elbows with turning vanes whenever space allows. Except for very large ducts, full radius elbows will cost less than square elbows with turning vanes, yet they have a similar pressure drop and much improved acoustics. On medium and high velocity VAV systems where a full radius elbow cannot fit, use a part-radius elbow with one or more splitters. Turning vanes should only be used on low velocity systems where radius elbows will not fit.

Use either conical or 45-degree taps at VAV box connections to medium pressure duct mains. Taps in low velocity mains to air outlets will have a low pressure drop no matter how they are designed. Inexpensive straight 90-degree taps (e.g., spin-ins) can be used for round ducts; 45-degree saddle taps are appropriate for rectangular ducts. Never use taps with extractors or splitter dampers; they are expensive, noisy, and increase the pressure drop of the duct main, which can increase fan energy.

Specify sheet metal inlets to VAV boxes; do not use flex duct. Sheet metal inlets will reduce pressure drop, improve airflow measurement accuracy, and reduce breakout noise from the VAV damper.

Avoid consecutive fittings because they can dramatically increase pressure drop. Two consecutive elbows, for instance, can have a 50 percent higher pressure drop than two elbows separated by a long straight section.

For VAV system supply air duct mains, use a starting friction rate of 0.25 to 0.30 in. per 100 ft. at the air handler. Gradually reduce the friction rate at each major juncture or transition down to a minimum friction rate of 0.10 to 0.15 in. per 100 ft at the end of the duct system. This design condition represents a reasonable balance between first costs (including the cost of sheet metal ducts plus the space required to house them) and energy costs, recognizing that VAV systems seldom operate at their design capacity.

For unducted return air shaft sizing, maximum velocities should be in the 800 fpm to 1200 fpm range through the free area at the top of the shaft (highest airflow rate). Gradually reduce the velocity down to about 500 fpm at the zone level. For systems with return fans, return air ducts are typically sized using the same technique used to size supply air ducts.

To minimize system effects, fans should discharge into duct sections that remain straight for as long as possible— up to 10 duct diameters from the fan discharge—to allow flow to fully develop. Where this is not possible, the effect can be minimized by orienting the fan so that an elbow close to the discharge bends in the direction of the fan rotation, or by discharging the fan into a large plenum and then tapping duct mains into the plenum with conical taps.

Use duct liner only as much as required for adequate sound attenuation. Avoid using sound traps. Duct liner is commonly used for sound attenuation. But use of duct liner
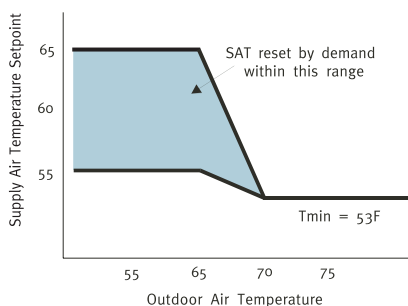
raises some indoor air quality concerns because it can be a breeding ground for microbial growth, may break down over time and be blown into occupied spaces, and can be difficult to clean. However, designing HVAC systems without duct liner remains a major—and often expensive—challenge. The best designs may be those that use duct liner only where needed for sound attenuation, that locate it in clean and dry areas, and that use protective facings to limit damage and erosion.

Supply Air Temperature Control

In most buildings, the optimal setting for supply air temperature varies over time, often from one hour to the next, and supply air temperature reset controls can provide significant energy savings. The optimal supply air temperature minimizes fan, cooling, and heating energy. But this is a fairly complex tradeoff, and the optimal setpoint at any point in time is not obvious.

Key Recommendations

The recommended control sequence for supply air temperature reset, as shown in Figure 9, is to lead with supply temperature setpoint reset in cool weather where reheat might dominate, and to keep the chillers (or compressors) off as long as possible, and then return to a fixed low setpoint in warmer weather when the chillers are likely to be on, and it is unlikely that any zones will require reheat. During reset, employ a demand-based control that uses the warmest supply air temperature that satisfies all the zones in cooling.

During occupied mode, the setpoint is reset from T-min (53°F) when the outdoor air temperature is 70°F and above, proportionally up to T-max when the outdoor air temperature is 65°F and below. T-max shall range from 55°F to 65°F and shall be the output of a slow reverse-acting PI loop that maintains the Cooling Loop of the zone served by the system with the highest Cooling Loop at a setpoint of 90 percent. In other words, the setpoint should be the highest temperature that can still satisfy the cooling demand in the warmest zone.

Optimal supply air temperature minimizes fan, cooling, and heating energy, but it's a complex tradeoff.

Figure 9: Recommended Supply Air Temperature Reset Method

Show recommended control sequence for supply air temperature reset. SAT = supply air temperature (°F).

Fan Selection

Fans account for a large part of mechanical-system energy use. In California's new commercial buildings with built-up HVAC systems, fans account for 1 terawatt-hour of electricity use per year, representing approximately 50 percent of all HVAC energy use.[6] Standard design and operating practices can lead to designs that use as much as twice the energy of optimized designs. This can be attributed in part to the lack of analysis of fan-system performance across the full range of operation, and to the lack of tools to perform such an analysis (see the sidebar, "Visualizing Fan Performance" for more about analyzing fan-system performance).

Table 2 shows factors to consider when selecting today's most common fan types for typical large VAV applications. In addition to these factors, redundancy must be considered; the primary

In new commercial buildings with built-up HVAC systems, fans account for approximately 50 percent of all HVAC energy use.

Table 2. Comparison of Common VAV Supply Fan Types

Fan Type	Typical Applications	First Cost	Space Constraints	Efficiency	Noise	Other
Notes:	Duty: CFM and static pressure at design conditions	More efficient fans are often more expensive	A tight space may limit fan choices	Varies greatly by type and sizing	Different fan types have different acoustical performance	Some fans are more likely to operate in surge at part-load conditions
Housed forward-curved (FC) centrifugal	CFM ‹25000 1" ‹SP‹3.5"	Lowest	Requires more space than plenum fans for smooth discharge	Least efficient than housed airfoil for ›3", better or same as housed airfoil, BI for ‹2"	Slower speed than housed airfoil so usually quieter	Small surge region; may be unstable for parallel fans at low airflow, high static
Housed backwardly inclined (BI) centrifugal	CFM ‹70000 2"‹SP ‹6"	Medium/Low	Requires more space than plenum fans for smooth discharge	Somewhat less than housed airfoil	Similar to housed airfoil	Larger surge region than housed airfoil
Housed airfoil (AF) centrifugal (double width).	CFM ‹100000 2"‹SP ‹8"	Medium	Requires more space than plenum fans for smooth discharge	Highest efficiency	Noisier than plenum and forward-curved	Small surge region; high shut-off pressure
Mixed flow	CFM ‹60000 2"‹SP ‹6"	Highest	Good for inline use	Similar to housed airfoil but drops off in surge region	Quieter than other housed fans	Small surge region
Plenum airfoil centrifugal	CFM ‹80000 2"‹SP ‹6"	High. Cost of discharge plenum also must be included	Requires least space, particularly for multiple fans	Lower efficiency than housed airfoil unless space is constrained	Quietest when plenum effects included	Large surge region

advantage of parallel fans over a single fan is that they offer some redundancy if a fan fails or is down for servicing. However, parallel fans can create space problems, are more expensive, and make fan control and isolation more complex.

Key Recommendations

Use housed airfoil fans whenever possible. Housed airfoil fans are usually more efficient than plenum fans, even if space constraints result in a poor discharge arrangement and system effects. The extra pressure drop on a housed airfoil fan has to be surprisingly high before it is less efficient than a plenum fan for the same application. However, in some applications, noise and space constraints may mean that plenum fans are the best choice.

An airfoil fan selected near its peak efficiency will stay out of surge longer than a plenum fan selected near its peak efficiency, because a housed airfoil fan's peak efficiency is well to the right of the surge line but plenum fans peak at the surge line.

Consider using a smaller fan if static pressure setpoint reset is not likely to be implemented. If there is a good chance that static pressure setpoint reset will be successfully implemented, fan sizing is fairly straightforward since fan efficiency remains fairly constant. If static pressure setpoint reset is not likely to be implemented (e.g., no direct digital controls, or DDC, at the zone level), consider using a smaller fan (i.e., lower efficiency at design condition) because it will stay out of surge longer and the efficiency will actually improve as it rides down the system curve.

Factor in discharge plenum, motor, and VSD costs when estimating first costs. A fair comparison of fan types for built-up systems should include the cost to construct the discharge plenum for plenum fans. Motor and VSD costs should also be considered since less efficient fans may require larger motors and drives.

Although plenum fans are quieter, steps can be taken to attenuate noise from housed fans. Plenum fans are inherently quieter than housed fans due to the attenuation of the discharge

plenum, and they work better with sound traps in cases where traps are necessary. A sound trap can be placed much closer to a plenum fan than to a housed fan. But don't rule out housed airfoil and other types of housed fans on the basis of noise. Instead, consider locating the air handler as far away from noise-sensitive spaces as possible. Use duct liner to attenuate noise. Or use a sound trap, if necessary, but only if it can be located at least three duct diameters downstream of the fan.

In the process of developing the Advanced VAV System Design Guide, the project team created a new simulation model of fan-system efficiency as a function of flow and pressure. This Characteristic System Curve Fan Model can be used to develop three-dimensional fan curves. These curves add fan efficiency to the z-axis on top of the pressure (y-axis) and volume (x-axis) of the manufacturer's curve. As Figure 11 shows, this makes it easy to see the breadth of the high efficiency region for the same plenum fan shown in Figure 10 across a range of operating conditions. More detailed information about this fan model can be found in the Advanced VAV System Design Guide, and in articles in HPAC Engineering and ASHRAE Transactions.[7]
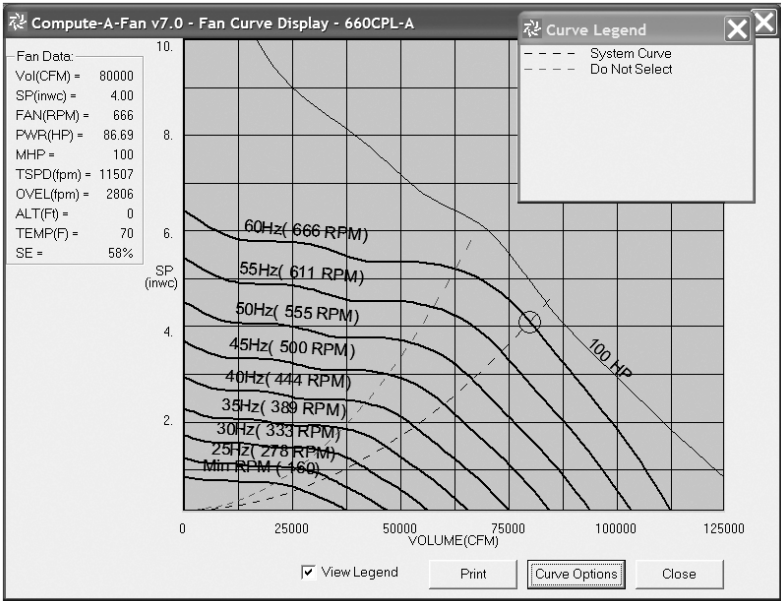
Figure 10: Typical Manufacturer's Fan Curve

Sixty-six inch plenum fan showing efficiency reported for a single operating point.

Figure 11: Three-Dimensional Fan Curve

Sixty-six inch plenum airfoil fan. Shows fan efficiency as a function of airflow and static pressure. EFF = efficiency, percent; DP = differential pressure; CFM = cubic feet per minute.
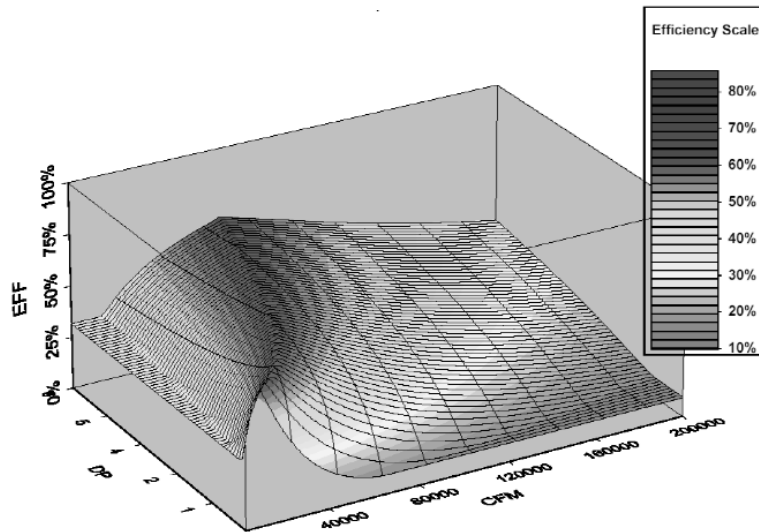
Supply Air Pressure Reset

How a fan is controlled is probably more important to fan energy than the type of fan selected. More specifically, demand-based static pressure setpoint reset has tremendous potential for saving energy and reducing noise, as well as reducing or eliminating fan operation in surge.

By far the most common and most efficient way of controlling medium to large VAV fans is with variable-speed drives (VSDs). Title 24 currently requires fans of 25 horsepower (HP) and larger to have either VSDs (or variable pitch blades for vane-axial fans), while the 2005 version is lowering this minimum to 10 HP.

Key Recommendations

Locate the static pressure sensor as far out in the system as possible. The location of the static pressure (SP) sensor can greatly affect the energy efficiency potential of a system when a fixed static pressure setpoint is used. The sensor should be as far out in the system as possible, with multiple sensors used if there are branches in the duct main. The design condition SP setpoint should be the minimum SP necessary to get the air from the

sensor location through the ductwork to the hydraulically most remote VAV box, through its discharge ductwork and air outlets, and into the space. The further the sensor is located from the fan, the lower the SP setpoint needs to be, and vice versa.

If the static pressure setpoint is reset (see below), the location of the sensor theoretically makes no difference since its setpoint will always be only as high as needed for the box requiring the highest pressure. However, it is recommended that it be located as far out in the system as practical to ensure proper operation if the reset logic fails.

Use demand-based static pressure setpoint reset to reduce fan energy, reduce fan operation in surge, reduce noise, and improve control stability. Demand-based SP setpoint reset has tremendous potential for saving energy, reducing noise, and reducing or eliminating fan operation in surge. But it can only be effectively implemented on a system with zone-level direct digital controls (DDC) and some signal from the VAV box controllers back to the DDC system indicating VAV box damper position. This signal may be either the damper signal if modulating actuators are used, or estimates of damper position based on timing open/close signals if floating actuators are used. A full-open position switch on the actuator may also be used, although with more zones not satisfied. The basic concept is to reset the static pressure setpoint in order to maintain the damper of the VAV box requiring the most static pressure at 90 percent open.

Static pressure reset requires careful loop tuning. The reset control loop must be very slow relative to zone airflow control loops because a change in static pressure has an immediate effect on VAV airflow and hence damper position.

Static pressure reset also relies on reasonably good agreement between box sizing and actual loads. If one particular box or branch duct is significantly undersized, that box may always be wide open and the zone undercooled, in which case no static pressure setpoint reset is possible. A single "rogue" zone or undersized box on a large VAV system could result in thousands of dollars of lost energy savings on an annual basis.

Demand-based SP setpoint reset has tremendous potential for saving energy, reducing noise, and reducing or eliminating fan operation in surge.

Even if DDC is available at the zone level and reset controls are to be used, the design SP setpoint must be determined in the field in conjunction with the air balancer.

Avoid operating fans in parallel at low flow, particularly if static pressure is not successfully reset. By dividing the flow in half, it pushes the fans into the surge region and can cause them to operate in particularly unstable areas within the surge region.

Isolate fans in parallel. Fans in parallel must be isolated, either with inlet cones, barometric back dampers, or motorized backdraft dampers. Any type of isolation will add static pressure to the fan system, although to widely varying degrees, and will have some leakage rate that will also increase fan energy. Motorized backdraft dampers also add controls complexity and should generally be avoided.

Plenum fans are best isolated with inlet cones. These have very low pressure drop when fully open and do not leak as much as backdraft dampers.

Coils and Filters

Selection of coils and filters needs to balance energy savings against first costs.

Key Recommendations

Use filters during construction. If air handlers must be used during construction, filtration media with a Minimum Efficiency Reporting Value (MERV) of 6 should be used to protect coils and supply systems. Replace all filtration media immediately prior to occupancy.

Avoid using pre-filters. They typically do not extend the life of the main filters because most dust passes through them. Also, pre-filters increase energy and maintenance costs.

Specify final filters with 80 to 85 percent dust spot efficiency (MERV 12). Maximum initial pressure drop at 500 fpm should not exceed 0.60 in. W.C. Filters should be changed long before they reach the maximum pressure drop indicated by the filter manufacturer. More frequent change intervals (for

example, once per year) are now being recommended by indoor air quality experts.

Utilize the maximum available area in the air handler for filters rather than installing blank-off panels. If the entire cross-sectional area of the air handler is filled with filters, then pressure drop will be reduced and filter life will be extended. The energy and maintenance savings can pay for the added first cost in a reasonably short period of time.

Use extended surface filters. These filters, which fit conventional filter framing, have a higher dust-holding capacity, longer life, and lower pressure drops than standard filters. They may cost more, but they will usually pay for themselves in energy and maintenance savings.

Consider selecting lower face-velocity coils, in the range of 400 to 550 fpm. Select the largest coil that can reasonably fit in the allocated space. Many designers select cooling coils for a face velocity of 550 fpm, but it is well worth looking at face velocities in the range of 400 to 550 fpm. Most air handler manufacturers offer multiple coil sizes for a given air handler casing size. Selecting the largest coil for a particular casing can significantly reduce fan energy and has little impact on first cost.

Consider placing a bypass damper between coil sections where the intermediate coil headers are located. Since this space is already allocated for piping, it provides a low-cost option to further reduce the fan pressure drop.

## Outside, Return, and Exhaust Air Control

Ventilation that meets Title 24 minimums is required for all spaces when they are normally occupied. Furthermore, the 2005 version of the Standard mandates that VAV systems be tested for code-required ventilation both at design supply airflow and with all VAV boxes at minimum position.

A fixed minimum position setting for the outside air damper is appropriate for constant volume systems (e.g., package single zone) but will not work for VAV systems. The *Advanced VAV System Design Guide* and ASHRAE Guideline 16[8] both describe

appropriate methods for maintaining minimum ventilation in VAV systems, including direct measurement of the outside air flow (Figure 12) and injection fans. While there are several methods to dynamically control the minimum outdoor air in VAV systems, the recommended method is to use a dedicated minimum ventilation damper with pressure control (Figure 13).

Figure 12: Airside Economizer Configuration with Barometric Relief

In addition to energy savings, an advantage to barometric relief is the simplicity of controls for building pressurization, since no automatic control is required. A distinct disadvantage is that it only works for low-pressure returns, typically limiting it to low-rise projects. Source: ASHRAE Guideline 16–2003.[9]
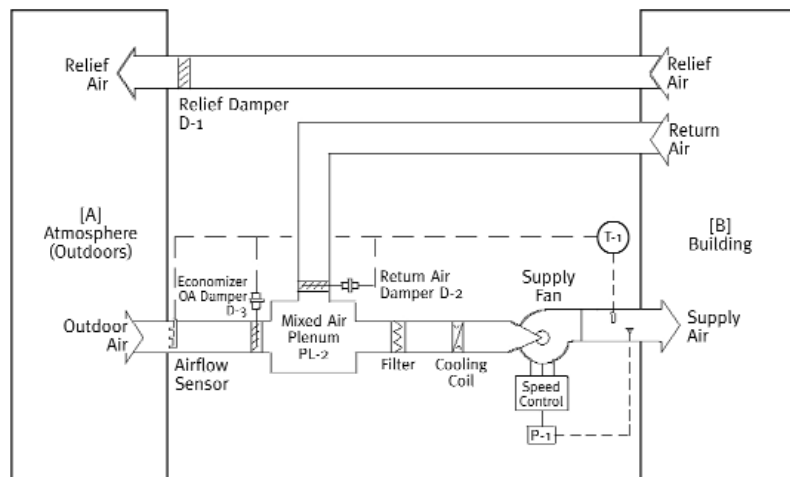
Key Recommendations

For outdoor air control, use a dedicated minimum ventilation damper with differential pressure control. This is an inexpensive and effective design that involves using a two-part economizer damper: a small two-position damper controlled for minimum ventilation air, and a larger, modulating, maximum outdoor air damper that is used in economizer mode. The basic concept is that differential pressure across a fixed orifice can be accurately correlated to airflow. The minimum outdoor air damper in the open position is the fixed orifice. The pressure drop across it at design minimum flow rate is determined in the field. The return air damper is then used to ensure the correct negative pressure is maintained in the mixed air plenum.

Where barometric relief is not an option, relief fans are the best bet. Relief fans always use less energy than return fans and can incorporate barometric relief as the first stage of building pressure control. In addition to the energy benefits, relief fans are relatively compact. Noise can often be controlled by placing the relief fans out of the line of sight from the return shaft. Source: ASHRAE Guideline 16–2003.[10]
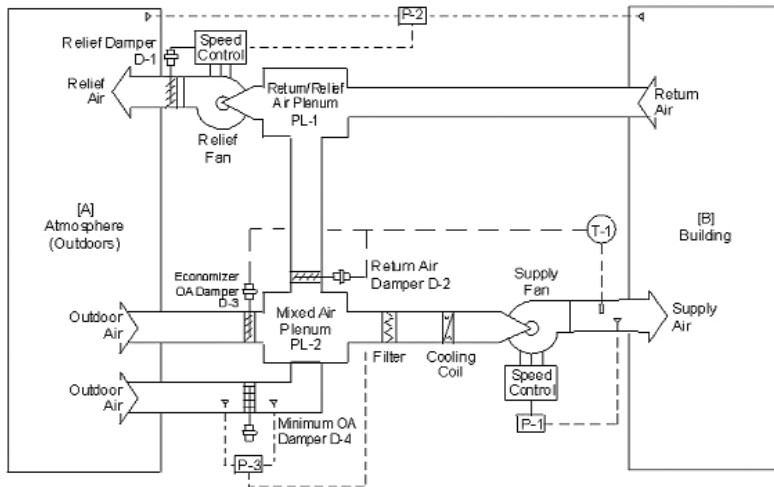

Use barometric relief if possible; otherwise, use relief fans rather than return fans. Common to all airside economizer systems is the need to relieve up to 100 percent design airflow minus anticipated exfiltration and building exhaust, due to the fact that the economizer could be providing up to 100 percent outdoor air. Economizers can be designed with barometric relief (Figure 12), a relief fan or fans (Figure 13), or return fans.

From an energy standpoint, barometric relief is the most efficient, relief fans are less efficient, and return fans are the least efficient. Systems with high return pressures (e.g., ducted returns) will generally require return fans. Large unducted return systems should use relief fans, while small unducted return systems can get away with barometric relief. To work effectively, barometric dampers must be chosen for low pressure drop (typically a maximum of 0.08 in. W.C. from the space to ambient) at relatively high flow rates. As a result, the barometric relief openings can be excessively large—a challenge to the architectural design.

In this staged sequence, the outdoor air damper is opened as the first stage of cooling, while the return damper remains open (provided that the economizer is operating). This sequence provides less than 100 percent outdoor air but a very low-pressure air path for the supply fan. If this is sufficient to cool the building, energy savings will result from the reduced fan pressure. If more cooling is needed, the return damper is modulated closed to ensure that the system has 100 percent outdoor air. Source: ASHRAE Guideline 16–2003.[11]
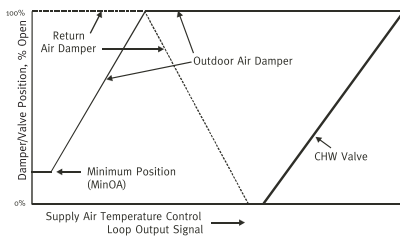
For economizer control, sequence the outdoor and return air dampers in series rather than in tandem. Most economizer control sequences stage the outdoor and return dampers in tandem, with the return dampers closing as the outdoor dampers open. Although this sequence works, fan energy can be saved by staging these dampers in series (Figure 14).

Specify differential drybulb control for economizers in California climates. Title 24 has requirements for economizer high-limit switches, which is the control that disables the economizer when the outdoor air is warmer (or has higher enthalpy) than the return air. While there are a number of different types of high-limit controls, dry-bulb temperature control is the most robust, as dry-bulb temperature sensors are easy to calibrate and don't drift excessively over time.

Differential control is recommended throughout California. The sensors should be selected for a through-system resolution of 0.5°F. Dry-bulb sensors work well in all but humid climates, which are not typical in California.
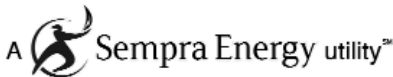
Notes and Additional Information

1   Eley Associates and Taylor Engineering. 2003. *Advanced Variable Air Volume System Design Guide*. California Energy Commission Publication #500-03-082-A-11. This detailed report on VAV system design was developed for California Energy Commission's Public Interest Energy Research (PIER) program and is the primary source document for this Design Brief. Available at www.newbuildings.org (see "PIER") or www.energy.ca.gov/pier/reports/500-03-082.html.

2   For complete simulation assumptions and results, see Appendix 6 of the *Advanced VAV System Design Guide* (ibid.).

3   This example is by no means comprehensive. For example, these savings do not include the impact of reducing duct pressure drop through careful design, the impact of properly designing 24/7 spaces and conference rooms, or the potential savings from demand-based ventilation controls in high density occupancies. The assumptions in this example are presented in Appendix 6 of the *Advanced VAV System Design Guide* (ibid.).

4   For the Design Guide, see New Buildings Institute and California Energy Commission websites above. For the Energy Design Resources Design Brief, "Improving Mechanical System Energy Efficiency Through Architect and Engineer Coordination," see www.energydesignresources.com.

5   W.J. Fisk, D. Faulkner, D. Sullivan, and F.S. Bauman. 1997. "Air Change Effectiveness And Pollutant Removal Efficiency During Adverse Conditions." *Indoor Air*; 7:55-63. Denmark: Munksgaard. Also: Fred Bauman, Charlie Huizenga, Tengfang Xu, and Takashi Akimoto. 1995. "Thermal Comfort With A Variable Air Volume (VAV) System." Center for Environmental Design Research, University of California, Berkeley, California. Presents research on ADPI for diffusers over a range of flows.

6   Mark Hydeman and Jeff Stein, "A Fresh Look at Fans,"
    HPAC (Heating/Piping/Air Conditioning) Engineering,
    May 2003: 28–41.

7   Hydeman and Stein (ibid.). Also: Jeff Stein and Mark
    Hydeman, "Development and Testing of the Characteristic
    Curve Fan Model," *ASHRAE Transactions*, January 2004.

8   ASHRAE Guideline 16–2003, Selecting Outdoor, Return,
    and Relief Dampers for Air-Side Economizer Systems,
    www.ashrae.org.

9   Ibid.

10 Ibid.

11 Ibid.

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