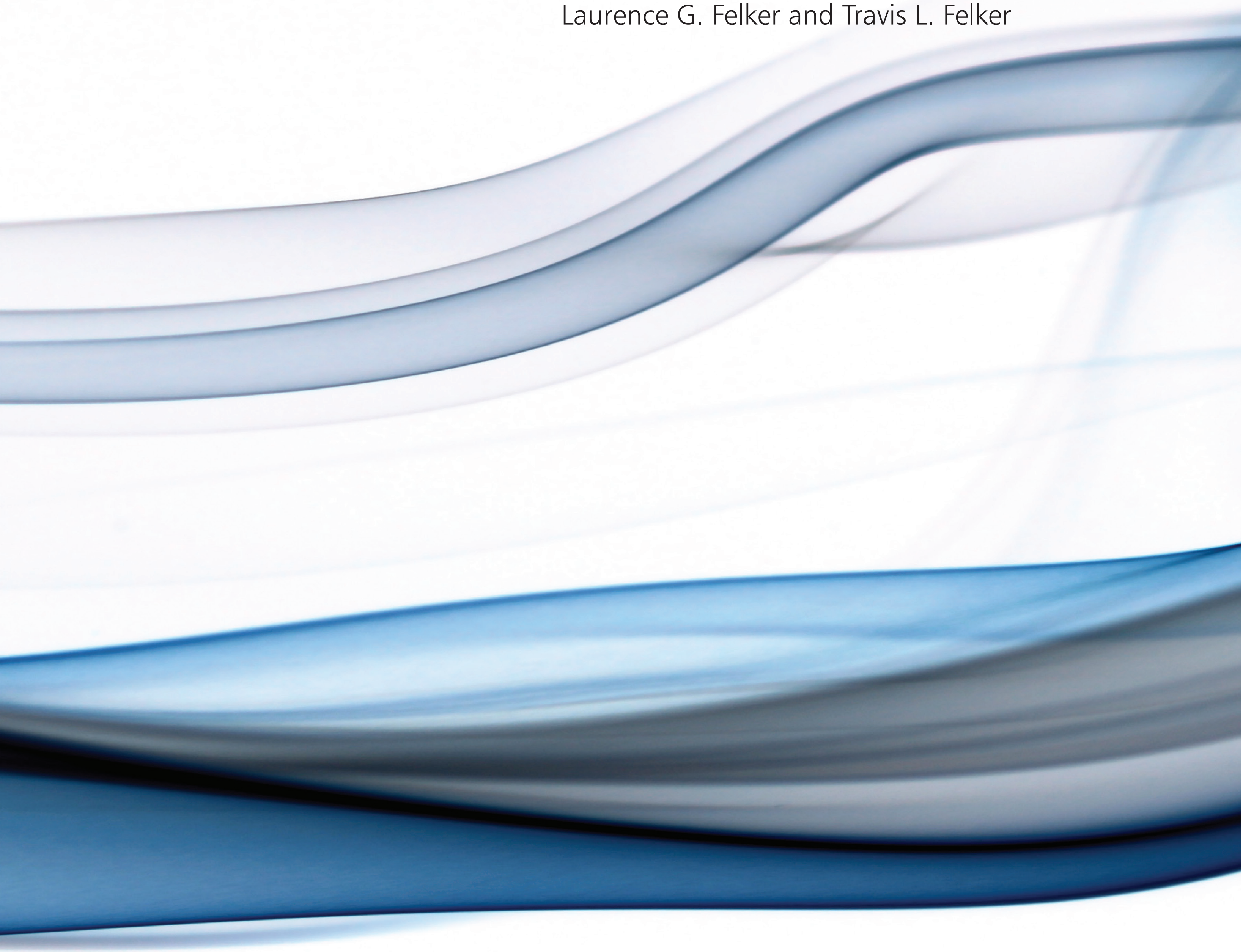
Dampers and Airflow Control

Laurence G. Felker and Travis L. Felker



American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Dampers and Airflow Control

About the Authors

Laurence G. Felker, a mechanical engineer, has worked as an electrician, HVAC service technician, and mechanical system designer. He has 19 years of experience designing pneumatic, analog electronic, and direct digital control systems. Since 1991 he has specialized in actuated damper and valve applications for Belimo Americas. He has been a member of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) since 1978 and is a member of the International Code Council (ICC) and the National Fire Protection Association (NFPA). His previously published work includes *Evans Equations of Unified Field Theory* (arima publishing), a theoretical physics book.

Travis L. Felker earned a bachelor of science degree in mathematics and worked for Belimo Americas for two years. Presently, he is working on his master's degree in education and teaching junior college.

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Laurence G. Felker
Travis L. Felker

American Society of Heating, Refrigerating and
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Atlanta

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In memory of Laurence Patrick Felker, 1919–1995

Paperboy

Chimney sweep

WWII halftrack gunner

Husband

HVAC controls sales manager

Father

Grandfather

Contents

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Introduction

GOAL OF THIS BOOK

Fans, duct systems, duct elements (such as filters and coils), dampers, and actuators all work together to control airflow. This text provides resources for building good judgment of the engineering principles needed to size, select, install, and adjust control dampers. Mechanical designers; mechanical and control contractors; and testing, adjusting, and balancing (TAB) contractors are the target audience of this book. Each specializes in a different part of the heating, ventilating, and air-conditioning (HVAC) system. Some material presented here is already familiar to each of these groups, and some material falls outside their normal areas of concern. For example, the information on loss coefficients and pressure loss is basic to the mechanical engineer; however, it is unfamiliar to most control contractors. Much of the subject material bridges gaps that exist between disciplines. The gaps observed in air systems are as follows:

- Lack of understanding of the final control element—the damper
- Lack of understanding of the detailed methods of the other trades on a project
- Lack of understanding of the complexity of airflows

There are many articles published regarding, for example, indoor air quality (IAQ) that say *what* actions to take. However, there are few suggestions for *how* they should be taken. This book addresses how to apply dampers within systems to achieve clearly defined goals.

The control of airflow in any commercial building is necessary for a variety of functions:

- Temperature control
- IAQ (ventilation)
- Envelope pressure control (mold and infiltration control)
- Elevator door operation
- Exterior door operation
- Exhaust
- Makeup air
- Pressure cascade from one zone to another
- Mitigation/control of chemical, biological, and radiological (CBR) release
- Smoke containment using shaft and wall dampers
- Smoke control using pressure differentials
    - Atria and large-space smoke extraction
    - Stairwell pressurization
    - Egress corridor smoke control
    - Zone smoke control

Fans and dampers are the primary control devices for airflow management. Fans are adequately covered in many available technical publications, such as those published by the Air Movement and Control Association (AMCA 2002). However, this is the first book written about dampers. Many articles and manufacturer bulletins exist, and product information is available from manufacturers, but nothing is comprehensive.

The principles explained are applicable to all damper applications. Fire and smoke control texts offer good insight into airflow strategy and tactics and are sources of information not available elsewhere. For more information, see Chapter 16.

PROBLEMS

Studies of building tenant complaints by the International Facilities Management Association indicate that the number-one problem is temperature control (IFMA 2003). Other studies show that the number-one cause of IAQ complaints is the ventilation system (Seppänen et al. 2002; NIOSH 2007). While people do not complain on a daily basis about IAQ, they do complain about temperature. A system not controlling temperature well is almost inevitably not controlling ventilation adequately to protect tenants from health problems. Nor are the owner, architect, and mechanical consultant protected from lawsuits (Katz and Portner 1993).

ANSI/ASHRAE Standard 62.1-2007, Ventilation for Acceptable Indoor Air Quality (ASHRAE 2007a), focuses on ventilation. *ANSI/IESNA/ASHRAE Standard 90.1-2007, Energy Standard for Buildings Except Low-Rise Residential Buildings* (ASHRAE 2007b), provides guidance on energy use. Application of each standard depends on airflow. When applying ASHRAE Standard 62.1, particularly in sophisticated applications like dynamic reset (Stanke 2006), it is necessary to ensure the dampers can perform correctly.

Seventy percent of the total number of commercial buildings are owner occupied. The owner and occupants directly benefit from IAQ; however, building owners have limited finances, and it is normal to consider first cost as a primary concern, meet minimal code requirements, and ignore life-cycle costs. An engineer typically works with time, financial, code, and standard constraints, and the mechanical designer has to do more work at the same pay to meet a high level of standards. Efforts by the U.S. Green Building Council (USGBC 2007) are a step in the right direction toward bridging the gap between building costs and good IAQ. Knowledge of dampers also helps engineers and mechanical designers avoid problems and contributes to optimal design and cost efficiency.

Poor temperature control causes lost productivity and wasted energy. Inclusive formal surveys are not available on built-up systems; however, performance of most packaged equipment is suspect. For example, according to the California Energy Commission "Small System Design Guide" study,

> Economizers show a high rate of failure in the study. Of the units equipped with economizers, 64% were not operating correctly. Failure modes included dampers that were stuck or inoperable (38%), sensor or control failure (46%), or poor operation (16%). The average energy impact of inoperable economizers is about 37% of the annual cooling energy (Jacobs et al. 2003).

In the authors' experience, about 15% of buildings have airflow problems that lead to bad temperature control, IAQ deterioration, space pressurization irregularities, and wasted energy. Another 20% can limp along for years with moderate troubles and property owner indifference. Seasonal difficulties are common.

There are no airflow or pressure control contractors. The temperature control and TAB contractors are responsible for airflow and pressure control, but first they require a solid mechanical engineering design. The control contractor tends to think first of the sensor and logic portion of his system and last about the final control element. There is little information available on how and why to size dampers. Both valves and dampers are fluid-flow control devices, and proper sizing requires a mechanical engineering approach.

It is not atypical for engineers to design the mechanical system and then overlay the control system. The two must be designed together. The controls can fine-tune a good mechanical design; however, they must be included as a part of the design for maximum effectiveness. Controls cannot compensate for many mechanical design problems.

Since each building is unique, and the mechanical system must work the first time, design is critical. Of course, the space, money, and time constraints placed on the mechanical designer are sometimes severe and shortsighted on the part of the architect and owner if long-term quality is a goal.

IMPORTANCE OF DAMPERS AND ACTUATORS

Dampers are the final control devices for almost all airflow in HVAC systems. Actuators are the interface between the control system and the mechanical system and are critical to accurate control.

Typically, 80% or more of direct digital control (DDC) outputs in the HVAC portion of the system go to actuators. If they are not positioned accurately, then all other portions of the air system suffer. Too often, the torque necessary to move a damper is the only selection criterion; accuracy is not an important issue. A controller cannot compensate for incorrect damper or actuator sizing. The TAB contractor or commissioning authority can only sometimes set airflows to correct for problems.

Concentration on the final control elements in the design process can solve most of these problems. Equipment sizing is very rarely a problem. The airflow system delivering most of the conditioned air to the occupants causes the difficulties. Factory-built mixing boxes control flow with difficulty due to space constraints. The willingness of design engineers to design around some space constraints leads to systems that work only part time.

AIRFLOWS IN A BUILDING AND SYSTEM

As the size of a project grows, there is more complexity and more chance for small miscalculations to become a major problem. New design ideas present a greater chance of problems. Figure I-1 shows some of the following sources of airflows that affect space conditions:

- Wind effect on intakes and exits (overflow and underflow occur periodically)
- Damper leakage
- Duct leakage—both return and supply
- Infiltration
- Exfiltration
- Cross flows from air handlers or multiple-tenant equipment
- Doors—both between internal spaces and to the outside

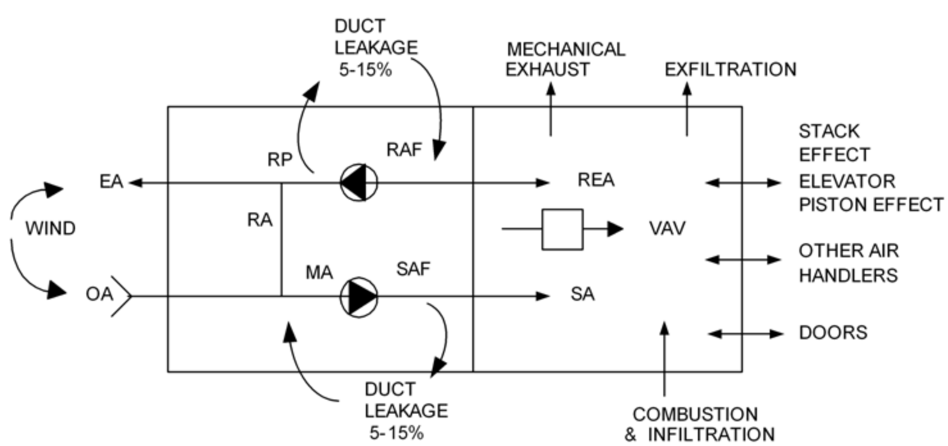

Figure I-1 Airflow paths in a typical system.

- Elevator shafts
- Exhaust fans
- Plenum vs. ducted returns
- Relief-exhaust vs. return-air fans
- Ventilation air quality
- Heat-recovery equipment
- Flows in voids and hidden spaces
- Fans
- Stack effect and space or zone static
- Ventilation effectiveness (fresh air must be distributed well)
- Cross contamination of exhaust and intake (potentially a major problem)
- Variable-air-volume (VAV) box flow variation (typically, while the supply varies by location, the returns do not vary; this can result in irregular flow patterns)
- Leakage paths through ceilings and floors
- Leakage paths through pipes, chases, shafts, and any penetrations
- Unknown variables and factors

KEY TO TERMS

Our attention to airflow details is poor enough that we have not defined terms for all distinct duct flows. The tee where recirculation and exhaust air diverts to exhaust air and return air is the return-relief plenum. All the recirculation and relief air for an air-handling unit (AHU) flows through this point, yet it does not have a commonly accepted name. This indicates the weakness of hydraulic analysis we give to economizer duct systems. Relief air is used interchangeably with exhaust air; although, possibly, *exhaust* should be a term reserved for local exhaust or powered exhausts. Clarity in terminology is necessary for clear thinking.

The term *recirculation and exhaust air* could be used instead of return air. It diverts at the return plenum into relief air and recirculation air. In this book, each term is used to reflect actual practices.

AHU	Air-handling unit
DA	Discharge air—discharge from air handler going to supply diffusers or VAV boxes
EA	Exhaust air
IGV	Inlet guide vane
MA	Mixed air—part outdoor and part recirculated from the space
OA	Outdoor air
RA	Recirculation air or, alternately, return air
RAD	Return-air damper
RAF	Return-air fan
REA	Recirculation and exhaust air—part goes back to the AHU, and part is relieved.
REF	Relief-exhaust fan
RFA	Relief air
RP	Return-relief plenum—diverting tee in the return-air path
SA	Supply air to space
SAF	Supply-air fan
VAV	Variable air volume
VFD	Variable-frequency drive

TYPICAL PRESSURES IN AIR-HANDLING SYSTEMS

Figure I-2 shows the typical pressures in several types of AHUs. Damper sizing sometimes requires different methods, depending on the system. One must match system pressures with control methods and damper performance.

When there is only an SAF, the system has little flexibility. An exhaust damper is useless as an exhaust until the RAD closes, because the RP is negative due to the supply fan. The pressure loss in the ducts and space eats up most of the pressure on the discharge side of the SAF. Systems with this geometry have air coming in the exhaust until the RAD closes. Local

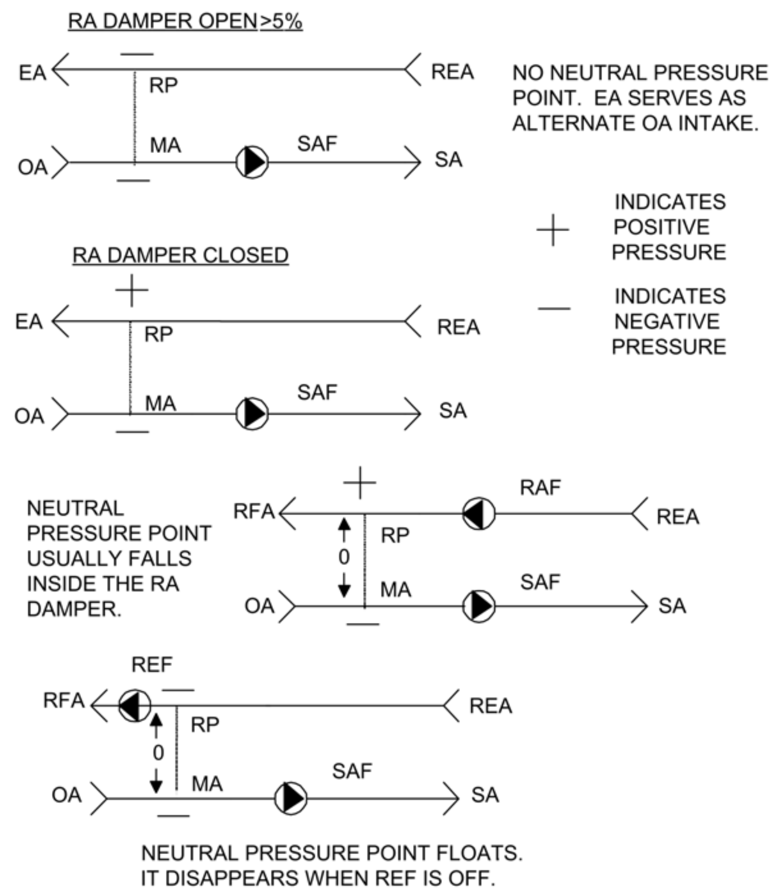

Figure I-2 Pressure points in a typical air-handling unit (AHU).

exhausts may help in economizer mode; however, the system overall is adapted for neither economizer operation nor space pressure control.

When there is both an SAF and an RAF, the RP is positive in all damper positions (unless the supply overpowers the RAF in error). The MA is negative; the value varies with the damper positions. When the OA damper is fully open, there is little pressure loss in that path. When the RAD is fully open (and the OA damper is closed), the RAF pushes air into the MA duct in some cases. The MA could go positive if the fans are unbalanced.

Typically, the MA is negative and the RP is positive. The balance or neutral pressure point where the static pressure is zero is often inside the RAD itself. The velocity pressure is always positive, since air is moving. The pressures at various points vary as the economizer dampers and VAV dampers proportion.

With an REF, there is no neutral pressure point when the REF is OFF. The SAF pulls the plenum negative. When the REF is ON, both the SAF and REF pull the duct negative until the RAD is closed. The balance point is within the RAD when it is closing; however, the neutral pressure point floats around in some cases. This is a difficult system to control.

The exact location of the neutral pressure point is not critical as long as the rest of the system is operating correctly.

Designing the fans' static pressures with respect to the duct and damper pressure losses at the various velocities is necessary. Given the unknowns, it is necessary to provide methods to balance the system. Balancing dampers and the use of VFDs help control airflow.

A hydraulic analysis is always necessary to know what is going to happen inside an AHU. A built-up unit is easier to analyze since the components are known. A factory-built unit has the external static defined in the specifications; however, the dampers' response curves are unknowns.

A significant effort is being made by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) and the federal government to reduce energy consumption. The concept of designing buildings that have a net zero energy use has gained traction. Development of ultra-efficient insulation, energy recovery, construction methods, equipment, and control tactics, along with solar and wind generation, could make concept a reality in some cases. The first steps toward net zero energy use buildings have been taken (ASHRAE 2008a).

New control and damper methods may reduce pressure loss and, thus, energy loss. In this book, an increase in damper pressure loss to balance part of a system is one basic tactic. This is best for control; it is not best for energy use, although the increase may be very slight.

Methods that are more sophisticated are applied after the basics are understood. One such method, untried and untested, is presented in Chapter 15, "Control Systems." DDC is required for sophisticated control. Most economizer and thermostat systems are simple analog and digital output devices. While they are much more flexible than old pneumatic controls, they are not as desirable as we can imagine. Once set, they are difficult to fine-tune. Canned programs and tactics are desirable so that the control contractor programmer does not have to reinvent the wheel with each system setup. Yet canned programs accept average methods and do not fine-tune for the idiosyncrasies that are normal in every system. It costs money to have the consultant and contractors review, fine-tune, continuously monitor, and recommission buildings. There is a trade-off between energy saved and cost of labor. Apply specialist knowledge to gauge the value of different actions.

OVERVIEW

This book concentrates on outdoor, recirculation, and relief damper operation in air-handling systems. This includes operation for economizers, OA intake for ventilation, and space pressurization. Apply the principles used for the economizer dampers to all similar applications. The principles apply also to VAV systems, refrigeration condensers, generators, relief dampers, and other simpler single-damper applications.

While the term *economizer* refers to the use of OA for free cooling, the ventilation minimum is maintained by the same controls, actuators, and dampers in most cases, and the two functions are usually inseparable. In this book, the word *economizer* refers to both functions unless specifically separated. Dampers operate in conjunction with other duct elements in order to obtain good airflow control. Control airflow by sizing and selecting dampers with respect to duct systems, by applying actuators and control loops properly, and by coordinating damper control in the entire system. It is necessary to develop engineering judgment based on various associated topics.

The *2009 ASHRAE Handbook—Fundamentals* (ASHRAE 2009) covers infiltration, ventilation, duct and fitting losses, stack effect, and other basics. The Sheet Metal and Air Conditioning Contractors National Association publishes material covering areas not covered completely elsewhere and some of the explanations are more basic (SMACNA 2002a, 2002b). Smoke-control texts provide strategy and tactics not covered elsewhere. The AMCA offers an excellent base of information that is well worth studying (AMCA 1986, 2002, 2003, 2006a).

There has been a scarcity of damper testing information in the past. Several ASHRAE-sponsored projects have addressed this in part. RP-1157, "Flow Resistance and Modulating Characteristics of Control Dampers" (Van Becelaere et al. 2004), tested the proportional characteristics of a variety of dampers in different geometrical arrangements and provided new and more accurate data, which are used extensively in this book.

1 Mechanical System Design and Airflow

ECONOMIZER FUNCTION

Airflow in any system can be complex. Figure 1.1 shows the primary airflows that require evaluation. Proper design of the economizer system includes consideration of the control of all airflow in a building. The distribution system is critical to local control; however, without good central flow control, the zones cannot function well. One cannot compensate for the other's failings; they work in conjunction.

The function of an economizer is to provide cooling at an economical cost. When outdoor-air (OA) enthalpy is below that of the return-air (RA) enthalpy, it completely or partially cools a building, depending on the required discharge-air (DA) enthalpy. Some special conditions may exist, for example special filtration requirements (Sorell 2007).

At any demand condition, the required quantity of system airflow is about the same regardless of whether the system is at minimum OA plus recirculation or at 100% economizer. Operation of the economizer system should not change the air volume. Exceptions exist. In unoccupied mode, a lower volume of air can adequately condition a space. A sophisticated system may use colder night air to subcool a space.

The minimum OA ventilation requirement is factored into consideration of economizer design (see Chapters 9 and 11). The design of simpler small systems and packaged equipment that have only a supply fan and allow space pressurization to float may use some of the principles explained here; however, the controls are generally inadequate for accuracy. Built-up systems are larger and require the most care.

Almost all of the air in a building travels through the economizer system and affects all the flows in the system.

MECHANICALLY BALANCED SYSTEMS

In Figure 1.2A, the dampers have under-the-linear flow characteristics, and the total flow is reduced as the dampers modulate to midposition. In Figure 1.2B, the dampers have over-the-linear flow characteristics, and the total flow increases as the dampers modulate to midposition. Figure 1.2C shows the design goal. The damper modulation has relatively little effect on airflow quantity (Belimo 1994).

The respective ratios of losses between the economizer section and the supply and return sections are particularly important. The more loss in the economizer system, the more the fans move about on their curves due to unavoidable nonlinearities in damper response curves. The more loss there is in the supply and return sections and the more vertical the fan curve at the operation point, the more self-correction there is in the system and the less the economizer dampers affect total flow.

A number of problems occur due to the failure to have a constant flow when the economizer functions. If the actuation is too fast, hunting and interacting control loops can occur. The fan volume control (typically a variable-frequency drive [VFD]) then responds when the

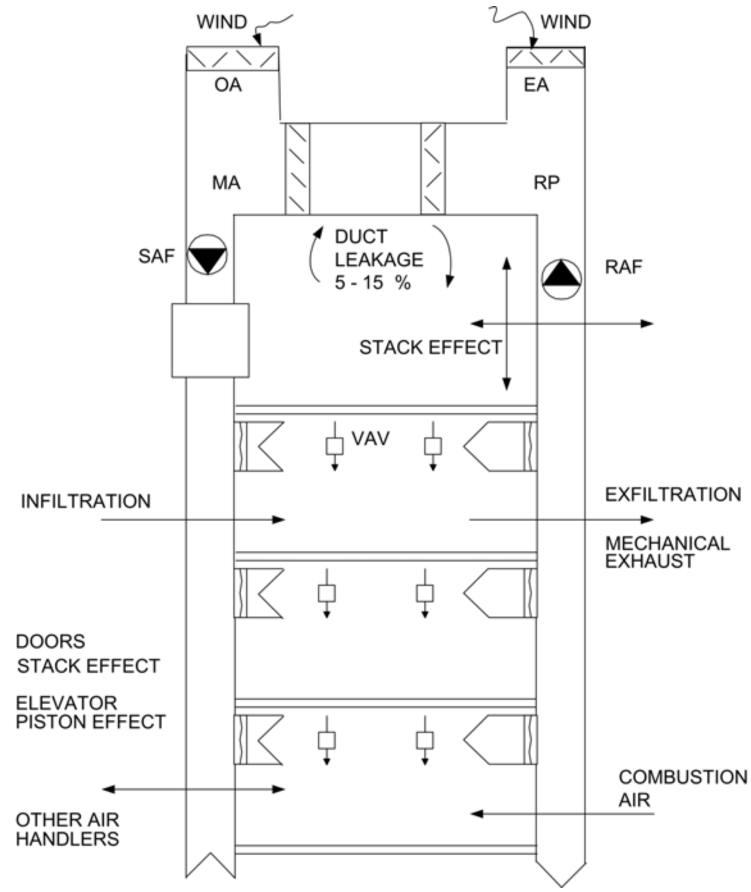

Figure 1.1 Some airflow paths.

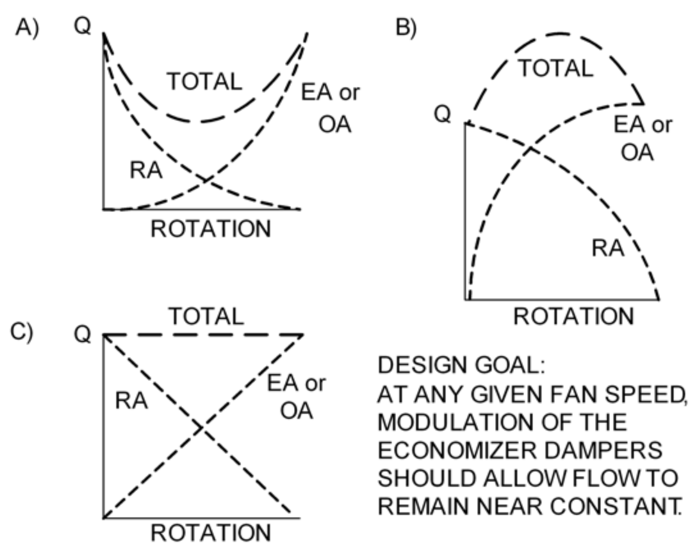

Figure 1.2 Under, over, and design airflow.

dampers move, since pressure and volume change in the main ducts. Oscillation of loops can occur. Temperature control may be poor. The volume changes at variable-air-volume (VAV) boxes can also cause them to hunt.

When the economizer portion of a system accounts for very little of the pressure loss seen by the fans, then the dampers can modulate with varying pressure loss and cause little system-wide effect (Figure 1.3, line B). The system curve meets the fan curve at the design point. This point does not move much. This is typically the case when the recirculation duct is less than 20% of the pressure loss of the combined return-air fan (RAF) and supply air fan (SAF) losses.

When the economizer portion of a system is a significant amount of the total pressure drop, nonlinearities in the dampers have more effect on the total airflow quantity.

When the economizer accounts for a substantial amount of pressure loss as compared to the total of the system's losses, then the system curve moves from the design point enough to cause changes in flow volume, as indicated in Figure 1.3, lines A and C. With a plenum return and low pressure losses in the ducts and coils, the recirculation duct could be as much as 30% of the RAF's loss. In this case, deviation from the linear by the exhaust air (EA) and return-air dampers (RADs) can cause high variation in return airflow. The system must be balanced at full recirculation, full economizer, and in between.

Figure 1.4 shows the fan curves in full economizer mode and when they are in series in full recirculation mode. The point is that, for any given set of fan speeds, $Q_1 + Q_2$ must be near constant regardless of the damper operation mode—full economizer or some midpoint or full recirculation. If this is accomplished at design, then it is also accomplished at all other fan speeds unless the fan curves are so different that the differential pressures vary appreciably.

Whether the system is in full economizer mode or in recirculation with minimum OA, the volume of airflow should be nearly the same. In Figure 1.5, setting the pressure loss from OA to mixed air (MA) and from EA to return-relief plenum (RP) to be equal to the pressure loss from RP to MA at the design quantity or velocity accomplishes the goal. This sets the endpoints of operation as noted in Figure 1.2.

The points RP and MA become near-constant pressure points for any given set of fan speeds regardless of the dampers' positions when dampers are linearized and flow is near constant.

Typically, the design engineer sets these flows roughly and expects the balancer to fine-tune them. This is often impossible without balancing dampers. The fan pressures are set for only one condition—minimum OA with recirculation (SMACNA 2002a). It is not certain that a system will balance if the designer cannot envision how to do it.

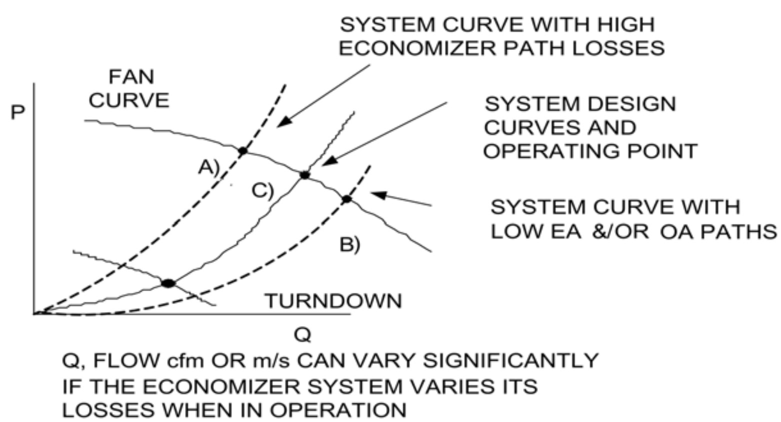

Figure 1.3 Variation in flow possible due to economizer damper positions.

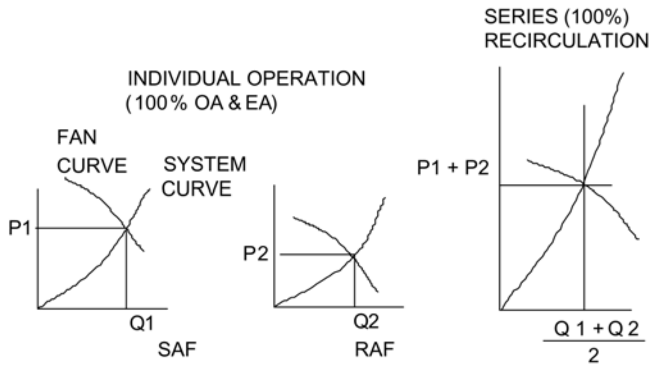

Figure 1.4 System curves in parallel and in series.

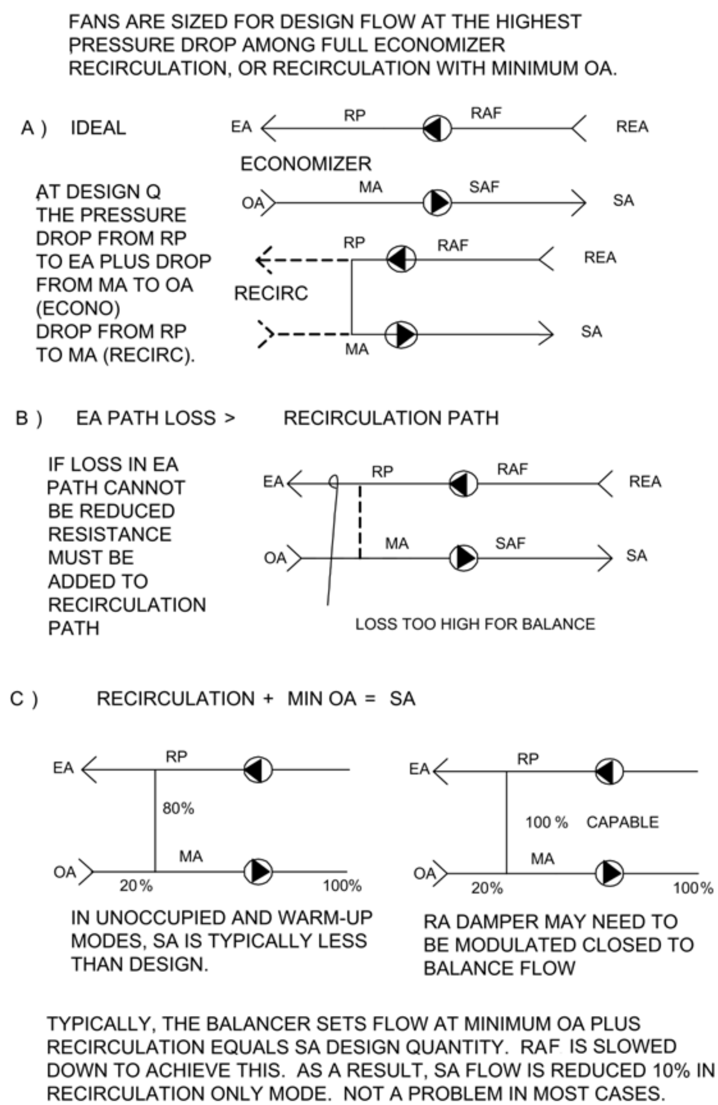

Figure 1.5 Fan and duct sizing and flow quantity relationships.

The design process has two steps here: (1) set the pressure drops so that the volume based on duct size and air velocity is constant in full economizer and full recirculation and (2) linearize the dampers so the intermediate operating points do not cause high variations. The next chapters detail this latter process.

VARIABLE AIR VOLUME (VAV) VERSUS CONSTANT VOLUME (CV)

Regarding duct sizing and economizer section, there is little difference between VAV and constant volume (CV). If a return fan system operates as described above, it does not matter whether it is VAV or CV. At any given set of fan speeds, the system behaves as if it were a CV system. Some complexity must be met in assuring that the fan speeds stay in synchronization; otherwise, the systems are the same (Kettler 1998).

Multistory buildings experience stack effect (see Chapter 10 for more information). When the OA is colder than the indoor air, the warm indoor air pushes up and pressurizes upper-floor spaces and can make the point RP very high positive, and MA can be positive if the neutral pressure point moves into the MA chamber. In summer, the cold indoor air has reverse stack effect. The upper floors are negative, and RP in air handlers on those floors can be negative due to both the SAF and reduced space pressures. The RAF must be able to adjust for conditions.

Sound mechanical design allows controls to fine-tune operation to respond to varying temperature, ventilation requirements, external pressure, and other conditions.

The balancer can set the system for one set of conditions. Only controls can adjust for the seasonal or other operating changes that can occur daily. The MA can become positive if the RAF dominates. This can be a result of poor balancing, inability to balance due to the mechanical system, fan capacities, or damper and duct sizing.

Control solutions should not be considered until a mechanically sound design is produced. Controls should not set the initial balancing. They should fine-tune a mechanically balanced system. The dynamics during operation require accuracy that cannot be achieved if actuators and dampers are attempting to perform two conflicting functions.

The control sequence chosen depends on whether the ventilation requirement is greater than the pressurization plus exhaust requirement, whether local exhausts are variable, and whether elevators or stack effect exist. With these problems to overcome, the standard temperature control functions are inadequate to also balance the airflow.

METHODS OF SIZING THE RECIRCULATION DUCT

Figure 1.6 shows three common methods of sizing the recirculation duct. Normally, design the OA intake through filters and coils to space path and the space through RP and relief outlet systems first. Size fans to move air in and out of the space. The recirculation duct is an important bypass. The system here is simple; typically, there are fittings in the duct. Most systems have a relatively simple recirculation path that has less pressure loss than the relief path plus the OA intake path. There are exceptions.

In each case, the pressure across the duct is the sum of the pressures across the OA to MA and the EA to RP or, simply, the MA to RP. The recirculation duct moves the design supply-air (SA) quantity at this pressure loss.

If the RA duct is large or short and the RA path has a low pressure loss at design, it needs balancing to take more pressure loss. With properly balanced ducts, the fans do not move off their operating points enough to hinder operation or control accuracy. In Figure 1.6A, the duct area is reduced. This has the advantage of low cost; however, it has no flexibility. Given the unknowns that occur during construction, it is not often practical. In Figure 1.6B, a septum-mounted blank-off achieves an orifice effect for more pressure loss.

In Figure 1.6C, a balancing damper is in series with the control damper. This is technically superior. It allows complete flexibility and the adjustment of authority during the selection process. (Authority is a measure of the damper's pressure drop in relation to the rest of the system. We cover it in depth in later chapters.)

RELIEF EXHAUST FAN SYSTEMS

The relief exhaust fan (REF) system is slightly different. Just as when there is no RAF, the pressure loss at design velocity or volume from the OA to MA must equal REA to MA at design velocities to achieve the same volume endpoints (see Figure 1.7). At any given set of fan speeds, the loss from the space to the point RP should be near constant. Regardless of

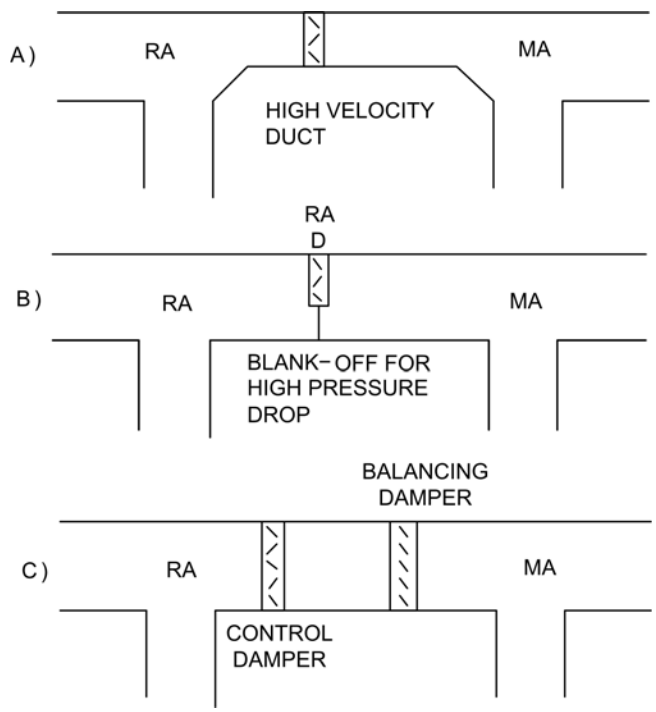

Figure 1.6 Methods of balancing the recirculation duct.

what proportion of flow is going through either fan, the point RP is a near constant pressure point for any given flow; however, unlike the RAF system, the point RP is always negative. Figure 1.7 shows a typical system.

Balancing dampers may be necessary to adjust the system to meet the loss criteria stated above. In some cases, one of the duct systems has a very high or low pressure loss. Balancing dampers also allows the linearization of dampers in modulated positions, as discussed in Chapter 5. If required by codes, special fire and smoke dampers can balance ducts. Limiting the full-open rotation of a control damper is not good practice, as it limits the accuracy of the control.

REF systems have some demanding design requirements. The pressure drops are analyzed and mechanically balanced. Inlet guide vane (IGV) control is possible; however, VFDs are more accurate, and the life-cycle cost is usually lower. The REF must be nonoverloading, since it must operate at very low flow and high pressure at times. One must avoid operation in the unstable portion of the fan curve. In full economizer mode and at design, it must handle a high volume at a much lower pressure drop. At low volume, the fan motor may need its own cooling motor and fan.

Variations include injection fans and mixed-air plenum control via the RAD. Figure 1.8 shows a system with two fans and discharge dampers. Control of pressures is difficult during changeover. Turning a fan ON pulls pressures down, sometimes with too large a reduction in one step. It is sometimes necessary to slow control loops to avoid hunting.

VFD control and/or staged proportional dampers are solutions. Several damper stages per fan are possible. If space pressure can bounce around by 0.1 to 0.2 in. w.g. (25 to 50 Pa), then careful control is not necessary. Abrupt changeover always causes space pressures to vary noticeably.

During the time the system is moving from minimum OA to economizer mode, slow and careful control action is necessary to keep space pressures from getting out of control.

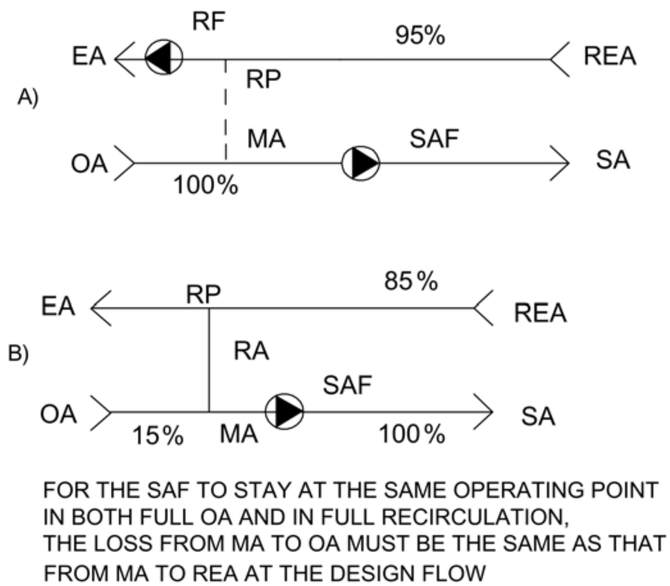

Figure 1.7 Balancing the pressure drops in the economizer versus recirculation.

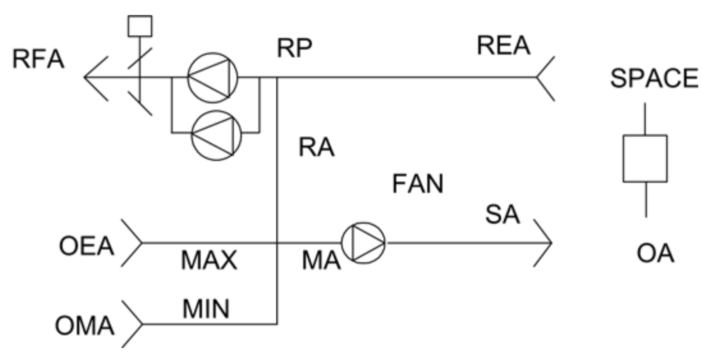

Figure 1.8 Relief exhaust fan system.

The point RP is always negative, and opening the relief air damper before the fan starts causes air to flow in through the relief louver. Space pressure can rise for a short period since the RA duct has less air flowing and the OA has increased. Starting the REF as the relief damper modulates open can avoid long periods of pressure variations.

When the economizer operates, space pressure rises to an upper limit. The REF fan starts at low speed and the relief damper modulates open until space pressure falls to within its control point deadband or limit. During initial transition from recirculation to economizer and vice versa, even with a VFD, the outlet damper may need to modulate from a separate direct digital control (DDC) output to make fine adjustments possible.

If design and controls are not coordinated, a time period of 1 to 2 min incorrect pressures and accompanying flow reversals may be inevitable.

In Figure 1.9, *a–d* mark the extreme operating points. At summer design point, *a* is the REF's operating point. It must pull against a high negative at RP due to the SAF yet move a very low quantity of air. Good accuracy is required to keep space pressure near 0.05 in. w.g.

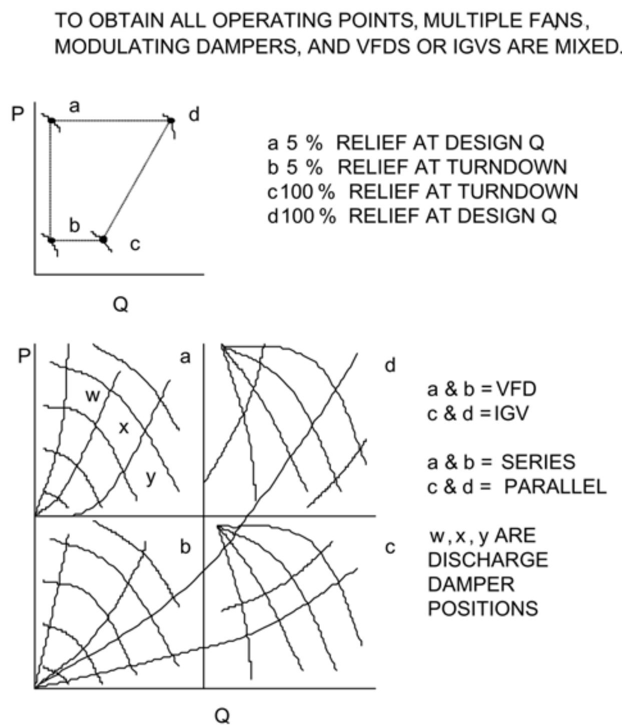

Figure 1.9 Relief-fan requirements at different operating conditions.

(13 Pa). At winter design point, *b* is not atypical. In intermediate seasons, the points *c* and *d* occur during economizer operation.

REVIEW OF PRESSURE LOSS IN A DUCT SYSTEM

The terms and concepts that follow are found in most fluid mechanics textbooks. Idelchik's (1986) is recommended.

Wall resistance causes some loss in fan pressure. Some loss is due to elements in the duct that affect flow directions and the shape of the flow profile; elbows are typical.

P_t, total pressure loss for fan sizing, is equal to the sum of the individual elements' losses in the critical path. Each element's product of the dynamic loss coefficient C and the velocity pressure P_v is equal to the loss in total pressure. Subscripts of C indicate other elements. C_0 is defined in this book as the full-open loss coefficient of a damper. $C_{3/4}$ is the 3/4 open loss, etc.

$$\Delta P_t = C \times P_v \tag{1.1}$$

for any individual element.

C is dimensionless and is the same value in both I-P and SI systems of measurement. It is found by laboratory testing. The *ASHRAE Duct Fitting Database* (ASHRAE 2008b) contains values of different fittings and elements.

In most of Europe, equipment variations occur, and the examples used here are not usually appropriate. For example, flanged dampers have about 90% free area, and louvers are more often a chevron or inverted "Y" with higher losses than the common 45° parallel blade used in the U.S. C is defined as

$$\Delta P_t / P_v . \tag{1.2}$$

C is nearly constant above a Reynolds number (Re) of about 5000, above which most systems operate. Re is a measurement of turbulence. D is the hydraulic diameter, which is the diameter of a circle with the same area as whatever shape through which the air moves, typically a square or rectangle.

$$\mathrm{Re} = 8.6DV \quad \text{(I-P)} \tag{1.3a}$$

where D is in inches and V is in fpm, and

$$\mathrm{Re} = 66DV \quad \text{(SI)} \tag{1.3b}$$

where D is in mm and V is in m/s.

At standard conditions,

$$P_v = (V/40005)^2 \text{ in. w.g.} \quad \text{(I-P)} \tag{1.4a}$$

$$P_v = 0.6V^2 \text{ in Pa, with } V \text{ in m/s} \quad \text{(SI)} \tag{1.4b}$$

Air density in economizer systems is not a major factor and is not included in any calculations in this book. It is nearly constant, since only small temperature changes occur. This is not true in all parts of the system, at high altitudes, or in a high-rise building. However, density is nearly constant. Compression of air does not occur at the low velocities in air-handling systems for HVAC.

If the velocity V is 4000 fpm (20.3 m/s), then the velocity pressure is 1 in. w.g. (247 Pa), since $P_v = (4000/4005)^2$ (P_v in Pa = $0.6V^2$, where V is in m/s).

If V = 2000, then P_v = 0.25 in. w.g. (if V = 10.1 m/s, then P_v = 61 Pa).

$$P_t = P_s + P_v \tag{1.5}$$

There is sometimes confusion between static pressure P_s and P_t for damper sizing. Many manufacturer graphs show P as the pressure loss. By this, they usually mean static pressure instead of total pressure P_t; however, they are not always clearly distinguished. Since $P_t = P_s + P_v$, the static requirement is part of P_t and is sometimes interconvertable for the purposes here. The loss through an exit damper includes P_v if P_t is measured. If P_s is measured as is typical, P_v is not included. The total pressure obeys Bernoulli's law, and static pressure does not. In addition, the definition of C is based on P_t not P_s. Recalculation of losses is necessary when referencing some manufacturers' specifications. In the absence of a subscript, assume P_s is given. For fully ducted dampers, it is not necessary to make a distinction. However, when the damper exits into a large space or atmosphere (exits) or pulls air from a large space or atmosphere (entrances), P_t is used, since exit and acceleration losses exist.

P_v is sometimes referred to as *velocity head*. Strictly speaking, this is incorrect; the meaning is that a loss of one velocity head is the same as C = 1. If an element's loss coefficient C is 3.5, then 3.5 velocity heads are lost or used up in order to push air through it. That is, $\Delta P_t = 3.5 \times P_v$.

$\sum C$ is the sum of the individual loss coefficients in a section of duct of equal area and, therefore, equal velocity. When changing directions or velocity, the loss in the system due to turbulence, expansion, or acceleration requirement is proportional to P_v or differences in initial and final P_v. In the economizer portion of the duct system, the loss due to duct resistance is usually negligible. It is normally ignored in this book. Almost all duct loss is below 0.01 in. w.g. (2.5 Pa) in the examples here; C is the primary value required. If a particularly long and small duct exists, then the wall resistance might matter. A high velocity system above 4000 fpm (20 m/s) in a 20 ft (7 m) long recirculation air duct might have 0.1 in. w.g. (25 Pa) loss.

The engineer calculates the pressure required in a system and adds the losses to size the fan. Dampers are a variable resistance and, when they are fully closed, the engineer sees the entire pressure difference of the subsystem in which they are located.

DUCT ELEMENTS

Figure 1.10 shows a duct system with examples of pressure losses and flow coefficients. At element A, the static pressure is 1.25 in. w.g. (312 Pa); at element B, an orifice with $C = 2$ causes 2 velocity heads loss in static. At element C, the duct is reduced, and the velocity increases to 2000 fpm (10 m/s). At element D, the duct again enlarges, and velocity goes back down to 1000 fpm (5 m/s). Last, there is an elbow with $C = 1$, and the pressure near the end is 1 in. w.g. (250 Pa). Table 1.1 shows loss calculations.

A known equation for this geometry reflects that the loss is 30% of the difference. So, 0.3(0.25 – 0.06) = 0.06 in. w.g. (0.3 of [60 – 15] is the loss in Pa). A duct calculator gives the same result. ($V = 5$ m/s, $P_v = 15$ Pa, and P_s at $A = 310$ Pa. $\Delta P_t = 2 \times 15 + 0.3 \times [60 - 15] + 1 \times 15 = 60$ Pa.)

At the end of the duct section, the static is 1 in. w.g. (250 Pa), since 0.25 in. w.g. (60 Pa) was lost. The sizing of dampers requires calculations dependent on duct and element losses. The mechanical designer must size dampers. The control contractor and the testing, adjusting, and balancingcontractor must work together and have a common understanding.

In Figure 1.11,

$$Q = V \times A, \tag{1.6}$$

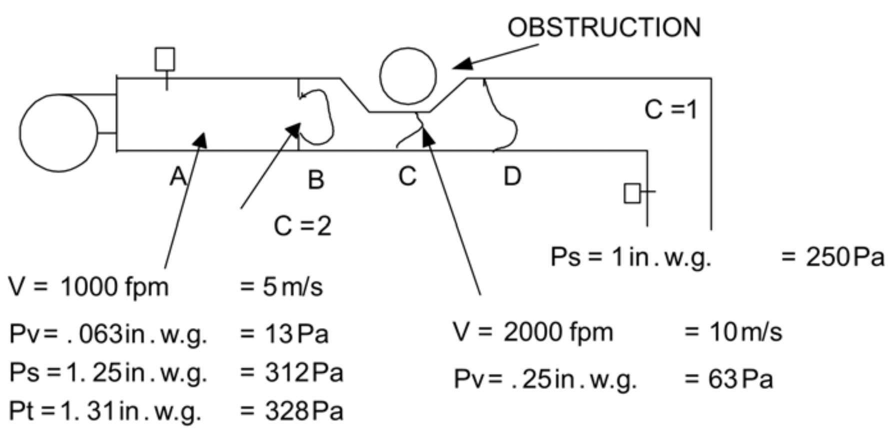

Figure 1.10 Pressure losses and flow coefficients.

Table 1.1 Loss Calculations for Figure 1.10

Element	**Loss ($P_t = C \times P_v$)**	**Loss, in. w.g.**	**Loss, Pa**
B	2 × 0.062 in. w.g. (2 × 15 Pa)	0.12	30[1]
C	0.3 × difference	0.06	15[2]
D	Included in element C (0.3 difference above)		
Elbow	1 × 0.062 in. w.g.	0.06	15
		—	—
Total Loss		0.24	60

[1] $P_v = (1000/4005)^2 = 0.0625$ in. w.g. ($P_v = 0.6(5)^2 = 15$ Pa)
[2] P_v of 2000 fpm = 0.25 in. w.g., and P_v of 1000 fpm = 0.062 in. w.g. (P_v of 10 m/s = 60 Pa, and P_v of 5 m/s = 15 Pa)

where Q is the quantity of air in ft^3/min (m^3/s), V is average velocity in fpm (m/s), and and A is the cross-sectional area in ft^2 (m^2). In Figure 1.11, plane A indicates where the average velocity is found. The flow profile in a duct in three dimensions looks like a section of a balloon. The velocity is highest at the center of a regular or fully established flow profile.

While it takes from 3 to 5 diameters of duct to bring a mildly irregular flow back to normal, it can take 10 diameters for normalization after some elements. In most economizer systems, there is not enough duct length after any element to have a regular flow profile. The velocity profile returns to normal before the static pressure profile.

In Figure 1.12, a backflow at Y occurs after an elbow. The molecules of air tend to keep moving in the direction in which they are traveling. After turning against the wall, the flow profile is distorted. X has an established flow. However, after the elbow, velocity at Z is very high. After three diameters, the flow becomes regular again.

Turning vanes, shown in Figure 1.13, reduce pressure loss and improve the flow profile downstream of an elbow if they correctly guide the change in the direction of the air molecules.

Figures 1.14–1.15 show streamlines, the vena contracta, and flow contours (ACGIH 2006). To calculate the distance a duct must be kept from a louver to avoid high velocity spots that could pull in rain, use the equation in Figure 1.15. Figure 1.16 demonstrates a pitot tube and its readout. The velocity at the center of the duct is about 115% of the average. Turbulence is constant and random. Velocity is an average over time and area; however, the instantaneous speed of air in a duct varies quite a bit.

Figure 1.17 shows a straightener. Swirl occurs due to axial fans and other factors. Straighteners stop swirl, as do coils. If a bad flow profile enters a straightener, it continues down the duct farther than if no straightener exists, because turbulence—airflow perpendicular to the velocity direction—is reduced (AMCA 2002).

Figure 1.18 shows a pressure gradient graph and the ductwork to which it applies. Flow always occurs from a positive pressure area to a negative one or from a less negative area to a more negative one.

Figure 1.19 shows the basic fan and system curve intersections with a IGV and/or VFD. An IGV and a VFD do not produce the same decrease in pressure and volume. Generalizations are difficult, since there are a variety of mechanical differences among IGVs.

Figure 1.20 shows the standard area and velocity designations for flow through an orifice. V_1 is the duct velocity before the orifice (or damper); it is the same as V_4 if the duct is the same size. V_2 is the *conversion velocity* or the velocity inside the orifice (damper). V_3 is the velocity in the vena contracta. A square orifice has a vena contracta area about 60% of the orifice area: it is typically one-half diameter downstream. Jets form at any contraction. When air molecules start angling toward the center of a restriction, they tend to keep moving in the direction in which they have turned.

Figure 1.21 shows the approximate vena contracta in expansion and contraction fittings. It is possible that the vena contracta area and velocity are more appropriate for loss calculations than the free area as used in this book; however, no studies are available giving contracta sizes.

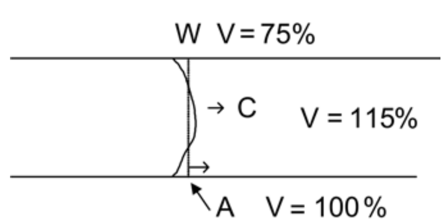

Figure 1.11 Flow profile velocities.

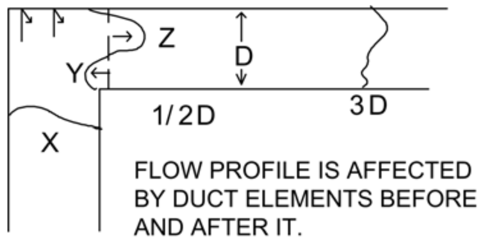

Figure 1.12 Flow profiles at an elbow.

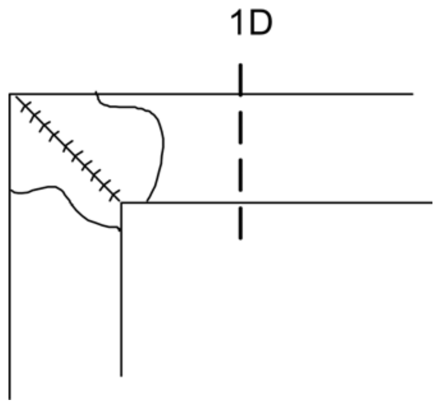

Figure 1.13 Turning vanes.

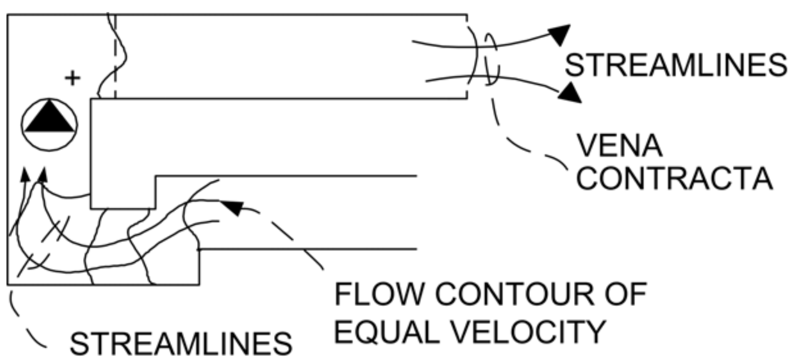

Figure 1.14 Definitions from fluid mechanics.

Figure 1.22 shows the flow streamlines through a damper. The narrowest part is the vena contracta. A damper has a vena contracta near 85% of the damper-free area, and it is near the outlet of the damper in the authors' field observations. Where air velocity is very low, no vena contracta is observed. When damper blades start closing, individual contracta can form.

FREE-AREA RATIO

The free area of an orifice or damper is the amount of open area inside. Add the area between blades to find the total. A damper with hat frame takes up about 15% of the face area with the

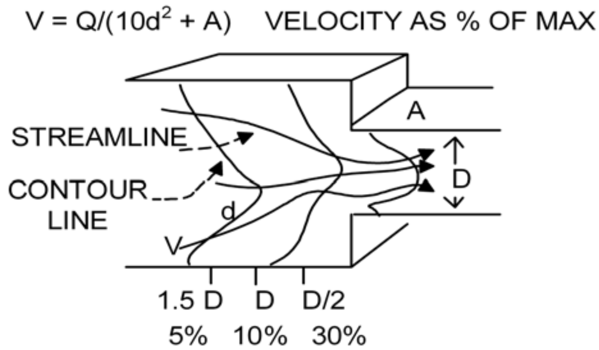

Figure 1.15 Flow profile and velocity entering a restriction.

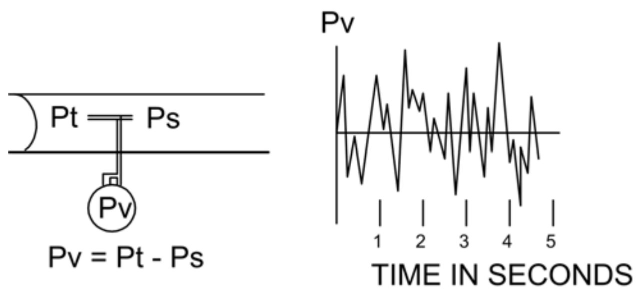

Figure 1.16 Turbulence in pressure readings.

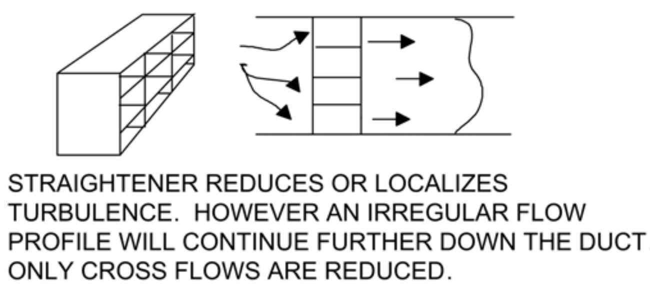

Figure 1.17 Egg crate straightener.

blades taking about 5%. A flange-mounted damper has about 10% of the face area taken up with frame and blades. Small dampers have much less free area in both absolute area and as compared to face area. The ratio A_2/A_1 is the free-area ratio; this is the orifice area/duct cross-sectional area. It is labeled F in this book. Figure 1.20 shows the normal labels. A_1 is the duct or wall area. A_2 is the free area. C_1 is the loss coefficient with respect to V_1 at Area 1. C_2 refers to the loss coefficient with respect to V_2 at A_2.

$$C_1 = C_2/F^2 \tag{1.7}$$

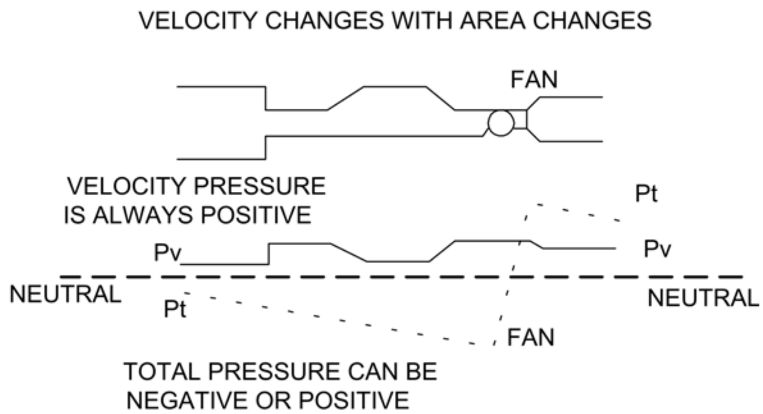

Figure 1.18 Total and velocity pressures in a pressure gradient graph.

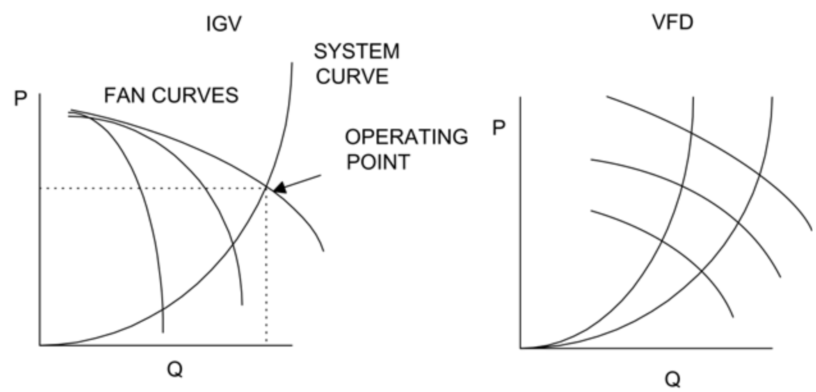

Figure 1.19 Different fan curves for inlet guide vane (IGV) and variable-frequency drive (VFD).

To find free-area velocity V_2, divide V_1 by F. The damper face area is inconsequential in a plenum-mounted application, although it is often given. *Face velocity* typically refers to duct velocity. Many damper manufacturers refer to face velocity in their graphs. Free-area velocity is often more usable. Do not be tricked when given face velocity inappropriately; it is wall-area velocity or the velocity inside the orifice or damper that is useful for calculations.

VARIABLE-AIR-VOLUME (VAV) TERMINALS

Other sources and manufacturers' literature cover VAV well. Most individual thermostat and some DDC systems use pressure independent control, which has three setpoints: temperature, minimum flow, and maximum flow. A flow sensor reads velocity. A common method of control reads the space temperature, compares it to setpoint, and records deviation. This is error. The desired airflow is then calculated based on the error. The desired airflow is controlled between the min and max values. Thus, the temperature is an input to the primary control loop. This method is based on old pneumatic volume reset controllers.

When inlet pressure varies, the box actuator moves the damper to maintain the flow volume at setpoint.

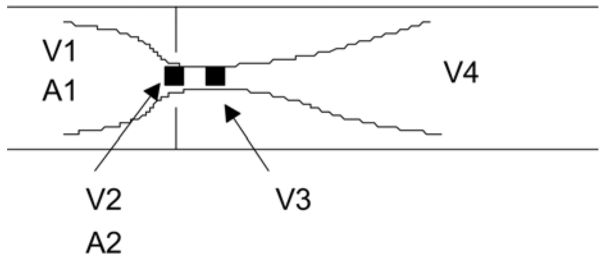

Figure 1.20 Definitions of areas and velocity points through an orifice.

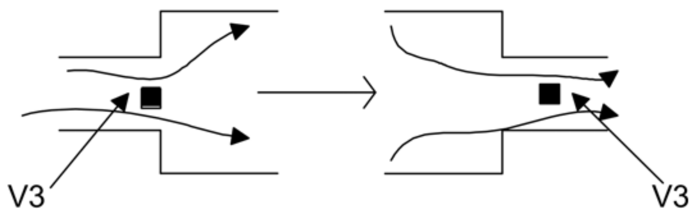

Figure 1.21 V3 is the vena contracta.

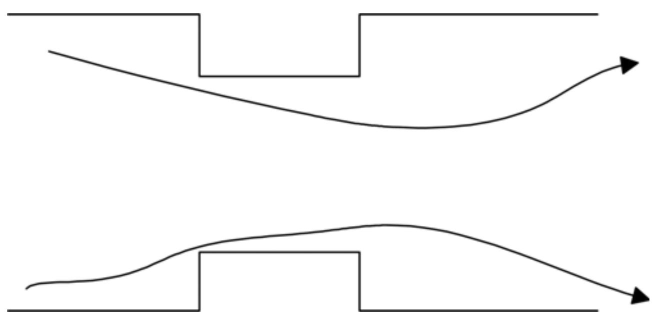

Figure 1.22 Flow streamlines at a restriction.

The maximum setpoint is the design quantity for the zone. The minimum setpoint is an indoor air quality (IAQ) requirement. The amount of air delivered to the space varies with the load. Figure 1.23 shows the equipment layout.

Pressure-dependent control is based on simple zone demand. A proportioning thermostat or sensor and DDC controller adjust the damper position according to deviation from setpoint. There is no min/max control unless the actuator or damper is mechanically restrained from closing completely or a minimum signal voltage is established. Pressure-dependent control is simpler and lower cost than pressure-independent control but not as accurate for ventilation requirement.

The interactions of the OA dampers, VFD or IGVs, duct elements, VAV SA, and RA affect the temperature, ventilation air, space pressure, and energy use.

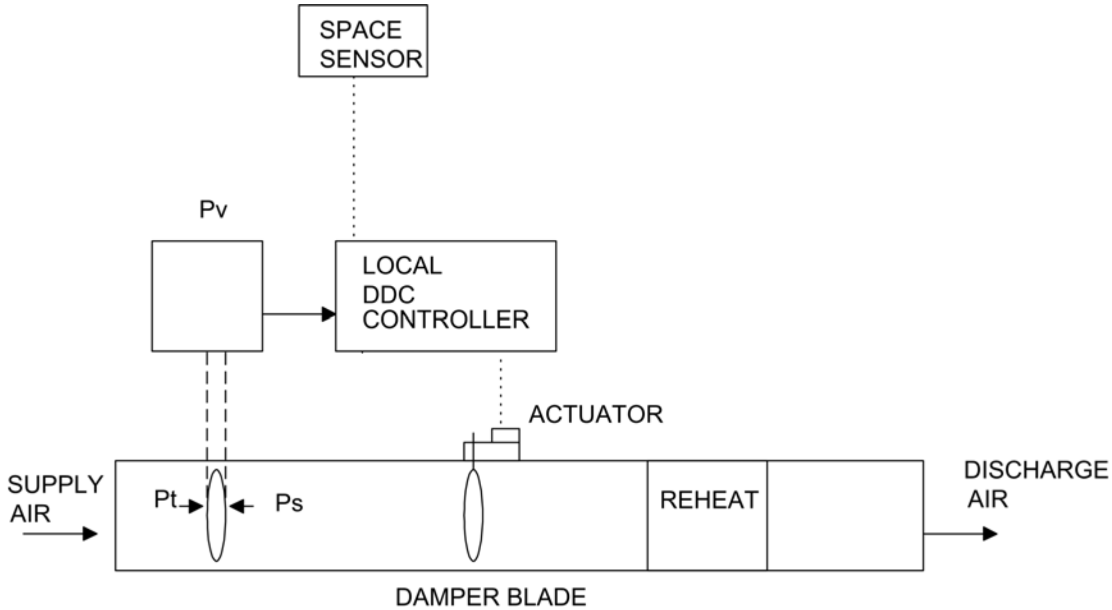

Figure 1.23 Variable-air-volume (VAV) box components.

DDC allows sophisticated control. Strategies exist for varying minimum OA ventilation requirements while distributing the air correctly. A strategy with different box minimums and maximums for heating and cooling is efficient. Summing the demands from zones and ensuring one box is fully open to keep fan pressures as low as possible is a conservative tactic that keeps energy use at a minimum while allowing correct ventilation airflow (Stein and Hydeman 2006).

Figure 1.24 shows one problem with VAV. While the zone SA volume varies with temperature demand, the RA volume does not. In some cases, this causes unwanted pressure differences from zone to zone. If necessary, install a return VAV box with an actuator that follows the movement of the supply box actuator.

SUMMARY

- Air-handling systems must be mechanically balanced to function properly and accurately. Simple systems can be set to maintain the same flow at full economizer and full recirculation by following a few design practices. Analyze the losses involved.
- More complex systems, such as some REF systems and systems with paralleled fans and ducts shared with other air-handling units, are more difficult to analyze.
- After setting the endpoint—full economizer and recirculation—adjust the dampers to ensure modulation does not seriously affect the volume.
- Pressure loss and airflows are complex. Design of duct systems is subject to a number of construction unknowns that may have unintended consequences.
- VAV terminals are often the final control devices for room air distribution. In the U.S., it is not common to control RA and, thereby, room airflow.
- Precision in applying and calculating the pressure losses through the final control elements (dampers and actuators) is necessary to achieve the design goals in air systems with respect to the other duct elements.

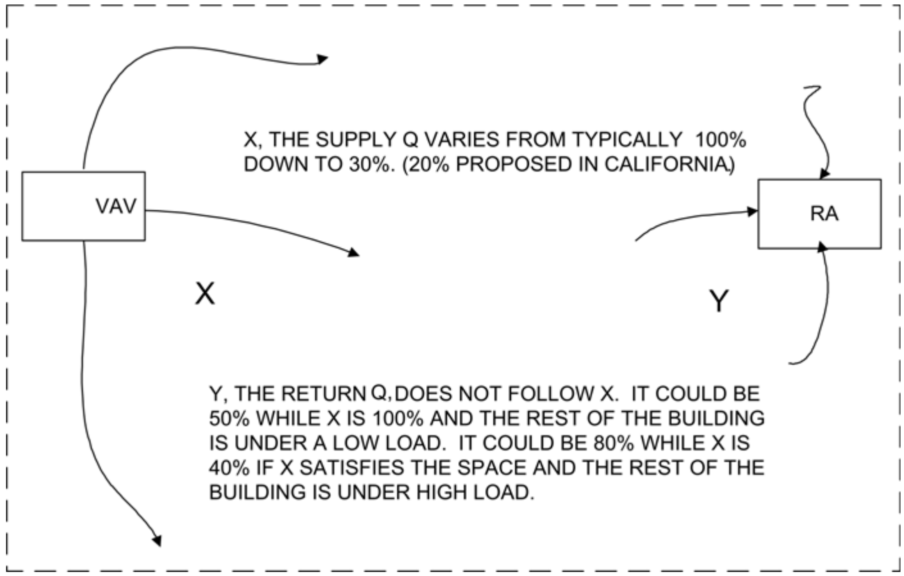

Figure 1.24 Single zone with varying supply and constant return airflow.