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Manual for Ventilation Assessment in Mechanically Ventilated Commercial Buildings

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ABSTRACT

This manual describes procedures for assessing ventilation system performance and other aspects of building ventilation in mechanically ventilated commercial buildings. These procedures are intended to provide basic information on building ventilation for comparing ventilation performance to standards, guidelines and building design values and for investigating indoor air quality problems. The procedures in the manual are based on established measurement techniques and available instrumentation and provide practical means for obtaining reliable information on ventilation performance. The manual does not describe complete system evaluations that are performed during testing and balancing efforts or sophisticated measurement techniques that are used in ventilation research. The manual is written for technically competent indoor air quality investigators, building operators and others who need to perform ventilation assessments in order to address existing problems or as part of preventive maintenance programs. The manual provides background information on building ventilation, discusses instrumentation used in ventilation assessments, describes measurement techniques for determining the values of key ventilation performance parameters, and presents procedures to evaluate building ventilation using these techniques.

Key words: building diagnostics, commercial buildings, indoor air quality, measurement, mechanical ventilation, office buildings, ventilation.

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TABLE OF CONTENTS

Abstract i

Acknowledgements iii

Table of Contents v

Preface vii

1 Introduction 1
1.1 Objective of this Manual 2
1.2 The Relation of this Manual to Design, Operation, Maintenance and Testing 3
1.2.1 Testing, Adjusting and Balancing 4
1.2.2 Indoor Air Quality Diagnostics 4
1.3 Assessment Procedures in This Manual 5
1.3.1 Space-Use and Ventilation Requirements 5
1.3.2 System Design Evaluation 6
1.3.3 System Performance 7
1.4 References 8

2 Background Information 9
2.1 Air and Airflow 9
2.1.1 Air Pressure 9
2.1.2 Air Density 11
2.1.3 Air Speed and Airflow Measurement 11
2.2 Ventilation Systems and Components 12
2.2.1 Systems 12
2.2.2 Components 14
2.3 Building Ventilation 21
2.3.1 Definition of Terms 21
2.3.2 Ventilation Problems 22
2.3.3 Air Change with Outdoors 23
2.3.4 Airflow Inside Buildings 24
2.3.5 Key Issues 25
2.4 References 28

3 Instrumentation 31
3.1 Air Temperature 31
3.2 Relative Humidity 31
3.3 Differential Pressure 32
3.4 Static Pressure Taps 33
3.5 Pitot-Static Tubes 33
3.6 Hot-Wire Anemometers 34
3.7 Rotating Vane Anemometers 34
3.8 Flow Hoods 35
3.9 Tracer Gas Monitors 35
3.10 Smoke Tubes 35
3.11 References 36

4 Measurement Techniques....37
4.1 Air Temperature....37
4.2 Barometric Pressure....38
4.3 Pressure Differences....39
4.4 Relative Humidity....39
4.5 Air Density....40
4.6 Tracer Gas Concentration....41
4.7 Percent Outdoor Air Intake....42
4.7.1 Percent Outdoor Air - Temperature....43
4.7.2 Percent Outdoor Air - Tracer Gas....44
4.8 Airflow Rate....47
4.8.1 Pitot Traverses in Ducts....47
4.8.2 Hot-Wire Traverses in Ducts....52
4.8.3 Vane Anemometer Traverses at System Coils....53
4.8.4 Flow Hoods at Outlets and Inlets....53
4.8.5 Vane Anemometers at Outlets and Inlets....54
4.8.6 Multiplicative Method of Determining Outdoor Airflow Rates....55
4.9 Air Change Rate....58
4.9.1 Tracer Gas Decay....58
4.9.2 Equilibrium Carbon Dioxide Analysis....59
4.10 References....60

5 Assessment Procedures....63
5.1 Procedures....63
5.1.1 Space Use Analysis....63
5.1.2 System Design Evaluation....66
5.1.2.1 Design Parameters....66
5.1.2.2 Sources of Design Information....68
5.1.3 Comparing Space-Use and Design....73
5.1.3.1 Minimum Outdoor Air Intake....73
5.1.3.2 Supply Airflow Rate....73
5.1.3.3 Exhaust Airflow Rate....74
5.1.3.4 Pressure Relationship Between Spaces....75
5.1.4 Performance Measurements....75
5.1.4.1 Air Handlers....77
5.1.4.2 Air Distribution System....79
5.1.4.3 Ventilated Space....82
5.1.4.4 Exhaust Systems....83
5.1.4.5 Building Pressure Relationships....85
5.1.4.6 Whole Building Air Change Rate....86
5.2 Applications....88
5.2.1 Preventive Maintenance....88
5.2.2 Indoor Air Quality Diagnosis....89
5.2.3 Energy Assessment....89
5.3 References....90

Appendices
A Terminology....91
B Altitude Correction Factors for Barometric Pressure....97
C Forms....99
D Ventilation System Classification....105
E Verification of ASHRAE Standard 62-1989 Ventilation Recommendations....115
F Resources....121

PREFACE

This manual describes procedures for assessing ventilation performance in mechanically ventilated commercial buildings. The potential users of this manual include indoor air quality investigators, building operators and others who deal with indoor air quality and energy in buildings. In many situations, their needs for ventilation performance information can be met with a limited assessment of building ventilation, as opposed to a complete system evaluation. This manual was developed to describe procedures for obtaining this basic performance information, although the results of such a preliminary effort may point out the need for a more detailed assessment. Currently available guidance on ventilation assessment is too detailed and complex to meet the requirements of the intended audience of this manual. This manual provides these users with tools more appropriate to their background and needs. In some sense, this manual is a distillation of established HVAC assessment approaches, along with other relevant tools. The procedures in this manual are not a substitute for a complete HVAC assessment as is done in a testing, adjusting and balancing (TAB) effort. As discussed in this manual, testing, adjusting and balancing is a complex procedure requiring extensive training and experience, as are indoor air quality diagnosis and building energy evaluation. This manual is not a substitute for the training and experience required to perform these other tasks, but they will empower the user to make reliable assessments of building ventilation.

This manual is based, in large part, on measurement procedures described in HVAC and TAB industry standards and guidance documents. Other procedures that have proven to be useful and reliable in building ventilation evaluation are also included. In all cases, the procedures incorporated into this manual were selected based on their appropriateness to preliminary ventilation assessments and their usability by the intended audience of this manual. Measurement procedures requiring more extensive training and experience are not included.

While this manual is not a consensus document, its development has benefitted from the input of many qualified individuals. In August of 1992 a workshop was held at NIST to discuss the development of the manual. The workshop participants included indoor air quality researchers, industrial hygienists, individuals from several federal agencies, and individuals from organizations that represent the HVAC industry, building owners and managers, building operators and facility managers. The workshop participants provided valuable input on the objective of the manual, its format and the intended audience. Review drafts of the document were subsequently sent to the workshop participants and others for comment. It is hoped that the participation of these individuals has served to increase the usefulness and technical accuracy of this manual.

1 INTRODUCTION

Buildings are ventilated with outdoor air for the comfort and health of the occupants. Depending on the weather conditions, this outdoor air may be heated or cooled, which consumes energy and costs money. Building designers, owners and operators are faced with the balancing act of bringing in enough outdoor air for the occupants and controlling energy costs. To balance indoor air quality and energy, the building operator needs to understand how the building is being ventilated, including both how the ventilation system is designed to operate and how it is actually performing. Ventilation performance evaluation is used to determine if a ventilation system is operating as designed and is meeting appropriate guidelines, such as ASHRAE Standard 62-1989. Some of the questions that are answered in a ventilation evaluation include: How much outdoor air is being brought into the building? How much outdoor air is getting to the people? What unintentional airflows exist because of air leakage and system operation?

Ventilation system evaluation is also important when trying to find the causes of indoor air quality problems. However, some indoor air quality investigations do not include adequate evaluation of building ventilation. In many studies that are done today, ventilation evaluation ranges from nothing at all, to the consideration of only design values with no measurement, to the use of unreliable measurement procedures. Ventilation evaluation is sometimes inadequate because the investigator does not understand buildings and ventilation systems and is not familiar with ventilation measurement techniques. Ventilation measurement is sometimes viewed as too expensive and complex for anyone but a researcher. But when an indoor air quality investigator does not evaluate ventilation system performance, he or she is less likely to be able to understand the indoor air quality problems in the building.

While the term ventilation assessment sounds simple, understanding ventilation and airflow in a mechanically ventilated commercial building can be very difficult. Buildings contain many different spaces that exchange air with each other, the outdoors and the ventilation system. There are many different kinds of ventilation systems and system control strategies. And building airflow rates vary with time as weather conditions change, the occupants operate doors and building equipment, and the controls modulate ventilation system operation. Given the complexity and variability of buildings and airflows in buildings, there is no single ventilation assessment procedure that can be applied to all buildings under all circumstances. Instead, a variety of procedures are available, and an appropriate assessment strategy must be based on the building and ventilation system design and the objectives of the assessment.

1.1 Objective of this Manual

The objective of this manual is to describe practical procedures for evaluating the performance of commercial building ventilation systems. The system performance can then be compared to the system design specifications, and to ventilation requirements based on building codes, ventilation standards and other guidelines. There are many different ventilation system performance parameters including supply airflow rates, outdoor air intake rates, supply air temperatures, interior temperatures and pressure differences across system components. The purpose of the ventilation assessment being conducted in a building and the resources available determine which parameters are measured, when and how often they are measured, and which measurement techniques are used to determine their values. The procedures described in this manual are intended for limited ventilation evaluations lasting about 3 to 5 person-days, as compared to a complete HVAC system evaluation such as that performed in a testing, adjusting and balancing (TAB) effort.

This manual is written for people who need a basic indication of how the ventilation system in a building is performing. They do not need a complete testing, adjusting and balancing effort or a long-term research project. Intended users include building operators with a technically trained and experienced staff. The manual is also written for technically competent indoor air quality investigators such as industrial hygienists and other professionals.

There are several situations in which this manual could be used. For example, a building operator is using a preventive maintenance program to insure good indoor air quality and to control energy consumption. As part of this effort they are conducting an initial evaluation of the ventilation system and periodic follow-up evaluations. In other situations, a building operator is trying to address a suspected indoor air quality problem and needs to evaluate the design and performance of the ventilation system. This manual could also be used by an indoor air quality consultant, such as an industrial hygienist, who is trying to solve an air quality problem in a building.

These situations require answers to several important questions: How was the ventilation system designed to perform? Is the ventilation system being operated as designed? Is it performing as designed? Is the design and performance adequate given the current activities and occupancy levels in the building? Have any space-use changes occurred that require a change in the ventilation system operation? Are the design and performance consistent with current ventilation standards and guidelines?

The outcome of these evaluations may be that the system needs to be examined in more detail than is possible using the procedures described in this manual or that there are significant system deficiencies. In these cases, the system should be evaluated by an experienced testing and balancing service.

1.2 The Relation of this Manual to Design, Operation, Maintenance and Testing

This section discusses how the ventilation assessment procedures in this manual are related to ventilation system design, building commissioning, system maintenance, and indoor air quality testing.

The design of HVAC systems in commercial buildings is a complex process. The system design specifies airflow rates at various points in the ventilation system and how these airflow rates should change in response to weather conditions, internal loads and time of day. These specifications are based on the activities in the building zones, the thermal loads generated in these zones, the number of occupants, and recommended or minimum outdoor air ventilation rates from appropriate building codes, ventilation standards and guidelines. The specification of system airflow rates is often limited by uncertainties in how the building will be used and in the number of occupants in the zones. It is important to document the assumptions on thermal loads and occupancy levels used in the design. This information is very helpful when the ventilation system is evaluated and when space-use changes occur in the building.

Building commissioning is the process in which the HVAC system installation and performance is evaluated to ensure that the system will perform as designed. Commissioning is often thought of as a process that only occurs when a building is first constructed, but it is important throughout the life of a building as system components deteriorate and as space-use and occupancy change. ASHRAE's *Guideline for Commissioning of HVAC Systems* (1989) describes procedures and methods for commissioning during the various phases of a building's life, i.e., predesign, design, construction, acceptance and post-acceptance. If a building has been thoroughly commissioned, the documentation needed to evaluate the ventilation system performance later in the life of the building will be available. In terms of ventilation evaluation, commissioning includes a complete testing, adjusting and balancing of the HVAC system.

HVAC system maintenance is crucial to reliable system performance over time. It involves many factors including inspecting and repairing system components, changing filters, cleaning system components such as coils, calibrating control sensors, and periodically evaluating ventilation system performance. If such maintenance procedures are not routinely employed, system performance will deteriorate, leading to the potential for increased energy consumption, reduced equipment life, poor thermal comfort and indoor air quality problems. This manual describes basic procedures for ventilation system performance evaluation that could be used in a preventive maintenance program.

1.2.1 Testing, Adjusting and Balancing

This section discusses how the procedures in this manual are related to testing, adjusting and balancing. Testing, adjusting and balancing (TAB) is the systematic process of checking and adjusting building HVAC systems to meet the design performance requirements. TAB procedures involve all HVAC-related systems including air and water distribution, electrical and mechanical equipment, controls, and sound and vibration. TAB equipment and procedures are well-documented, and there are established industry certification and training programs (AABC 1989, ASHRAE 1988, Bevirt 1984, NEBB 1991, SMACNA 1983). The application of TAB procedures in a new building to make the HVAC systems perform as intended is a critical part of the system installation. Systems in existing buildings also need to be rebalanced periodically to ensure that they are performing as intended and are meeting the current ventilation requirements of the building. TAB should be applied to existing buildings after there have been significant changes in space-use and internal loads, major design and equipment modifications, and after several years have elapsed since the original system installation.

There are several important differences between TAB procedures and the procedures in this manual. TAB procedures address every component of the HVAC system, while this manual covers only the air distribution system. A complete TAB effort also requires considerably more time and effort, and provides much more information, than the evaluations in this manual. A TAB effort measures all system airflow rates, while the procedures in this manual involve only a subset of these airflow rates and do not involve any adjustment or balancing of the air distribution system. The procedures in this manual are not a substitute for testing and balancing, however, the results may indicate that a complete TAB effort is needed.

1.2.2 Indoor Air Quality Diagnostics

The procedures in this manual can be used when diagnosing indoor air quality problems or as part of a proactive program to prevent such problems. Approaches to indoor air quality diagnosis and prevention are described in *Building Air Quality* (1991) published by the U.S. Environmental Protection Agency and the National Institute of Occupational Safety and Health. *Building Air Quality* and other documents describe how to respond to indoor air quality problems and how to prevent future problems. Their approach to existing problems is to identify pollutant sources inside and outside the building, to develop an understanding of the ventilation system design, operation and performance, and to identify the pathways by which pollutants move into and within the building. These strategies refer to the impact of the ventilation system and provide limited information on how to assess system performance, but they do not include many specifics on evaluating system design and measuring system performance.

In the area of problem prevention, *Building Air Quality* and other resources describe how to develop an IAQ profile of a building by documenting information on building activities, pollutant sources and HVAC system design. The discussions of ventilation system evaluation in these documents are fairly general. This ventilation assessment manual discusses the specific parameters to extract from the design, how to use ventilation standards and guidelines to develop ventilation requirements for a space, and how to measure ventilation system performance. These other resources also point out the importance of building management practices in preventing indoor air quality problems and describe building management programs including building staff responsibilities and record-keeping. This manual describes ventilation system design evaluation and system performance assessment that could be used in such a proactive management program.

1.3 Assessment Procedures in this Manual

As mentioned earlier, there is no simple ventilation assessment procedure that can be applied to all buildings regardless of size, layout, ventilation system design and the goals of the evaluation. Instead, a variety of procedures exist to address different ventilation performance issues. This section describes the assessment procedures that are presented in the manual. These procedures are appropriate to ventilation evaluations of limited scope and intensity, as opposed to the detailed and thorough evaluations performed in a TAB effort.

1.3.1 Space-Use and Ventilation Requirements

The first step in performing a building ventilation assessment is to determine the ventilation requirements, including outdoor air intake rates and supply airflow rates. Ventilation requirements are determined by the activities occurring within the building and the occupant densities. The ventilation requirements determined from the current building use may be different from the original design values due to changes in space-use that have occurred over time or because the original design was based on out-of-date ventilation recommendations.

The analysis of ventilation requirements begins with an accounting of the activities in a building. The building is divided into a number of zones based on the activities in the zones, their location in the building, floor area and the number of occupants. The outdoor air requirements for each space are determined based on appropriate ventilation standards, guidelines or building codes (e.g. ASHRAE Standard 62-1989). Depending on the space type, they will be in units of total L/s (cfm), L/s per square meter (cfm per square foot) of floor area, or L/s (cfm) per person. Requirements for supply airflow are based on the thermal loads in the zones. Additional ventilation requirements may also exist, such as maintaining a pressure difference with adjoining spaces or exhausting air directly to the outdoors.

1.3.2 System Design Evaluation

The next phase of a ventilation evaluation is to document the ventilation system design to enable comparisons with the ventilation requirements and with ventilation performance measurements. The system design evaluation includes understanding the ventilation system layout and analyzing the system specifications, including its intended operation. The ventilation system layout includes the number of air handlers, their location, and the spaces within the building that they serve. Relevant system specifications include the following:

- **Design conditions in space**
  - **Air temperature**
  - **Relative humidity**
- **Air handler specifications**
  - **Supply airflow capacity**
  - **Supply air temperature and relative humidity**
  - **Minimum outdoor air intake**
- **Air handler control**
  - **Sequence of operations, specifically the modulation of the supply airflow rate and the outdoor air intake rate**
- **Exhaust and return fan specifications**
  - **Airflow rate**
  - **Sequence of operations**

1.3.3 System Performance

The final aspect of the ventilation evaluation is to conduct performance measurements on the ventilation system. The range of parameters considered in performance evaluation are presented below. A ventilation system performance assessment will not necessarily involve all of these parameters, depending on the objectives of the assessment, the resources available and the ventilation system design.

Air Handlers
- **Supply airflow rate**
- **Supply air properties: temperature and relative humidity**
- **Outdoor air intake rate**
- **Sequence of operations**

Air Distribution Ductwork
- **Supply airflow rate through the ductwork at various points in the air distribution system**

Terminal Units
- **Supply airflow rate**
- **Supply air properties: temperature and relative humidity**
- **Outdoor air delivery rate**
- **Sequence of operations**

Ventilated Space
- **Supply airflow rate: airflow rate provided by the diffusers**
- **Supply air properties: temperature and relative humidity**
- **Air movement: airflow patterns within the ventilated space**

Exhaust Fans
- **Airflow rate**
- **Pressure relationship of space: pressure within the space being exhausted relative to adjoining spaces**
- **Sequence of operations**

Other Performance Issues
- **Pressure difference across the exterior envelope of the building at locations such as entrances and loading docks**
- **Pressure difference between spaces of interest within the building, such as between stairwells and the occupied space**

1.4 References

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2 BACKGROUND INFORMATION

This section contains basic information on airflow and building ventilation that is valuable when evaluating ventilation system performance. More complete discussions of airflow and ventilation are available in the references.

2.1 Air and Airflow

HVAC systems are designed to bring outdoor air into buildings and to keep the interior at a comfortable temperature and relative humidity. Therefore, it is important to understand something about air and airflow when evaluating HVAC system performance. Detailed discussions of temperature, heat transfer, psychrometrics, and fluid mechanics are presented in Bevirt (1984) and NEBB (1986). This section contains a limited discussion of some key concepts: air pressure, density, and velocity and airflow measurement.

2.1.1 Air Pressure

Air pressure is important in ventilation evaluation for three reasons: ventilation system fans must overcome system pressure losses; velocity pressure is used to determine airflow rates; and air pressure affects air density.

There are three types of air pressure: static, velocity and total. *Static pressure* is the capacity of the airstream to flow against the resistances in the system. The static pressure in a ventilation system increases across the fan and decreases across system components such as coils, dampers, and filters and through the ductwork. *Velocity pressure* describes the kinetic energy in the airstream and is equal to $\frac{1}{2}\rho v^2$, where ρ is the air density and v is the air speed. Velocity pressure is exerted in the direction of the airflow. *Total pressure* is the total amount of energy in the airstream at a point in the system and is equal to the sum of the static and velocity pressures at that point.

The static pressure drop across an air filter or other system component depends on its shape, size and configuration and on the air speed through the component. Manufacturers usually provide information on the pressure drop across their components as a function of air speed. Static pressure also decreases through ductwork because of friction at the duct walls, and this pressure drop depends on the air speed, duct diameter and roughness of the duct walls. Friction charts have been developed that give the pressure drop through ducts as a function of airflow rate and duct diameter (ASHRAE 1993, SMACNA 1990). Static pressure also decreases in ductwork due to dynamic losses when the speed or direction of the airstream changes, for example at transitions, elbows and junctions (ASHRAE 1993, SMACNA 1990). These losses depend on the square of the air speed.

Figure 2.1 is an example of the static pressure changes within a ventilation system (AABC 1990). This schematic shows the pressure drop (PD) across the system components (outdoor air louver, filters, coils, diffusers and return air grilles) and through the distribution ductwork. The selection of a fan for a ventilation system is based on such an analysis of pressures.

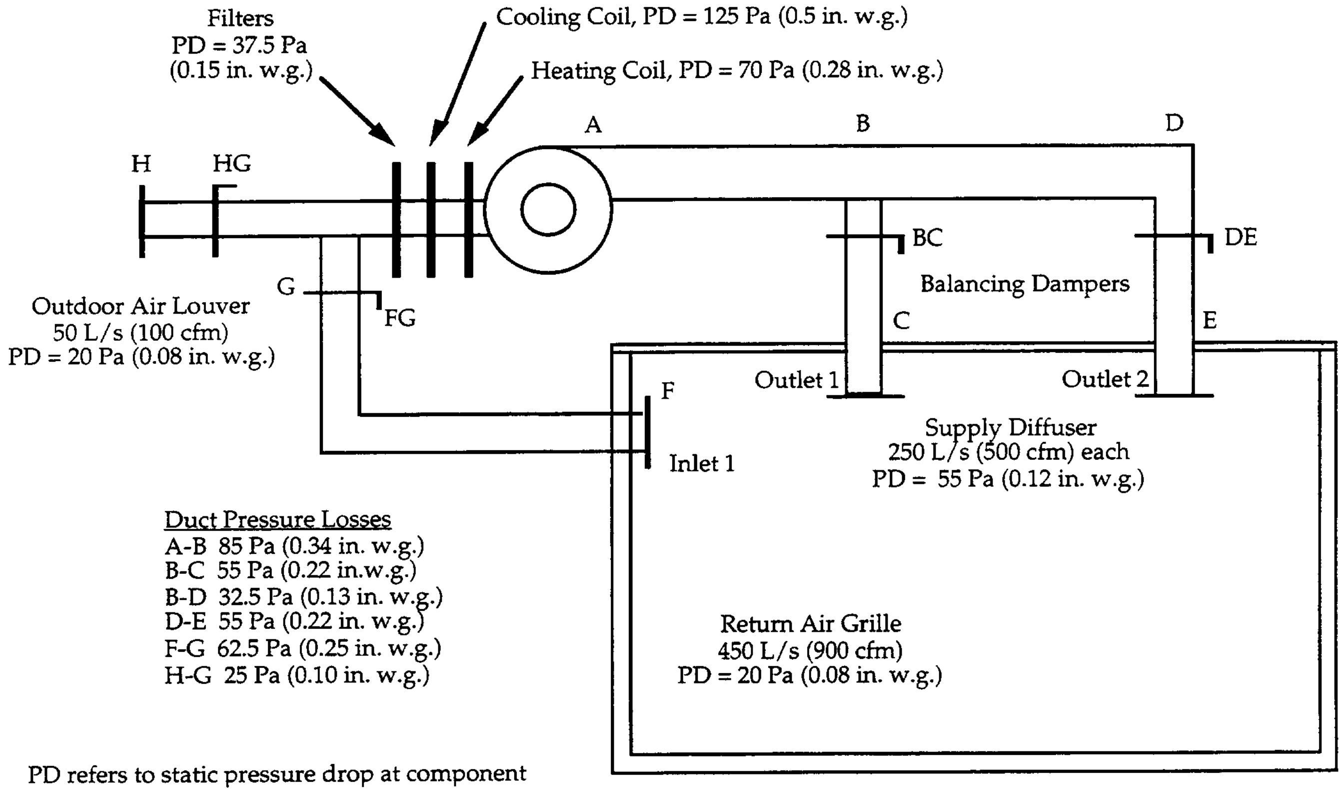

Figure 2.1 System Schematic Illustrating System Losses (AABC 1990)

Static pressure measurements across the fan, filters, coils and dampers, and through the distribution ductwork, can be compared to their expected values. Significant differences between the expected and measured pressures can be a sign of problems such as missing or dirty filters or obstructions in the ducts.

As mentioned earlier, velocity pressure is related to air speed, and therefore airflow rates in ductwork can be determined by measuring velocity pressure. Velocity pressure is measured with pitot tubes and pressure gauges and is converted to air speed based on the air density. The airflow rate is obtained by multiplying the average air velocity in the duct by the area of the duct.

Air density is needed to convert from velocity pressure to air velocity, and the barometric pressure is needed to calculate the air density. The *barometric pressure* at a point in a ventilation system is the ambient barometric pressure plus the static pressure at that point in the system. The ambient barometric pressure can be measured on site with a portable barometer, or it can be obtained from a local weather station. The barometric pressure at standard conditions at sea level is 101.3 kPa (1 atm, 14.7 psi, 406.8 in. water, 29.92 in. Hg, 760 mm Hg). Barometric pressure is a function of altitude, and weather station readings are usually corrected to sea level. Therefore, weather station readings must be corrected for altitude before using them to calculate air density. In most HVAC applications, the barometric pressure does not need to be corrected for duct static pressure.

2.1.2 Air Density

Converting velocity pressure to air speed requires a value of the air density. Air density depends on barometric pressure, dry-bulb temperature and water vapor content of the air. One can determine the air density from these quantities using equations in the ASHRAE Fundamentals Handbook (1993) or the charts and tables in ASHRAE Standard 111-1988. Appendix D of ASHRAE Standard 111 contains example density calculations. At standard conditions, i.e., dry air at 21.1 °C (70 °F) and a barometric pressure of 101.3 kPa (29.92 in. Hg.), the air density is 1.2 kg/m^3 (0.075 lb/ft^3). Section 4.5 of this manual describes the calculation of air density.

2.1.3 Air Speed and Airflow Measurement

The measurement of air speed (in units of m/s or fpm) and airflow rate (in units of L/s or cfm) is an important part of evaluating ventilation system performance. Air speeds can be measured directly, or indirectly based on velocity pressure. Air speed is used to determine the airflow rate in a ventilation system by multiplying the average air speed in a duct by the duct area. The average air speed is determined from multiple air speed measurements in a duct cross-section.

Air speed can be measured directly with hot-wire and vane anemometers; information on these devices is contained in section 3. When using air speed measurements to determine the airflow rate in a duct, velocity variations across the duct can cause errors in the average velocity. The uniformity of the velocity is affected by the location of the traverse plane in the ductwork relative to disturbances and obstructions such as elbows and dampers.

Air speed in a duct can also be determined by measuring velocity pressure with a pitot tube and converting it to air speed. The velocity pressure is measured with a manometer connected to the static and total pressure taps of the pitot tube. Pitot tubes and manometers are discussed in Section 3. As in the case of direct air speed measurements, the accuracy of airflow rate measurements using pitot tubes is affected by the uniformity of the velocity across the duct.

Airflow rates at supply outlets and return and exhaust inlets are measured with flow hoods or with vane anemometers. These devices and their use are discussed in Sections 3 and 4.

2.2 Ventilation Systems and Components

HVAC systems heat and cool buildings in response to thermal loads, deliver outdoor air to the occupants, and remove contaminants generated in the space. This section contains a basic description of ventilation systems and equipment in relation to the evaluation of ventilation system performance.

2.2.1 Systems

There are three general categories of ventilation systems: central, perimeter and unitary. Central systems are large, centrally-located HVAC systems that serve many zones through a system of air distribution ductwork. Perimeter systems condition zones located along the perimeter of a building, where the thermal loads are influenced by outside weather. Unitary systems are small, factory-assembled HVAC systems such as roof-top units.

Central Systems

Central systems, sometimes referred to as "all-air," supply conditioned air to the space through a system of ductwork. Cooling is not provided in the space, but some zones may have auxiliary heating. There are many types of central systems (ASHRAE 1992, Bevirt 1984, McQuiston and Parker 1977, NEBB 1986), and the major differences between them are described below. Appendix D contains descriptions of several types of central systems.

<u>Single Zone and Multizone</u>: Single zone systems serve one temperature control zone through a single-duct air distribution system. The supply air is heated or cooled depending on the thermal loads within the zone. The supply air temperature is controlled by varying the temperature of the heating or cooling fluid, varying the flow rate of the fluid, and using face and bypass dampers at the coils. Multizone systems have two supply airstreams at the air handler, a hot deck and a cold deck, and serve multiple zones. The thermal loads of each zone are met with a mixture of the two airstreams carried to the zone by a single duct. The mixing of the hot and cold airstreams for each zone is accomplished with dampers at the air handling unit.

<u>Single Duct and Dual Duct</u>: In single duct systems, a single distribution duct system carries the supply air to the zones. Dual duct systems have two conditioned airstreams, a hot deck and a cold deck. Two separate systems of ductwork distribute these airstreams to mixing boxes in the zones where they are mixed to meet the thermal loads of the zones.

<u>Reheat</u>: Reheat systems are similar to single zone systems, except they can provide additional heat to individual zones. Therefore, they are able to serve multiple zones with different thermal loads.

Constant Volume and Variable Air Volume: In constant volume systems, the supply airflow rate is constant and the supply air temperature varies to meet the thermal loads. The supply air temperature is controlled by varying the temperature of the heating or cooling fluid, varying the flow rate of the fluid, changing the proportions of outdoor and recirculation air, and using face and bypass dampers at the coils. Sometimes additional heat is provided in the zones (reheat). In variable air volume (VAV) systems, the supply air temperature is constant, and the supply airflow rate to the zones is varied using VAV boxes of various designs.

Return Fans: Some central systems have return fans to pull air from the space back to the air handler. Other systems have no return fan, and the supply fan is used to pull return air from the space.

Outdoor Air Intake Control: Outdoor air can be brought into the building through a dedicated fan. In systems without an outdoor air fan, the low pressure on the upstream side of the supply fan pulls outdoor air into the system. Several strategies are used to control the outdoor air intake rate: 100% outdoor air intake at all times with no recirculation, a fixed outdoor air intake damper position, and modulation of the outdoor air intake damper based on the outdoor and return air temperatures (or enthalpy). In *economizer* systems, the outdoor air intake rate is increased during mild weather to use the outdoor air for cooling the building.

Perimeter Systems

Perimeter systems condition zones along a building's exterior where thermal loads are affected by outdoor weather. Sometimes the central system serves the perimeter, but in other cases there is a dedicated perimeter system. In an *air-water induction unit,* a central air handler supplies conditioned air (primary air) to induction terminal units. The primary air flows through high-velocity nozzles within the units and induces room air (secondary air) into the unit. Chilled or hot water coils in the unit cool or heat the secondary air from the room. *Fan-coil units* consist of a finned-tube coil supplied with hot or chilled water and a fan that circulates room air over the coil. The units sometimes have an outdoor air connection through the exterior wall, in which case they are referred to as *unit ventilators* or *through-the-wall ventilators.* Perimeter heating can also be provided by radiators and electric baseboard heaters, and these devices do not involve outdoor air ventilation.

Unitary Systems

Unitary systems are factory-assembled systems that provide heating, cooling and outdoor air. There are many types of unitary systems including roof top units, through-the-wall air conditioner systems, and heat pumps. Both roof top and through-the-wall systems provide outdoor air, while heat pumps may or may not be connected to a source of outdoor air.

2.2.2 Components

This section discusses the following components of ventilation systems in relation to ventilation system evaluation:

- **Fans**
- **Filters**
- **Coils**
- **Dampers**
- **Controls**
- **Ductwork**
- **Terminal Units**
- **Air Outlets and Inlets**

Fans: Fans act as pumps by creating a pressure difference that moves air through a ventilation system. There are many types of fans, and the Air Movement and Control Association has developed classifications of fans and their arrangements (AMCA 1990).

The performance of a fan is described by a *fan curve* that relates the static pressure increase across a fan to the airflow rate through the fan at a constant fan speed in revolutions per minute (rpm). Figure 2.2 shows an example of a set of fan curves at different fan speeds.

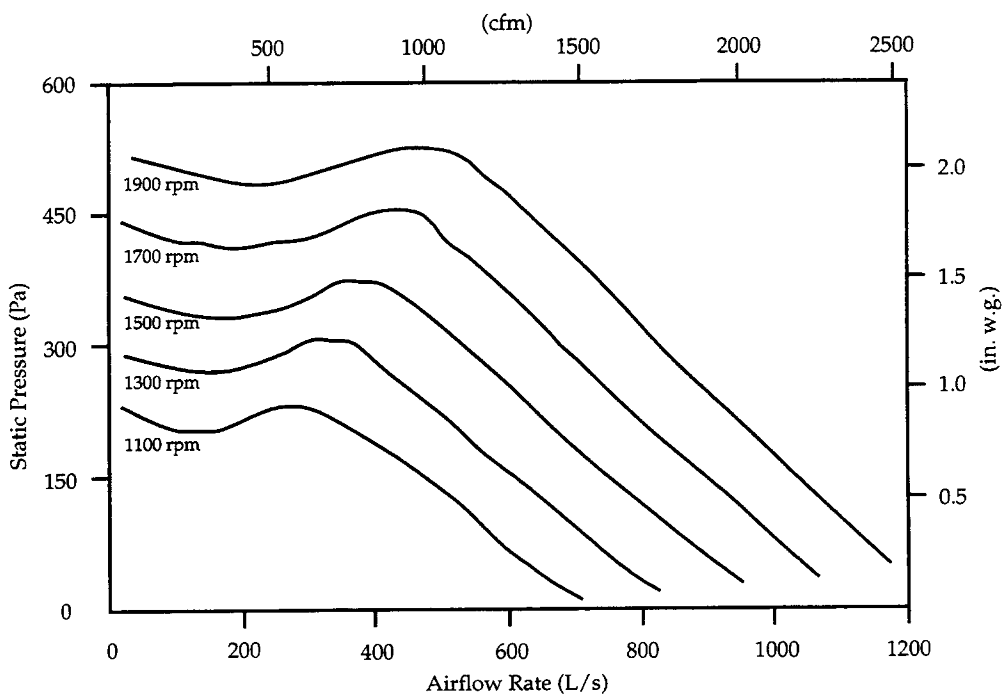

Figure 2.2 Example of a Set of Fan Curves

Air pressure decreases through the ventilation system, and this pressure drop is equal to the total airflow resistance of all the system components and the ductwork. This prèssure drop depends on the airflow rate and is described by a *system curve,* i.e., the pressure loss through the system versus the airflow rate. Figure 2.3 is an example of a system curve. The system curve is constant as long as there are no changes in the system components. The system curve is affected by changes in damper position, dirty filters, condensation on coils, holes in ductwork, and obstruction of outlets or inlets.

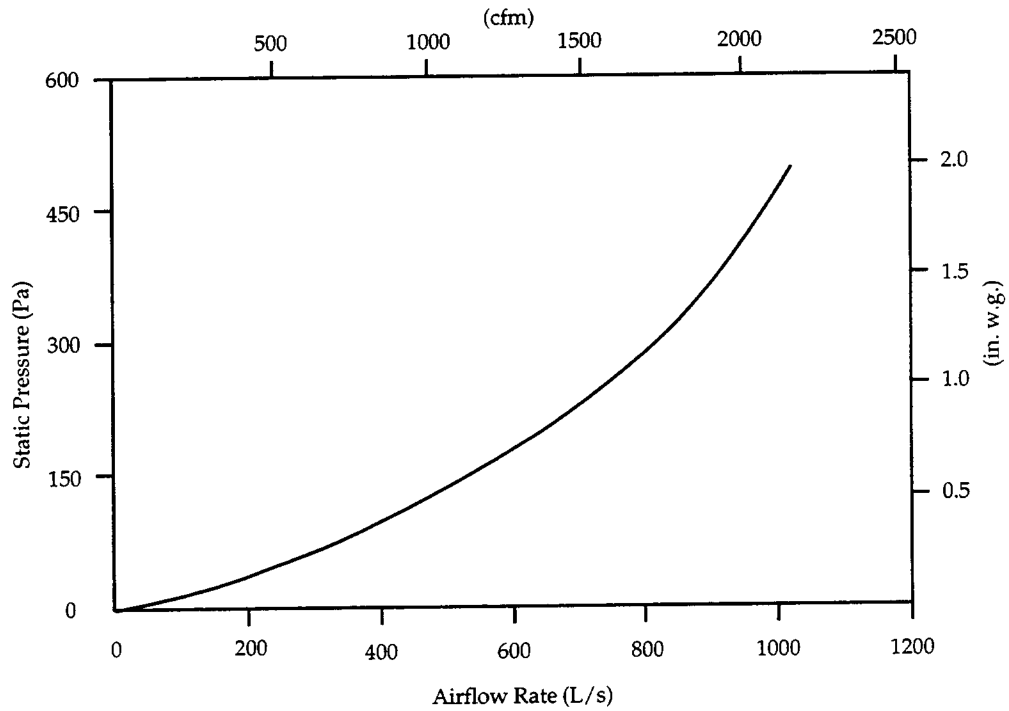

Figure 2.3 Example of a System Curve

When a ventilation system fan is operating, it raises the air pressure to overcome the airflow resistance of the system. The intersection of the system curve and the fan performance curve defines the point at which the pressure across the fan and through the system are equal, and thereby defines the airflow rate. If the airflow resistance of the system is accurately estimated during the design and the fan is properly selected and installed, then the point of intersection will be at the design airflow rate of the system. Figure 2.4 shows the intersection of a system curve S and a fan performance curve F.

If the system resistance increases, for example as filters become dirty, then a new system curve S′ replaces the original system curve S. The fan and system curves will intersect at a higher pressure difference and a lower airflow rate. The airflow rate can be returned to its design value by increasing the fan speed, such that a new fan curve F′ is in effect.

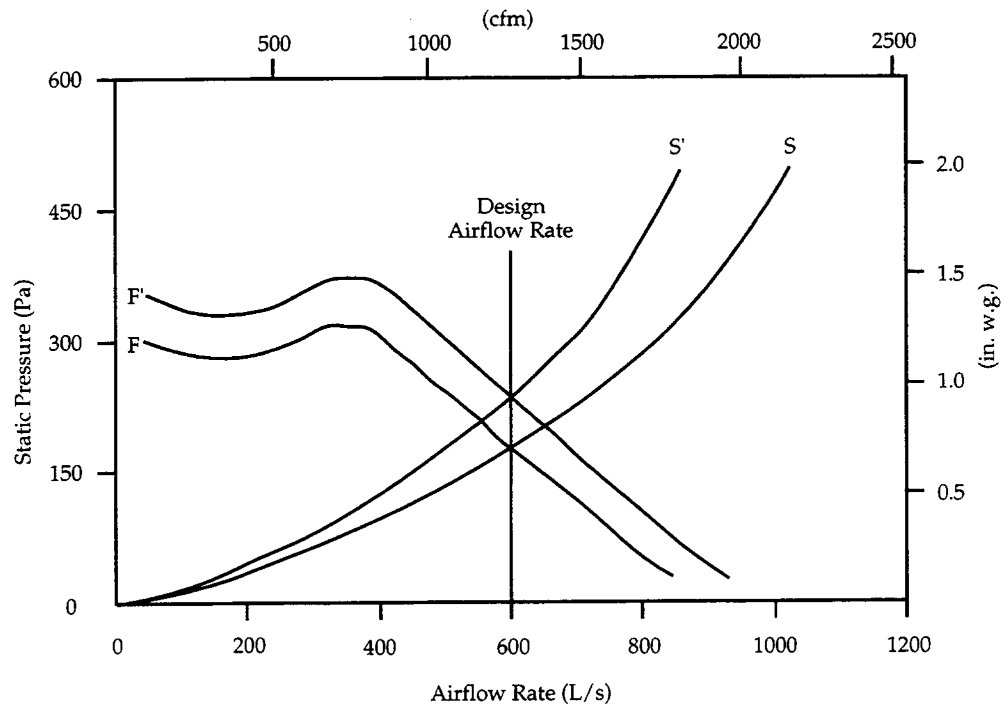

Figure 2.4 Example of the Interaction Between System and Fan Curves

The design airflow rate may not occur in practice due to the *system effect*,a condition where the duct arrangement at a fan inlet or outlet reduces the airflow rate through the fan. If elbows, tees and transitions are located too close to the inlet or outlet, then they can cause swirl or non-uniform flow. Correction factors are available for common system effects (AMCA 1990).

Filters: Filters remove dirt and debris from ventilation airstreams, keeping ventilation system components clean and reducing indoor particulate levels. Good filtration reduces energy consumption by keeping heating and cooling coils clean, reduces HVAC maintenance costs, improves occupant comfort and health, and reduces cleaning costs in the occupied space. Particulates are discussed in the ASHRAE Fundamentals Handbook (1993), and systems for removing particles are discussed in the ASHRAE HVAC Systems and Equipment Handbook (1992).

There are three basic types of air filters: fibrous media unit or panel filters, renewable media filters, and electronic air cleaners. There are two types of panel filters, *viscous impingement* and *dry media*. Both are available in a range of particulate removal efficiencies. Viscous impingement filters are made of coarse fibers coated with a viscous substance, such as oil, that collects the particles. Dry media filters consist of fibrous mats or blankets, and are sometimes pleated to increase the ratio of filter area to face area. The efficiency depends on the media thickness and density.

Renewable media or *roll filters* have a fibrous media that is fed manually or automatically into the airstream from a roll. Renewable filters can be dry or viscous, and a range of efficiencies are available.

Electronic air cleaners use electrostatic precipitation to remove particles from the airstream. These devices have an ionization section where the particles pick up an electric charge and a collection section where the charged particles are deposited on charged plates. Prefilters are often employed to collect larger particles so that the air cleaner does not need to be cleaned as often.

Filter efficiencies are determined with test methods described in ASHRAE Standard 52.1 (1992). These test methods yield two ratings, the *atmospheric dust spot efficiency* and the *arrestance percentage*. The dust spot efficiency is determined by measuring the ability of the filter to reduce the discoloration of a filter paper. The arrestance percentage is determined by measuring the ability of the filter to remove a synthetic test dust from an airstream. While both the dust spot efficiency and the arrestance percentage are reported on a scale from 0 to 100, their values are not interchangeable. A filter with an arrestance percentage of 95 may have a dust spot efficiency of only 40%.

The efficiency of high efficiency or HEPA filters is measured using the Thermal DOP method in U.S. Military Standard, MIL-STD-282 (1956). This standard determines the efficiency of the filter to remove small particles. DOP efficiency percentages are different from dust spot efficiencies. A filter with a dust spot efficiency of 90 may have a DOP efficiency of only 50%.

The pressure difference across panel and roll filters increases as they get dirty. When the pressure drop across filters increase to a recommended maximum value, panel filters should be changed and roll filters should be advanced. Design pressure differences, both clean and dirty, are usually available from filter manufacturers. When evaluating a ventilation system, it is important to determine whether the filter is overloaded and whether the pressure drop across the filter is so high that the ventilation system airflow is restricted. If an overloaded filter is not changed, the filters can "blow out," and unfiltered air will flow through the ventilation system and into the building.

<u>Heating and Cooling Coils</u>: Heating and cooling coils control the supply air temperature and come in a variety of arrangements (Bevirt 1984). Coils are selected based on their heat transfer performance and their resistance to airflow. The coil design and arrangement, i.e., fin size, fin spacing and ratio of surface area to tube area, determine the pressure drop across the coils. When cooling coils are wet with condensation, their resistance to airflow increases. Coil manufacturers usually provide wet and dry pressure drops across coils. This pressure drop can be measured during a ventilation system evaluation and compared to the design value. The pressure drop will be higher than the design value if the coils are dirty, perhaps because of poor filtration.

Dampers: Dampers are used to control the rates of outdoor air intake, return air recirculation, and exhaust to the outdoors. In addition, dampers are used in branch and zone ducts to control air distribution. Dampers are a significant resistance to airflow in the system, and their resistance relative to other resistances in the system affect the ability to control airflow rates.

Most ventilation system dampers are multi-blade, either parallel blade or opposed blade. As the damper closes, the resistance to airflow increases, but the change in airflow is not usually linear with the damper position in degrees. Dampers have flow characteristic curves that relate the airflow rate through a damper to the damper position (Bevirt 1984).

Controls: HVAC control systems modulate system variables (e.g., heating and cooling fluid temperatures and airflow rates) to meet thermal loads in the building (Bevirt 1984). Control system components include: sensors that measure the controlled variable; controllers that compare the sensor outputs with setpoint values and produce an output signal to cause control actions; and controlled devices such as valves, dampers, relays and motors.

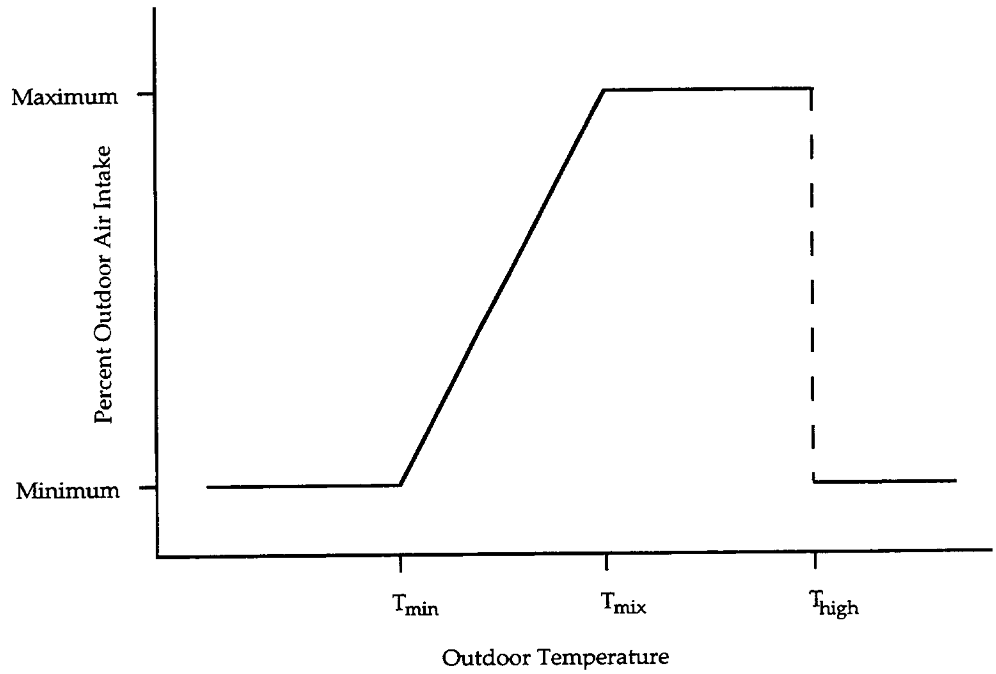

Figure 2.5 Schematic of Mixed Air Control Strategy

Airflow rates in ventilation systems are controlled by modulating damper positions and controlling fan speeds using one of several control approaches (Bevirt 1984). *Mixed air control* is the modulation of the outdoor, return and exhaust dampers to maintain the temperature of the mixture of outdoor air and recirculated return air (the mixed air temperature) at a setpoint value T_{mix}. As seen in Figure 2.5, the percent of outdoor air in the supply air is controlled as a function of outdoor air temperature. As the outdoor air temperature increases from the minimum setpoint T_{min} to T_{mix}, the percent outdoor air is increased by closing the return air damper and opening the outdoor air and exhaust dampers. When the outdoor temperature is equal to the mixed-air setpoint, the percent outdoor air intake is at its maximum value, usually 100%. This operating condition is called an *economizer cycle.* There is usually a high temperature limit T_{high} at which the system returns to minimum outdoor air intake because the outdoor air is too warm to cool the building. The outdoor air intake rate should meet the minimum ventilation requirements of the space served by the air handler under all conditions. Minimum outdoor air intake is maintained by a minimum outdoor damper position or a set of minimum dampers that are always open.

In, *face and bypass control* the supply airstream is split into two flows. One supply airstream passes through a heating or cooling coil with a face damper, and the other flows around the coil through a bypass damper. In *variable air volume control,* the airflow rate of constant temperature supply air is varied to meet the thermal loads in the space. The airflow rate control is based on either the air temperature in the zone or the static pressure in a terminal unit or in the space. *Warm-up cycles* are sometimes used in the morning to increase the temperature in the building prior to occupancy. These cycles usually involve no outdoor air intake to reduce energy consumption.

<u>Ductwork</u>: Ductwork distributes supply air throughout a building and is a major resistance to airflow in the ventilation system. Design information is available for sizing ductwork and calculating the pressure drop through air distribution systems (ASHRAE 1989, Bevirt 1984, SMACNA 1990). During the system design, design airflow rates through the various duct branches are determined based on the thermal load and outdoor air requirements of the building zones. These design airflow rates are compared to measured airflow rates during a ventilation assessment.

<u>Terminal Units</u>: A *terminal unit* is a device that connects the supply air ductwork to the supply outlets in the conditioned space. There are many types of terminal units, and they may contain dampers, fans, cooling and heating coils, and control devices (ASHRAE 1992, Bevirt 1984, NEBB 1986). Terminal units may be pressure dependent, pressure independent, constant or variable air volume, or fan powered.

Single-duct units provide supply air to the space from a single supply duct system, usually at a cool temperature. They may simply contain a volume damper that is adjusted manually when the system is balanced. Sometimes they contain a reheat coil for additional temperature control. *Dual-duct units* mix air from the hot and cold decks of a dual-duct system to meet the thermal loads of a zone.

Bypass boxes meet the thermal loads in a space by dumping excess supply air into the ceiling plenum. In *induction units*, primary air from the air handler flows through nozzles, inducing air from the room or plenum into the primary airstream. In some induction units, particularly those serving perimeter zones, the secondary air from the room is pulled through heating or cooling coils. *Pressure dependent units* contain a damper that is controlled manually or automatically in response to a thermostat. If the inlet pressure to the box changes, the airflow rate also changes. *Pressure independent units* use an automatic, constant-volume regulator that provides the same airflow rate regardless of the static pressure at the inlet.

Variable air volume (VAV) units control the airflow rate of constant-temperature supply air in response to the zone thermostat. The airflow rate varies between a minimum and maximum depending on the load within the zone. The minimum setting is important for maintaining thermal comfort and indoor air quality, and in some cases the minimum is set to zero flow. There are many different types of VAV boxes including pressure dependent and independent, single- and dual-duct, induction and fan powered. *Fan powered units* contain a fan and have an air inlet from the ceiling return plenum. The fan recirculates return air and mixes it into the supply air from the air handler. These units meet the thermal load in a zone by varying the amounts of cool primary air and recirculated return air.

<u>Air Outlets and Inlets</u>: Supply air outlets deliver conditioned air to the occupied space, and return or exhaust inlets remove air from the space. The design and performance of supply air outlets has been extensively studied (ASHRAE 1993, Nevins 1976). Supply outlets are selected to provide uniform temperatures in the occupied space, and to control drafts and noise levels.

Most commercial buildings use high velocity supply outlets or diffusers that induce room air into the primary supply airstream. The supply air is too cold and the outlet velocity is too high for thermal comfort in the occupied space. By inducing room air into the supply airstream, the temperature is increased and the air speed reduced. The performance of a supply air outlet depends on the supply air temperature, airflow rate, discharge angle and space loads.

When using high-velocity supply diffusers, the location of the return air inlets does not significantly affect air motion within the room. The inlets are selected and located based largely on architectural considerations. Unlike supply air outlets, the system design does not include a design airflow rate for return vents. In most commercial buildings, the suspended ceiling plenum is part of the return system, and the return inlets are simply openings in the suspended ceiling. Exhaust air inlets are used with exhaust systems to remove air from spaces such as bathrooms and copy rooms.

2.3 Building Ventilation

Ventilation system performance, indoor air quality and energy consumption are all affected by outdoor airflow into buildings and airflow within buildings. These airflows and their effects need to be understood when evaluating ventilation system performance. This section discusses building ventilation in relation to ventilation system performance evaluation. Additional discussion is contained in the ASHRAE Handbook of Fundamentals (1993). The section begins by defining several terms to lessen some of the confusion that often occurs in discussions of ventilation. The section on definitions is followed by a discussion of ventilation problems in buildings. The causes and impacts of outdoor airflow into buildings and airflows within buildings are then discussed. The last section presents some key issues in the area of building ventilation.

2.3.1 Definition of Terms

ASHRAE (1989) defines *ventilation* as the "process of supplying and removing air by natural or mechanical means to and from any space". The term is sometimes used to describe only the outdoor air provided to a space. It is also used to describe the total of the outdoor air and recirculated air. Confusion between outdoor air and supply air can be avoided by using the terms outdoor air ventilation and supply air or total ventilation instead of just ventilation. Ventilation rates are given in units of L/s (cfm) or as air change rates in units of air changes per hour (or ach). An air change rate is a ventilation rate in units of L/s (cfm) divided by the volume of the space.

Mechanical ventilation is ventilation provided by fans. Mechanical ventilation provides outdoor and supply air to a space and removes air from a space for recirculation by the ventilation system or exhaust to the outdoors. *Natural ventilation* is ventilation through intentionally provided openings, such as open windows and vents.

Infiltration is the flow of outdoor air into a space through unintentional openings such as cracks around window frames and wall-floor joints. Infiltration is driven by pressure differences across these openings caused by wind, inside-outside temperature differences and the operation of ventilation equipment. Exfiltration is airflow from a space to the outdoors through unintentional openings. Infiltration is often used to describe both infiltration and exfiltration. Infiltration rates are usually given in air changes per hour.

Air leakage is the airtightness of a building envelope and is independent of weather conditions and ventilation system operation. It is measured with a fan pressurization test using ASTM Standard E779. The air leakage is usually expressed as the building air change rate at a reference pressure difference across the building envelope or as an effective leakage area that accounts for all the leaks in the building.

The *air change rate* of a building is one of the above quantities in units of air changes per hour. It is used to describe the total or supply ventilation airflow rate, the mechanically-induced rate of outdoor air intake, the infiltration rate or the sum of all outdoor airflows into the building. Confusion is avoided if the air change rate is described as infiltration, total outdoor air or supply air.

2.3.2 Ventilation Problems

Even though most indoor air quality investigations do not evaluate building ventilation, it is blamed for many indoor air quality problems. It is often stated that about 50% of indoor air quality problems are caused by inadequate ventilation. This percentage may or may not be accurate, but the lack of information on building ventilation in most investigations makes this statistic questionable. Furthermore, *inadequate ventilation* is a vague term and tells us little about what is wrong. There are many different ventilation problems in buildings that affect indoor air quality, energy consumption and thermal comfort. Table 2.1 lists several of these problems and their impacts.

Problem	Impact
Too little outdoor air intake	Indoor air quality
Too much outdoor air intake	Energy consumption
High supply airflow rates	Thermal comfort
Low supply airflow rates	Thermal comfort
Inappropriate supply air properties (temperature and relative humidity)	Thermal comfort
Poor air distribution	Indoor air quality Thermal comfort
Too much infiltration	Indoor air quality Energy consumption Thermal comfort Ventilation system performance
Poor intake air quality	Indoor air quality
Undesired interzone airflows	Indoor air quality

Table 2.1 Ventilation Problems

Too little outdoor air intake increases indoor contaminant levels and the potential for indoor air quality problems. But too much outdoor air intake can lead to excessive energy consumption. Supply airflow rates that are too high or too low affect thermal comfort in the occupied space. Similarly, if the supply air temperature and relative humidity are very different from their setpoints, thermal comfort can suffer. Poor air distribution is the delivery of the supply air to the different zones of a building at rates that do not meet the requirements of the zones. If too little supply air is provided to a space, it will not receive enough outdoor air and may have poor thermal comfort.

Too much infiltration increases energy consumption and can interfere with the operation of ventilation systems. It also affects indoor air quality and thermal comfort because infiltrating air is not filtered and not heated or cooled. Poor intake air quality, i.e., high pollutant levels in the air being brought into a building, decreases indoor air quality. Finally, poor control of ventilation airflows can cause undesirable airflows between building zones. For example, airflows out of spaces such as bathrooms and garages can move contaminants into occupied spaces.

2.3.3 Air Change with Outdoors

Outdoor airflow into mechanically ventilated buildings includes both outdoor air intake via the air handling system and infiltration through leaks in the building envelope. Ideally the outdoor air intake rate is controlled, but the infiltration rate and the distribution of infiltration in a building is always uncontrolled. Commercial building infiltration rates are usually assumed to be small compared to outdoor air intake rates. However, field measurements have shown that infiltration and intake rates can be similar in size (Grot and Persily 1986, Persily and Norford 1987).

Outdoor air intake rates depend on the ventilation system design and operation, and vary with time depending on the system controls. While the system design may call for a specific rate of outdoor air intake under certain circumstances, the actual damper positions and air handler operation may cause the outdoor air intake rate to be very different from the design value. If the system is properly balanced and maintained, such differences will be minimized.

Infiltration is driven by pressure differences across leaks in the building envelope. These pressure differences are caused by wind, indoor-outdoor temperature differences and ventilation system operation. Envelope infiltration rates are usually assumed to range from zero to 0.2 air changes per hour in commercial buildings. However, because of leaky envelopes and poor ventilation system control, infiltration rates can be as high as 0.5 air changes per hour (Grot and Persily 1986). Pressurization tests of commercial buildings have shown that they are as leaky as typical U.S. residential buildings (Persily and Grot 1986). The infiltration rate of a mechanically ventilated building depends on the amount of envelope leakage, the location of these leaks, and the pressure differences across the leaks. Envelope leakage occurs at many locations over the building envelope, with most of the leaks at interfaces between envelope components such as window-wall and floor-wall intersections. The distribution of these leaks over the envelope depends on the envelope design, construction quality and deterioration over time. There is a relationship between the pressure difference across each opening and the airflow rate through it, and this pressure-flow relationship depends on the shape and size of the opening.

Pressure differences across building envelope leaks are caused by indoor-outdoor temperature differences (sometimes called the stack effect), wind, and ventilation system operation. If there is a temperature difference between the inside of a building and the outdoors, there will be a pressure difference across the envelope that varies with height. Under heating conditions, air will flow into the building at lower levels and out of the building at higher levels. During the cooling season, the directions of the pressure differences and airflows are reversed. The size of the stack pressures depends on the building height, the indoor-outdoor temperature difference and the resistance to vertical airflow within the building caused by the interior walls and floors.

Pressure differences across the envelope are also caused by wind, with higher pressures on the windward side and lower pressures on the other sides of the building. Wind-induced pressures vary over the building and depend on the wind speed, direction and obstructions surrounding the building.

Pressure differences are also caused by the operation of ventilation systems. If there is more supply airflow into a space than return airflow out (or vice versa), this will contribute to the pressure difference across the exterior walls. These pressure differences can be larger than stack and wind pressures. Most commercial buildings are designed to have more supply air than return air to reduce envelope infiltration and to ensure the proper operation of exhaust air systems. Good system maintenance is required to achieve this design goal.

2.3.4 Airflow Inside Buildings

Airflow within buildings is an important means of pollutant movement and can transport contaminants to spaces within buildings that are far from the pollutant sources. Airflow rates within buildings depend on the number and location of internal leaks, the pressure differences across these leaks, and the relationships between airflow rate and pressure difference for these leaks. The pressure differences which cause these airflows are caused mainly by the stack effect and ventilation system operation. The stack effect that pulls air into the building on lower floors during heating, also causes air to flow from lower floors to upper floors. When the building is being cooled, the direction of the pressure differences and airflows is reversed.

Ventilation system operation causes air movement between different spaces when the supply airflow into a space is different from return airflow out. If the supply airflow rate is higher, the space will be at a higher pressure than adjoining spaces, unless they have an even larger supply air excess. Air will flow from such a pressurized space through any available opening. Similarly, a space with excess return or exhaust airflow will be at a negative pressure, and air will flow into the space. System-induced pressure differences and airflows can be complex, with many spaces involved in the pressure and airflow interactions.

2.3.5 Key Issues

Airflow into and within multi-zone, mechanically ventilated commercial buildings can be very complicated. Understanding the following issues can be helpful when evaluating ventilation system performance.

- **Performance Versus Design**
- **Impact of Ventilation System Operation**
- **Stack Pressures and Vertical Airflow Paths**
- **Pressure Relationships Between Spaces**
- **Air Distribution and Ventilation Effectiveness**
- **Space Use Changes and Ventilation Requirements**

Performance Versus Design

Ventilation systems do not always perform as intended, even if the system design is very good. Differences between design and performance are caused by problems in the system installation, operation and maintenance. When assessing ventilation, it is still important to understand the system design to determine how it was intended to operate and perform. But because of potential differences between system performance and design, system airflows and other performance parameters must be measured when assessing building ventilation.

Impact of Ventilation System Operation

Mechanical ventilation system operation has significant impacts on outdoor air change rates of buildings and airflows within buildings. Obviously, the system brings outdoor air into the building through the air handler and may be designed to move air from room to room. However, system operation can also induce pressure differences across exterior walls and interior partitions.

An imbalance between the outdoor air intake and exhaust airflow rates for a building, will cause infiltration or exfiltration across the building envelope. Excess outdoor air intake will cause the building to be at a positive pressure, and the excess air will be forced out of the building through openings in the building envelope. If the exhaust airflow rate is larger than the intake rate, then the building will be at a negative pressure and excess air will be pulled into the building through envelope openings. System-induced pressure differences can dominate stack and wind pressures, and depend on how the ventilation system is operating, i.e., the percent outdoor air intake and the supply airflow rate. Under different modes of system operation, ventilation flow imbalances can vary in both magnitude and direction. As mentioned earlier, ventilation systems are often designed to maintain a positive pressure difference across the building envelope to reduce air infiltration. However, this excess supply air may not occur in practice if there are deficiencies in the system design, installation and maintenance.

Because supply and return airflow rates can be different in different building zones, system-induced pressures can vary in direction and magnitude over the surface of a building. An example of such pressure variations occurs in office space with suspended ceiling plenums. Air is supplied below the suspended ceiling, while return air flows from the ceiling plenum. Therefore, there can be exfiltration from the office space below the suspended ceiling and infiltration into the return air plenum above the ceiling.

Ventilation system airflow imbalances can also cause interzone airflow and pollutant transport within a building. Spaces with more supply airflow than return or exhaust will be at a higher pressure than adjoining spaces, unless the other spaces have an even larger excess of supply air. Air will flow out of a pressurized space through any available openings. Similarly, spaces with an excess of return or exhaust airflow will be at a negative pressure, and air will be pulled in from adjoining spaces. System-induced pressure differences and airflows within a building can be very complex, and unexpected and undesirable airflow patterns can occur. One example is an office space with more return airflow than supply that is next to a smaller space with its own exhaust system, e.g. a toilet or photo lab. Even though there is an exhaust system in the smaller space, the excess of return airflow in the office may still pull air and pollutants into the office space from the toilet or photo lab.

Stack Pressures and Vertical Airflow Paths

Air infiltration and internal airflow in commercial buildings are strongly affected by stack pressures. Stack pressures increase envelope leakage rates, interfere with the ability of ventilation systems to pressurize buildings, and cause vertical airflow within buildings. The size of these pressures depends on the indoor-outdoor temperature difference, the building height and the internal resistance to airflow. Increased resistance to vertical airflow in a building decreases the stack pressures. This resistance is caused by building floors and by openings on individual floors to vertical shafts such as elevators, stairs and ventilation chases. Plumbing and electrical chases and vertical passages in the building envelope construction can also be important. If there is no internal resistance to vertical airflow, building stack pressures will be at their maximum values, which depend only on temperature difference and building height. Field measurements in tall buildings have found stack pressures to be 80% to 90% of the values that would be expected with no vertical airflow resistance (Tamura and Wilson 1967).

The impact of stack pressures are well known to those who design smoke control systems in tall buildings and those who deal with pressure differences at building entrances. Vertical airflows that result from stack pressures can move a lot of air and pollutants within a building, and this phenomena must be recognized when dealing with indoor air quality problems.

Pressure Relationships Between Spaces

The pressures between building spaces are important in ventilation system design and performance. In some situations, the system design specifies that a space be maintained at a pressure difference relative to adjoining spaces. Pressure requirements are common for toilets, print shops, smoking lounges and photo labs. These pressure differences are important for preventing the movement of airborne contaminants to other building spaces. Meeting these requirement in practice depends on the design and performance of the exhaust system and the ventilation airflows in the adjoining spaces.

Unintentional pressure differences also exist between spaces and can be important mechanisms of air and pollutant transport within buildings. Such pressure differences can arise from the stack effect and from system-induced airflow imbalances. The existence of such pressure differences can move air and contaminants at large rates and over large distances. Their potential impact must be realized when evaluating building ventilation and when diagnosing the cause of an indoor air quality problem.

Air Distribution and Ventilation Effectiveness

Ventilation effectiveness has received much attention in discussions of ventilation system performance problems in buildings. Most references to ventilation effectiveness concern the distribution of ventilation air within a room or space and the flow of air between supply vents and return or exhaust vents. There are concerns that large quantities of supply air may flow directly into the return or exhaust without reaching the occupied space, so-called *short-circuiting*. Although only limited field measurements have been performed in commercial buildings, these measurements have shown that ventilation air mixes well within rooms (Fisk et. al. 1991). A standardized test method for determining ventilation effectiveness within rooms is still being developed, but there is currently little evidence of short-circuiting within mechanically ventilated commercial buildings.

Ventilation effectiveness can also be thought of as the ability of a ventilation system to achieve its design goals, including the distribution of ventilation air as specified in the design (Persily 1992). And although short-circuiting of ventilation air does not appear to be a problem in commercial buildings, poor air distribution can be a source of performance problems. Ventilation system designs specify supply airflow rates to different zones within buildings: whole floors, portions of floors and rooms. Field tests have shown that airflow rates to individual zones can be well below design values. These performance problems are more common and probably more important than short-circuiting within rooms. In these situations, the system will not deliver the required amounts of supply and outdoor air to a space, creating the potential for poor thermal comfort and elevated contaminant levels.

Space Use Changes and Ventilation Requirements

Ventilation system design specifications for a building or a space are based on the activities that will be occurring in that space and the occupancy levels. If the space use changes, the system design may not be able to meet the new ventilation requirements of the space. Such situations occur when new tenants move into a building, existing occupants change their activities and occupancy schedules, and the amount of office equipment increases. These changes often require changes in the ventilation system. When evaluating ventilation system performance, it is important to understand both the original design assumptions and any changes in the building that have occurred since the design.

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3 INSTRUMENTATION

This section describes the instrumentation used in the ventilation assessment techniques discussed in this manual. Accuracy requirements are included for evaluating manufacturer's specifications of these instruments. When using these devices in the field, the actual measurement errors will typically be larger than these accuracies. Information on cost is also included, but it is rough since the cost depends on features such as accuracy, range and response time. Some of the instrumentation has been traditionally used in testing, adjusting and balancing and is described in the references (ASHRAE 1988, Bevirt 1984, NEBB 1986 and 1991, SMACNA 1983). Hand-held, digital instruments have more recently become available and are much easier to use. Some of these instruments combine the measurement of several factors such as temperature, relative humidity, air speed and pressure difference and are capable of datalogging and averaging. These multi-channel systems range in cost from about $1000 to $3000.

3.1 Air Temperature

The temperature of indoor and outdoor air and ventilation system airstreams can be measured with digital electronic thermometers. These hand-held, battery-powered devices employ either a thermocouple, thermistor or RTD (resistance temperature detector). A variety of probes are available that differ in response time and measurement range. Digital electronic thermometers cost from $100 to $250. Devices that can also measure relative humidity cost from $200 to $400.

When performing a large number of measurements, a response time of 30 seconds or less is recommended. The accuracy of the air temperature readings should be 0.3 °C (0.5 °F). Digital thermometers should be calibrated every 12 months using a laboratory grade thermometer.

When measuring outdoor air temperatures, one must use a probe with an appropriate range, and not a probe intended for indoor temperatures. Also, air temperature measurements can be affected when surrounding surfaces are at significantly different temperatures from the air. For example, outdoor temperature measurements are affected by solar radiation, and indoor measurements can be influenced by nearby windows. Temperature probes with radiation shields are available to reduce these errors.

3.2 Relative Humidity

The relative humidity of indoor, outdoor and ventilation air can be measured with digital thermohygrometers, sling psychrometers or digital psychrometers. Digital thermohygrometers are hand-held, battery-powered devices with sensors that respond to a change in relative humidity with a change in resistance or capacitance. The device converts the sensor response into a digital output in percent relative humidity (%RH). Sling and digital psychrometers consist of two thermometers, one with a moistened wick around the bulb. When air flows across the bulb with the wet wick, this thermometer provides the wet-bulb temperature. For sling psychrometers, the airflow across the wet bulb is achieved by whirling the device through the air. A psychrometric chart is used to convert the dry and wet-bulb readings to relative humidity . Digital psychrometers have a battery-powered fan to provide the airflow and a digital readout of relative humidity. Digital thermohygrometers cost from $200 to $500, sling psychrometers from $50 to $100, and digital psychrometers from $100 to $200.

Digital thermohygrometers are easier to use than psychrometers, but they require careful calibration. The response time of these devices is important in field work and should be 30 seconds or less. The accuracy requirement for these devices is 2% RH over a range of 0 to 90% RH, and they need to calibrated every 6 months or whenever they are exposed to extreme humidities.

When using psychrometers, the temperature measurements need to be accurate within 0.5 °C (1 °F). Psychrometers require an air velocity from 5 to 7.5 m/s (1000 to 1500 fpm) across the wick to produce accurate readings. Significant errors occur if the wick is dirty. Therefore, the wick should be inspected and changed often, and distilled water should be used to moisten the wick.

3.3 Differential Pressure

Differential pressure is measured across ventilation system components and building partitions, and when performing pitot tube traverses. These measurements can be made with digital electronic pressure manometers, inclined manometers, and diaphragm-type gauges.

Digital electronic pressure manometers come in a variety of measurement ranges. Devices with very low ranges can be used to measure velocity pressures at air speeds well below the 3 m/s (600 fpm) limit for inclined manometers. Some electronic manometers can average a series of pressure readings. Since pressures in ventilation ductwork usually fluctuate, automated averaging is very helpful. Digital pressure gauges used for velocity pressure measurements need to be accurate within 2% of the reading. Pressure differences across interior and exterior partitions can be very small, on the order of 1 to 5 Pa (0.004 to 0.02 in. w.g.), and a gauge with an appropriate range and resolution must be used in these applications. Digital manometers range in price from $750 to $1500 depending on the accuracy and features of the device.

Inclined manometers are standard equipment in testing and balancing. They come in different scales, and are sometimes combined with vertical manometers to read higher pressures. Inclined manometers are only applicable to pressure differences greater than 5 Pa (0.02 in. w.g.), i.e., for velocity pressures corresponding to air speeds above 3 m/s (600 fpm). Inclined manometers must be leveled prior to use and take a long time to equilibrate when used with a pitot tube. Their cost ranges from $100 to $200 depending on the pressure scale.

Diaphragm-type pressure gauges, magnetic linkage or Magnehelic® gauges, have a magnetic spring that rotates when a pressure difference displaces a diaphragm. These devices are available in a variety of ranges, the lowest being 0 to 60 Pa (0.24 in. w.g.). These devices have an accuracy of 2% of full scale, limiting their ability to measure low pressure differences (less than 15 Pa (0.06 in. w.g.)). Magnetic linkage gauges should be used in the midrange of the measurement scale and in a vertical position. They should be checked against a known pressure source every time they are used. In addition to on-site accuracy checks, these gauges should be calibrated every 12 months. Controversy exists as to whether they should be mounted on a vibrating surface. ASHRAE (1988) says they should, and NEBB (1991) says they should not. These gauges cost about $50.

3.4 Static Pressure Taps

Static pressures are measured across ventilation system components and across interior partitions by connecting a pressure gauge to static pressure taps. When making measurements in ventilation systems, the airflow must not create a velocity pressure at the static pressure tap. To avoid velocity effects, static pressure taps must be designed so that the opening is at a 90° angle to the airflow. The opening must have sharp edges and be free of burrs. A variety of different static pressure taps are available from simple through-wall taps to static pressure tips that are inserted into the airstream (NEBB 1986). Static pressure taps are sometimes installed in duct walls during the system installation, for example to be able to monitor the pressure difference across a filter bank. Additional taps can be installed during a ventilation system evaluation or a rigid sensing tube can be inserted through a hole the duct wall. Such a tube must be constructed to avoid flow effects, and the tube opening must be positioned perpendicular to the flow stream. The static pressure tap of a pitot tube can be used, but flexible sensing tubes and careless positioning will cause measurement errors.

When measuring static pressure differences between interior spaces, the design and position of the pressure tap is less critical because of the low air velocities. A flexible tube open to the space is adequate, but the tube must not be compressed, for example by a closed door. Compressing the tube can increase the pressure within the tube and lead to an inaccurate measurement.

3.5 Pitot-Static Tubes

Pitot tubes are used to measure velocity pressures in ventilation airstreams. A pitot tube has one opening in its tip that is pointed directly into the airstream and is connected to a total pressure port at the base of the tube. Additional openings are located around the tip that are perpendicular to the airstream and are connected to a static pressure port. The total and static pressure ports are connected to a differential pressure gauge to measure the velocity pressure. The ports can also be used to measure the total and static pressure in the airstream.

Drawings of pitot tubes, showing critical dimensions, are provided in a variety of references (AMCA 1990, ASHRAE 1993, Bevirt 1984, NEBB 1986). Pitot tubes come in a variety of lengths for use in ducts of different dimensions. When using a pitot tube, the tip must be pointed into the airflow. Pitot tubes must be treated with care, as the tips are carefully machined to prevent turbulence effects.

3.6 Hot-Wire Anemometers

Hot-wire or thermal anemometers are used to measure air speeds in ducts. These devices use a probe with a heated wire that is either supplied with a constant current or kept at a constant temperature with a variable current. In constant-current devices, the wire's electric resistance depends on air speed. In constant-temperature devices, the current required to keep the probe at a constant temperature is related to air speed. The probes are very sensitive and can measure air speeds as low as 0.05 m/s (10 fpm). Hot-wire anemometers come in a variety of measurement ranges, have measurement accuracies of about 2% of full scale, and should be calibrated every 12 months. They cost from $500 to $1000 depending on their accuracy and additional features. These probes are usually directional and must be positioned properly in the airstream. The probes are also delicate and must be kept clean.

3.7 Rotating Vane Anemometers

Rotating vane anemometers consist of lightweight propellers and are used to measure airflow rates at air outlets and inlets, dampers, and filter and coil banks (ASHRAE 1988, NEBB 1986). Digital meters display the average air speed over a set time period; analog devices display the instantaneous air speed. Mechanical devices are available that display the linear meters (feet) of air passing through the propeller. Mechanical vane anemometers are used with a timer to convert the length into the average air speed over the measurement period, and usually require a correction factor to compensate for friction. Mechanical devices are applicable to air speeds between 1 and 10 m/s (200 and 2000 fpm), while electronic devices can be used below 1 m/s (200 fpm). Electronic vane anemometers cost from $500 to $1000. Vane anemometers are accurate within 5% of reading when the air speed is 1 m/s (200 fpm) or more, and within 10% of reading for readings below 1 m/s (200 fpm). These devices should be calibrated every 6 months.

To determine the airflow rate at coils and at air outlets and inlets, the average measured air speeds must be multiplied by an application factor, "K factor" or "free area factor," designated as K or A_k, . These factors have been studied for different configurations of coils, outlets and inlets, and are sometimes available from the manufacturers of these components. The average air speed should be determined from multiple measurements made at several locations across the airstream, not by moving the instrument across the opening or coil to obtain an average reading over the measurement period (ASHRAE 1988, NEBB 1986).

3.8 Flow Hoods

Flow hoods measure airflow rates at supply outlets and return or exhaust inlets. They consist of a hood that fits over the outlet or inlet, and a measuring device at the base of the hood. The airflow rate is measured with a pressure or velocity sensor connected to an analog or digital meter that reads out directly in L/s (cfm). Flow hoods usually have several hood attachments to fit over outlets and inlets of different sizes. They cost from $1500 to $2500, depending on the number of hoods and the type of display. Smaller hoods are also available for small outlets and inlets in hard to reach locations, and these devices cost about $1000.

Flows hoods have ranges from about 25 to 1000 L/s (50 to 2000 cfm) and accuracies from 5 to 10% of reading. When using flow hoods in the field, the measurement errors can be significantly larger than these values, and they should not be used at air speeds above 10 m/s (2000 fpm). ASHRAE (1988) recommends their use to proportion flows between air distribution devices and not to certify system performance.

Flow hoods can change the direction of the airstream being measured, causing a static pressure drop and reducing the airflow rate. This effect is more important at higher airflow rates, and correction factors are sometimes available.

3.9 Tracer Gas Monitors

Tracer gas monitors are used to measure the tracer concentrations when using tracer gas techniques to study ventilation. Several different tracer gases are used in these techniques, such as sulfur hexafluoride (SF_6) and carbon dioxide (CO_2). The type of monitor will depend on the tracer gas and the range of concentration expected during the test. Tracer gas concentrations should be measured with an accuracy of 5%, and the monitor should be calibrated with each use.

ASTM Standard E741 (1993) is a test method describing how to make tracer gas measurements of building air change rates. It presents equipment requirements and measurement procedures, and discusses desirable properties of tracer gases. These properties include being inert, not absorbed on building materials and furnishings, easily and inexpensively measured at very low concentrations, nontoxic, nonallergenic, nonflammable, and nonexplosive.

3.10 Smoke Tubes

Smoke tubes are used to study airflow patterns within buildings and in rooms. They are also used for finding leaks in ducts and from airtight spaces. Smoke tubes come in several forms, including sticks and guns that contain titanium tetrachloride, a chemical that reacts with water vapor to produce smoke. Smoke candles are also available in a variety of sizes and burning times. The smoke produced by these devices can be irritating, and they must be used with care. They are generally not appropriate for use in occupied spaces.

3.11 References

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4 MEASUREMENT TECHNIQUES

This section describes techniques to measure ventilation system performance parameters including air temperature, pressure difference, relative humidity, percent outdoor air intake, airflow rate and building air change rate.

4.1 Air Temperature

Air temperature is measured in the occupied space, outdoors and in ventilation ducts using the instruments described in Section 3.1.

Indoor and Outdoor Air Temperature

Indoor and outdoor air temperatures are measured by positioning the temperature probe at the measurement location for a period of time that is based on the probe's response time. Radiation effects are avoided by using a temperature probe with a radiation shield. Outdoor air temperatures should be measured in the shade.

Air Temperature in Ventilation Ducts

Return and supply air temperatures are measured by positioning the temperature probe in the appropriate ventilation duct for a period of time that is based on the probe's response time. If there is temperature stratification in the duct, then the average temperature is determined from multiple temperature measurements.

Multiple measurements are made in ducts by dividing the duct cross section into twelve equal-area sections and measuring the air temperature in the center of each section. The measurement locations in rectangular ducts are arranged in three rows of four points as shown in Figure 4.1. The positions of the measurement locations depend on the height H and length L of the duct.

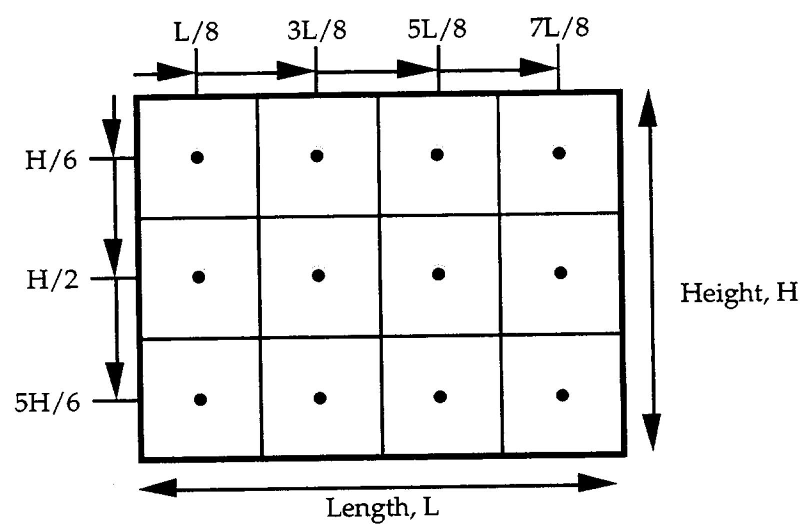

Figure 4.1 Temperature Measurement in a Rectangular Duct

In circular ducts, the twelve measurement points are located on two diameters, as shown in Figure 4.2. The positions of the points are based on the duct diameter D.

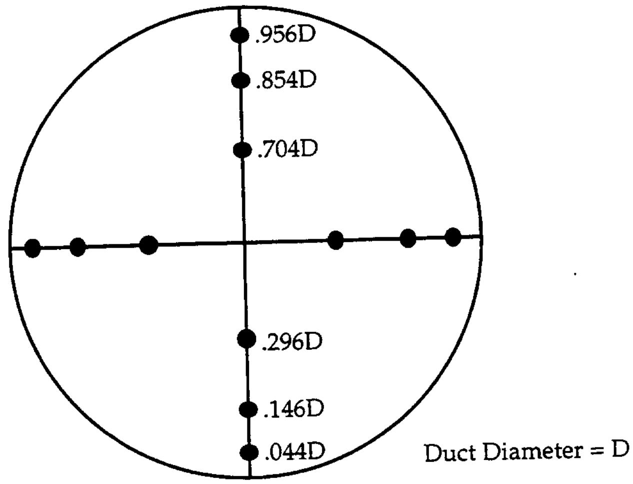

Figure 4.2 Temperature Measurement in a Round Duct

4.2 Barometric Pressure

Barometric pressure can be obtained from a nearby weather station, such as a local airport. However, barometric pressure readings from weather stations are usually given at sea level and must be corrected for the actual altitude of the building site. To convert a sea level barometric pressure to the pressure at another altitude, the sea level pressure is multiplied by a factor from the table in Appendix B.

Example: Correction of Barometric Pressure for Altitude

The barometric pressure at Tucumcari, New Mexico is reported to be 102 kPa (30.15 in. Hg) at sea level. Since Tucumcari is at an elevation of 1231 m (4048 ft), the table in Appendix B is used to determine a correction factor of 0.863. The reported barometric pressure is multiplied by 0.863, and therefore, the actual barometric pressure is 88 kPa (26.02 in. Hg).

4.3 Pressure Differences

Pressures differences are measured across components in air handling systems and across exterior and interior walls with differential pressure gauges (Section 3.3) and static pressure taps (Section 3.4).

Pressures Differences across System Components

Pressure differences across system components, such as filter banks or cooling and heating coils, are measured by connecting the two ports of a differential pressure gauge to static pressure taps located on either side of the component. The high pressure side of the gauge should be connected to the upstream pressure tap. These pressure differences usually range from 25 to 250 Pa (0.1 to 1 in. w.g.), and the gauge must have an appropriate scale. Pressure differences in ventilation systems are likely to fluctuate due to turbulence. Therefore, the pressure difference across a system component should be read six times, once every 10 seconds, and the measurements should be averaged.

Static Pressure Differences across Walls

To measure the pressure differences across interior or exterior walls, the two sides of a pressure gauge are connected to pressure taps on either side of the wall. These pressure taps can be the ends of tubes run underneath doorways or through other openings. The tubes must not be compressed, and this is avoided by positioning them carefully and using small diameter tubes. When measuring pressure differences across exterior walls, the pressure gauge should be located indoors to avoid temperature effects. Pressure differences across exterior walls should be measured under low wind speeds, unless one is interested in the effects of wind.

4.4 Relative Humidity

Relative humidity is measured in the occupied space, outdoors, and in ventilation ducts using the instruments described in Section 3.2.

Indoor and Outdoor Relative Humidity

Indoor and outdoor relative humidities are measured similar to air temperatures. The relative humidity probe is positioned at the measurement location for a length of time that is based on the probe's response time. Outdoor relative humidities should be measured in the shade.

Relative Humidity in Ventilation Ducts

Relative humidities are measured in ventilation ducts by positioning the probe for a period of time that is based on the probe's response time. If there is stratification in the duct, then the average relative humidity must be determined from multiple readings. The locations for multiple measurements are the same as those described for temperature in Section 4.1.

4.5 Air Density

As discussed in Section 2.1.2, air density depends on air pressure, dry-bulb temperature and relative humidity. ASHRAE (1993) describes how to calculate air density from these quantities, and ASHRAE Standard 111-1988 contains examples of these calculations. In ventilation system evaluations, the relative humidity correction to air density can be neglected. However, for air temperatures above 40 °C (105 °F), relative humidity should be calculated using the formulas in the ASHRAE (1993). For elevations below 150 m (500 ft), the air pressure does not need to be corrected for altitude.

For the purposes of ventilation performance evaluation, the air density ρ in kg/m^3 is determined within 2% using the following formula:

$$\rho = \frac{P_b}{(T + 273.15)} \times 3.47$$

where P_b is the barometric pressure in kPa corrected for altitude and T is the air temperature in °C. In inch-pound units, the air density in lb/ft^3 is expressed as

$$\rho = \frac{P_b}{(T + 460)} \times 1.32$$

where P_b is in units of in. Hg and T is in °F.

Therefore, to determine air density in the occupied space, outdoors or in ventilation ducts, the air temperature and barometric pressure are measured as described in Sections 4.1 and 4.2. The equations above are then used to calculate the air density.

Example: Air Density Calculation

Returning to Tucumcari, NM, the barometric pressure corrected for elevation is 88 kPa (26.02 in. Hg). The outdoor air temperature is 5 °C (41 °F), and the supply air temperature is 15 °C (59 °F). Using the density formulas, the outdoor air density is 1.10 kg/m^3 (0.069 lb/ft^3), and the supply air density is 1.06 kg/m^3 (0.066 lb/ft^3).

4.6 Tracer Gas Concentration

Tracer gas concentrations are measured in the occupied space, outdoors, and in ventilation ducts. The measurement techniques apply to all tracer gases, including carbon dioxide, and depend on whether the concentration monitor is portable or stationary. Portable monitors measure the tracer gas concentration at the sampling location. When using a portable monitor, the air must be sampled for a length of time that is based on the monitor's response time. Stationary monitors are used by collecting air samples in containers and bringing them to the concentration monitor for analysis. When collecting air samples for analysis with a stationary monitor, the volume of the air sample must be large enough to allow at least three concentration readings with the monitor. After the air samples are collected, they should be analyzed as soon as possible with the stationary monitor. Three concentration readings should be determined with the monitor, and their average should be calculated. When the tracer gas is carbon dioxide, the person making the measurements must avoid exhaling near the monitor or air sampling equipment.

Indoor and Outdoor Tracer Gas Concentrations

Tracer gas concentrations in the occupied space or in the outdoor air are measured by bringing a portable monitor to the measurement location or by collecting an air sample for analysis with a stationary monitor.

Tracer Gas Concentrations in Ventilation Ducts

Tracer gas concentrations are measured in ventilation ducts with portable or stationary monitors. If the concentration in the duct is stratified, then the average concentration must be determined from multiple readings. Stratification is more common in supply ducts where there can be incomplete mixing of the recirculation and outdoor air. The measurement locations are the same as those for temperature in Section 4.1.

4.7 Percent Outdoor Air Intake

The percentage of outdoor air in the supply airstream can be determined from an energy balance based on the temperatures of the outdoor, recirculation and supply airstreams or from a mass balance based on tracer gas concentrations in these three airstreams.

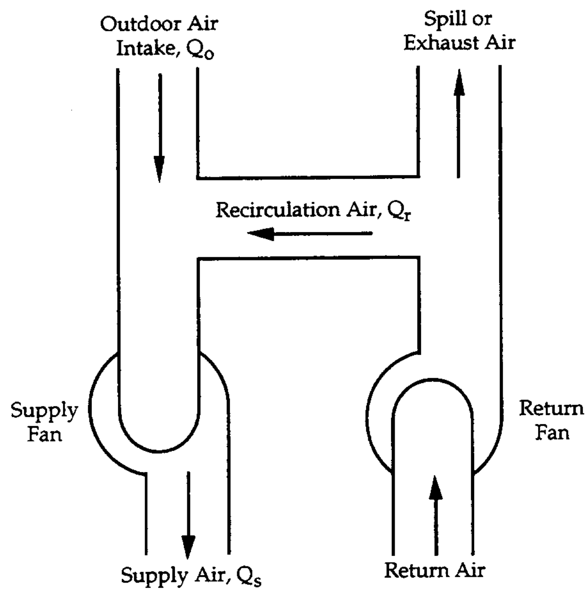

Figure 4.3 Air Handling System Schematic

Figure 4.3 shows the outdoor air intake, recirculation and supply airstreams in an air handling system. The percent outdoor air intake %OA is given by

$$\%OA = 100 \times \frac{Q_o}{Q_s}$$

As discussed below, both the temperature and tracer gas techniques require a significant difference between the indoor and outdoor air temperatures (tracer gas concentrations) relative to the uncertainty of the temperature (concentration) measurement to yield reliable results. The temperature approach is not useful under mild outdoor weather conditions.

4.7.1 Percent Outdoor Air - Temperature

Percent outdoor air intake can be determined from an energy balance of the recirculation, outdoor and supply airstreams. Based on this energy balance and a mass balance of the airflows, the percent outdoor air intake is given by

$$\%OA = 100 \times \frac{T_r - T_s}{T_r - T_o}$$

where T_r, T_S and T_O are the recirculation, supply and outdoor air temperatures. T_r can be measured in the return air duct which is often more accessible than the recirculation duct. The return air temperature should be measured in the main return duct of the air handler, not at a return vent in the occupied space or in a ceiling return air plenum. The supply air temperature should be measured at the air handler, as far downstream as possible from the point where the outdoor and return airstream mix. It should not be measured at a supply air outlet in the space.

The temperature approach requires that there is no energy input or removal between the three temperature measurement points. Therefore, the supply air temperature must be measured upstream of heating and cooling coils and the supply fan, and the return temperature should be measured downstream of the return fan.

To determine the percent outdoor air with a temperature balance, the recirculation, outdoor and supply air temperatures are measured as described in Section 4.1. All of the temperatures should be measured over a one-half hour period to avoid temperatures changes over time. The percent outdoor air is then calculated using the equation above.

The uncertainty in %OA calculated by a temperature balance depends on the uncertainties in the measured temperatures and on the differences between T_r and T_O and between T_r and T_S. The uncertainty in %OA, referred to as Δ%, is estimated with the following equation

$$\Delta\% = \%OA \left[\frac{\left(\Delta T_r^2 + \Delta T_o^2\right)}{(T_r - T_o)^2} + \frac{\left(\Delta T_r^2 + \Delta T_s^2\right)}{(T_r - T_s)^2} \right]^{1/2}$$

where ΔT_r, ΔT_O and ΔT_S are the uncertainties in the recirculation, outdoor and supply air temperatures. The uncertainty in %OA calculated with this equation accounts for only the precision of the measured temperatures and neglects bias due to faulty calibration and operator error.

As seen in the following example, the uncertainty in %OA is larger when the outdoor air temperature T_O is closer to the return air temperature T_r. When T_O = 20 °C, the uncertainty in %OA is very large. This example shows that the temperature balance technique should not be used when the outdoor air temperature is close to the indoor air temperature.

Example: Uncertainty in %OA Using Temperature Balance

In this example, T_r = 25 °C and the uncertainty in the measured temperatures equals 1 °C. For an outdoor air temperature of -5 °C, the supply temperature T_S equals 19 °C, 10 °C and 1 °C for the cases of 20%, 50% and 80% outdoor air intake. For T_O = 10 °C, the supply temperature T_S equals 22 °C (20%), 17.5 °C (50%) and 13 °C (80%). And for T_O = 20 °C, the supply temperature equals 24 °C (20%), 22.5 °C (50%) and 21 °C (80%). The table gives the uncertainty in %OA in units of %OA. Therefore, for the first case, the percent outdoor air intake equals 20 ± 5%.

	T_O = -5 °C	T_O = 10 °C	T_O = 20 °C
%OA = 20%	5%	10%	29%
%OA = 50%	5%	11%	32%
%OA = 80%	6%	12%	36%

4.7.2 Percent Outdoor Air - Tracer Gas

Percent outdoor air intake can be determined from a tracer gas mass balance in the recirculation, outdoor and supply airstreams. From this mass balance and a mass balance of the airflows, the percent outdoor air intake is given by

$$\%OA = 100 \times \frac{C_r - C_s}{C_r - C_o}$$

where C_r, C_S and C_O are the tracer gas concentrations in the recirculation, supply and outdoor airstreams. C_r can be measured in the return air duct, which is usually easier to access than the recirculation duct. The return concentration should be measured in the main return duct of the air handler, not at a return vent in the occupied space or in a ceiling return air plenum. The supply concentration should be measured at the air handler, downstream of the fan to enhance mixing of the outdoor and return airstreams. C_S should not be measured at a supply air outlet in the space.

This technique can be used with any tracer gas with a constant outdoor concentration. Sometimes occupant-generated carbon dioxide (CO_2) is used to measure percent outdoor air. In other cases a tracer gas, such as sulfur hexafluoride (SF_6), is injected into the building. When using CO_2, the measurement should be made when the indoor concentration is well above the outdoor concentration, i.e., between late morning and late afternoon.

The tracer gas approach is based on several requirements. First, there must be no tracer gas released into or removed from the system between the three concentration measurement points during the concentration measurements. Also, the tracer gas concentration of each airstream must be an average value across the airstream. This is more of an issue for the supply airstream where there can be stratification from imperfect mixing of the recirculation and outdoor air.

When injecting a tracer gas into the building, it should be released into the return or supply airstream so that the return concentration is raised significantly above the outdoor concentration. The volume of tracer gas released depends on the building volume and the concentration measurement range of the tracer gas monitor. If the building volume is V and the full-scale concentration of the tracer gas monitor is C, then the tracer gas volume released should be around 75% of V_t, where V_t is given by

$$V_t = V \times C$$

V_t and V are in the same volume units. If C is in units of parts per million (ppm), then the right side of the equation must be multiplied by 10^6. The value of C should be significantly above the outdoor concentration to reduce the uncertainty in the value of the percent outdoor air intake.

The amount of tracer gas released into the building does not have to be measured precisely, but it should be controlled. The tracer can be released all at once from a syringe or other container of known volume, or it can be injected at a constant rate using a flow meter and a compressed gas supply. There must be no additional tracer gas released in the building after the injection and during the concentration measurements.

After the tracer gas concentration in the building has stabilized, usually about one-half hour after injection, the tracer gas concentrations are measured in the recirculation, outdoor and supply airstreams as described in Section 4.6. These concentrations can be measured directly with a portable monitor or air samples can be collected for analysis with a stationary monitor.

The measurement procedure is as follows:

Measure the concentration, or collect an air sample, in the return duct.

Measure the concentration, or collect an air sample, in the supply duct.

Measure the concentration, or collect an air sample, in the outdoor air.

Repeat the concentration measurement in the supply duct, or collect another air sample.

Repeat the measurement of the concentration in the return duct, or collect another air sample.

This procedure should be completed over about one-half hour. If air samples are collected in containers, the tracer gas concentrations should be measured as soon as possible. The two return air concentrations are averaged and the average concentration used in the percent outdoor air calculation. Similarly, the two supply air concentrations are averaged for the calculations. The percent outdoor air intake %OA is calculated using the equation given earlier.

The uncertainty in %OA depends on the uncertainty of the measured concentrations and on the differences between C_r, C_o and C_s. The following equation should be used to estimate the uncertainty in %OA

$$\Delta\% = \%OA\left[\frac{\left(\Delta C_r^{\,2} + \Delta C_o^{\,2}\right)}{(C_r - C_o)^2} + \frac{\left(\Delta C_r^{\,2} + \Delta C_s^{\,2}\right)}{(C_r - C_s)^2}\right]^{1/2}$$

where ΔC_r, ΔC_o and ΔC_s are the uncertainties in the recirculation, outdoor air and supply air concentrations. The uncertainty in %OA calculated with this equation accounts for only the precision of the measured concentrations and neglects bias due to faulty calibration and operator error.

The following example shows that the uncertainty in the measured tracer gas (CO_2) concentration has a large impact on the uncertainty in the percent outdoor air intake. The uncertainty in %OA is also affected by the value of the percent outdoor air intake because the value of %OA determines the size of the differences between C_r and C_o and between C_r and C_s. Lower values of %OA lead to larger concentration differences and smaller uncertainties in %OA. The impact of these concentration differences is particularly important when using occupant-generated carbon dioxide as a tracer, because indoor CO_2 concentrations can not be controlled. Making these measurements after the CO_2 concentration has built up well above the outdoor level will increase the concentration differences and decrease the uncertainties in %OA, but this delay limits the times of day at which CO_2 can be used to measure %OA. When injecting another tracer gas into the building, the uncertainties in %OA can be reduced by making the difference between C_r and C_o as large as possible.

Example: Uncertainty in %OA Using Tracer Balance

In this example, C_o = 350 ppm. The return concentration equals 900 ppm, 570 ppm and 488 ppm for the cases of 20%, 50% and 80% outdoor air intake. The supply concentrations are 790 ppm (%OA = 20%), 460 ppm (50%) and 378 ppm (80%). The table contains the uncertainty in %OA for three values of the uncertainty in the measured CO_2 concentration ΔC. The uncertainty in %OA is given in units of %OA. Therefore, for the first case, the percent outdoor air intake equals 20 ± 3%.

	ΔC =10 ppm	ΔC =25 ppm	ΔC =50 ppm
%OA = 20%	3%	7%	13%
%OA = 50%	7%	18%	36%
%OA = 80%	13%	33%	66%

4.8 Airflow Rate

This section describes techniques to measure airflow rates in ventilation system ducts and at air outlets and inlets using pitot tubes, hot-wire and vane anemometers, and flow hoods.

4.8.1 Pitot Traverses in Ducts

Pitot tube traverses are commonly used to measure airflow rates in ducts in the testing and balancing, and the technique is described in several references (AMCA 1990, ASHRAE 1988, Bevirt 1984, NEBB 1986). In this technique, a pitot tube (Section 3.5) and a differential pressure gauge (Section 3.3) are used to measure the velocity pressure at several points in a duct cross section referred to as a *traverse plane*. The average air speed in the duct is calculated from the velocity pressure readings and is then multiplied by the cross-sectional area of the duct to determine the airflow rate.

The accuracy of pitot traverse measurements depends on the uniformity of the air speed in the duct and on the accuracy of the pressure gauge used to measure velocity pressure. Most discussions of pitot traverse measurements are based on the use of inclined manometers, and therefore they limit pitot traverses to air speeds above 3 m/s (600 fpm). However, the use of digital electronic manometers allows the measurement of lower air speeds. The accuracy of pitot tube traverse measurements is estimated to be from 5 to 10% under good field conditions (AMCA 1990, ASHRAE 1988). When the traverse plane is not in a good location, errors can exceed 10%.

Some references recommend that the traverse plane be in a straight section of ductwork, 6 to 10 duct diameters downstream and several diameters upstream of any elbows, branches, transitions or other obstructions (NEBB 1986). A more specific requirement is that at least 75% of the measured velocity pressures are greater than 10% of the maximum velocity pressure in the traverse plane (AMCA 1990, ASHRAE 1988). Figure 4.4 shows several situations that demonstrate this criteria. Case A is an ideal velocity pressure distribution, with all of the measured velocity pressures greater than 10% of the maximum velocity pressure. The velocity profiles in cases B and C are less uniform than case A, but they are still acceptable. The traverse plane in Case D is unacceptable since only 60% of the velocity pressures are greater than 10% of the maximum. Cases E and F are also unsatisfactory.

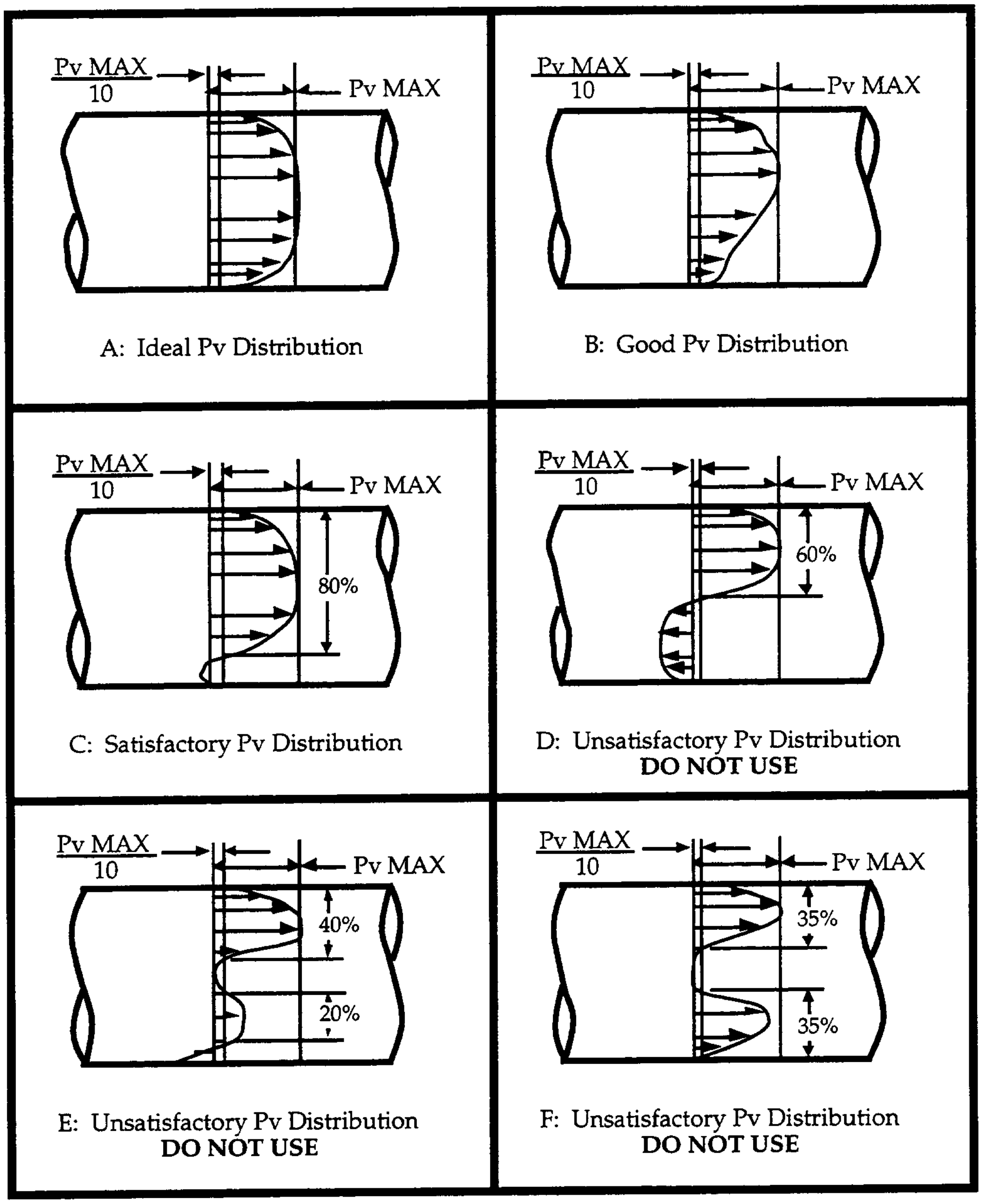

Figure 4.4 Velocity Pressure Distributions in Traverse Planes
(from ASHRAE Standard 111-1988)

The traverse plane should be located so that the air flows at a right angle to the traverse plane and the cross sectional shape of the duct is not irregular. It is often difficult, if not impossible, to find traverse planes in the field that meet these requirements. In these situations, one should use the best location available, avoiding elbows, offsets, transitions and obstructions. One can also increase the number of traverse points.

It is often difficult to find an acceptable traverse location at outdoor air intakes. In these situations, one can determine the outdoor air intake rate by measuring the supply airflow rate and multiplying it by the measured percent outdoor air intake rate. Section 4.8.6 describes the multiplicative method.

A pitot traverse measurement of airflow rate begins with the measurement of the inside dimensions of the duct. In a round duct, one measures the diameter. In a rectangular or oval duct, one measures the height and width. If the duct has interior insulation, these dimensions are measured inside the insulation. The duct areas are calculated using the following formula:

Rectangular duct:	Area = Length x Width
Round duct:	Area = $(\pi/4)\,(\text{Diameter})^2$
Oval duct:	Area = $\pi D_1^2 / 4 + D_1 (D_2 - D_1)$

D_1 is the length of the minor (short) axis of an oval duct and D_2 is the length of the major (long) axis.

Based on the shape and size of the duct, one determines the points at which the velocity pressure will be measured during the traverse. The traverse plane in rectangular ducts is divided into equal areas, with a traverse point in the center of each. There should be at least 20 traverse points, located no more than 0.15 m (6 in.) apart. Several references provide more details on the selection of traverse points (AMCA 1990, ASHRAE 1988, NEBB 1986). Figure 4.5 shows the traverse points for a 1 m by 1.5 m duct.

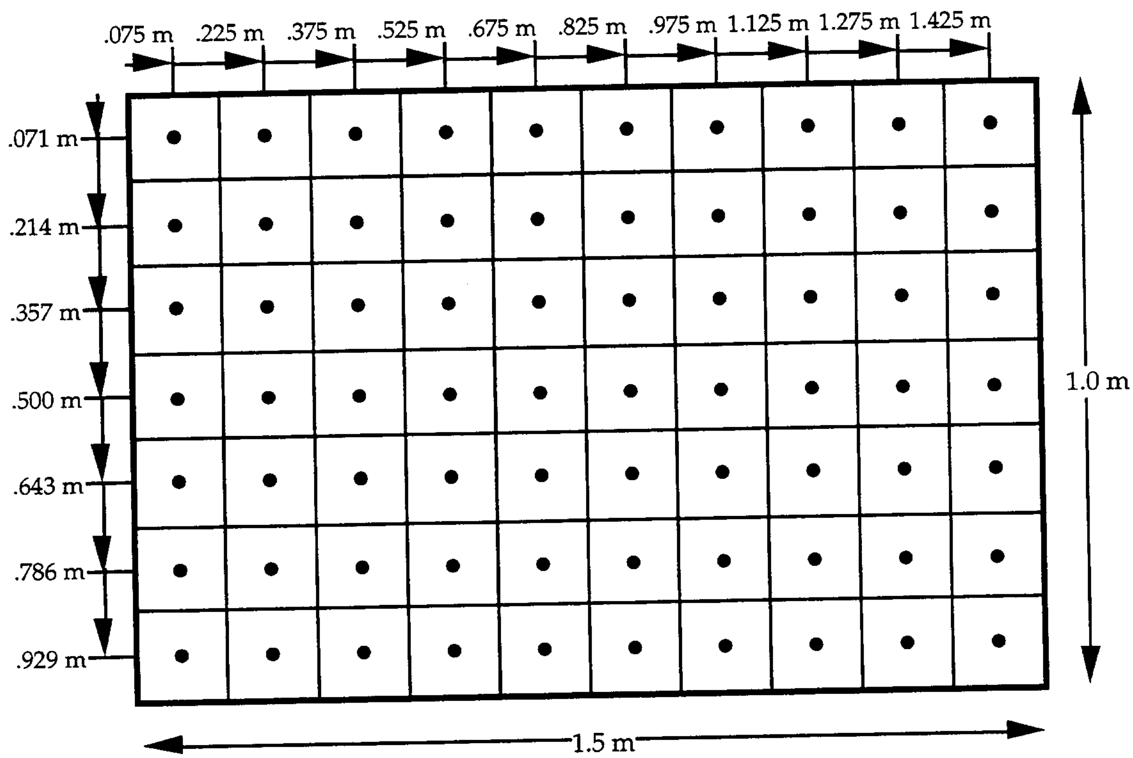

Figure 4.5 Rectangular Duct Traverse

For a rectangular duct with dimensions of 36 in. by 48 in., the traverse points would be located at distances of 3, 9, 15, 21, 27, 33, 39 and 45 inches along the 48 in. length and at distances of 3, 9, 15, 21, 27 and 33 inches along the 36 in. length.

In round ducts, the traverse points are located on two duct diameters as shown in Figure 4.6. The locations of the traverse points are based on the duct diameter D.

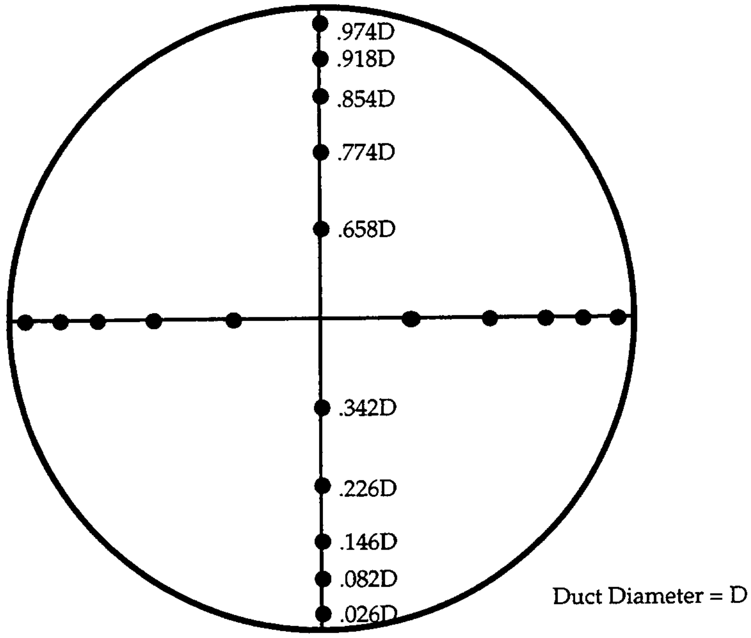

Figure 4.6 Round Duct Traverse

In oval ducts, the traverse points are located on the major and minor axes of the duct as shown in Figure 4.7. The locations of the traverse points are based on lengths of the minor axis D_1 and the major axis D_2.

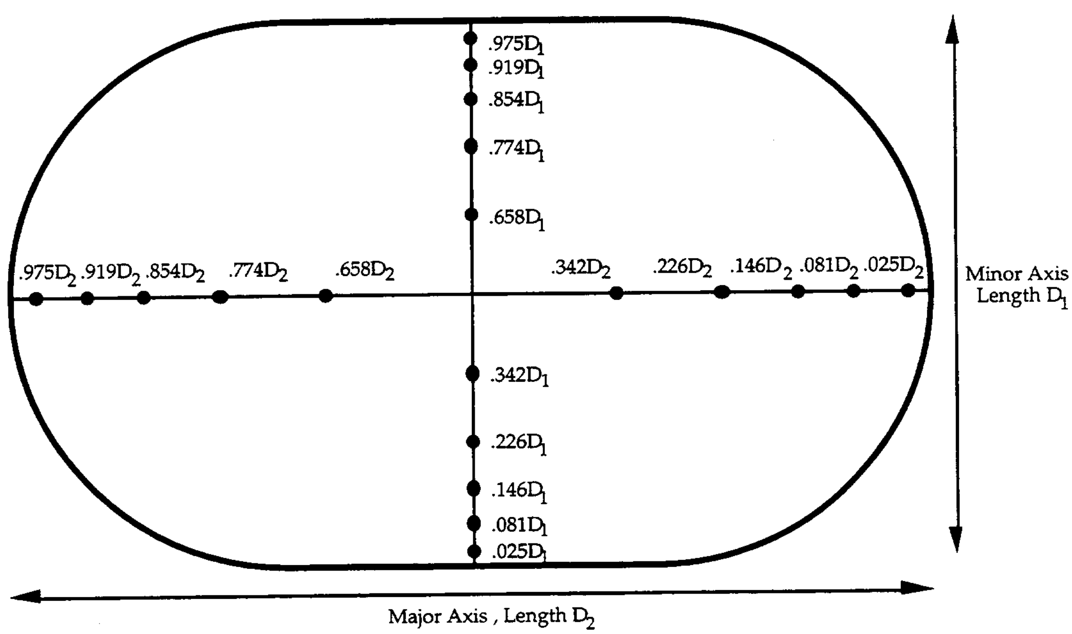

Figure 4.7 Oval Duct Traverse

During the pitot traverse, the velocity pressure at each traverse point is measured with a pitot tube and differential pressure gauge. Forms C1 through C3 are provided in Appendix C for recording the traverse data for rectangular, round and oval ducts. At each measurement point, the tip of the pitot tube must be pointed directly into the airstream and held in position for a period of time that is based on the response time of the gauge. The velocity pressure will usually fluctuate due to turbulence. Therefore, the velocity pressure should be read once every 10 seconds to obtain six readings, and the average of the six velocity pressures should be calculated. Forms C1 through C3 contain six lines for recording the six pressure readings. These forms also have lines for recording the air temperature and barometric pressure for the air density calculations.

The results of the traverse are the average velocity pressure in Pa (in. w.g.) at each traverse point. To convert these pressures to airflow rates, the root mean square velocity pressure $P_{v,rms}$ is calculated from the individual velocity pressure readings $P_{v,i}$ according to:

$$P_{v,rms} = [\ \Sigma(P_{v,i})^{0.5} \ / \ \text{Number of readings}]^2$$

The root mean square velocity pressure $P_{v,rms}$ is converted to the average air speed in the duct $\bar{V}$ according to:

$$\text{SI units} \qquad \bar{V} = \sqrt{2\,P_{v,rms} / \rho}$$

$\bar{V}$ is in m/s, $P_{v,rms}$ is in Pa, and the air density ρ is in kg/m^3.

$$\text{Inch-pound units} \qquad \bar{V} = 1096\sqrt{P_{v,rms} / \rho}$$

$\bar{V}$ is in fpm, $P_{v,rms}$ is in in. w.g., and ρ is in lb/ft^3. The air density is determined from air temperature and barometric pressure as discussed in Section 4.5.

The average air speed in the duct is multiplied by the cross-sectional area of the duct to determine the volumetric airflow rate Q_{meas} in m^3/s (cfm).

4.8.2 Hot-Wire Traverses in Ducts

The use of hot wire anemometers to measure airflow rates in ducts is similar to the use of pitot tubes, except the air speed is measured directly at each traverse point. The location of traverse planes is discussed in Section 4.8.1 for pitot tubes, and this discussion also applies to hot wires. Section 4.8.1 also discusses the measurement of the duct area and the location of the measurement points in the traverse plane. The uncertainties of airflow rates measured by hot wire traverses are not discussed in HVAC and TAB industry standards and guidance documents as they are for pitot tubes. Hot wire traverses are omitted, in part, because reliable hot wire anemometers have only recently been available at a reasonable cost and because these documents have not been updated for several years. However, if one assumes that air speeds measured with hot wire anemometers are as accurate as those obtained with pitot tubes, then the uncertainty of the airflow rate should be the same as that quoted for a pitot traverse, i.e., from 5 to 10% under good field conditions (AMCA 1990, ASHRAE 1988). When the traverse plane is not in a good location, the errors can exceed 10%.

During the hot-wire traverse, the air speed at each point is measured with a hot wire anemometer. Forms C4 through C6 in Appendix C can be used to record traverse data for rectangular, round and oval ducts. The hot wire probe must be positioned at each measurement point as directed in the instructions for the device and for a period of time that is based on the response time of the hot wire. The air speed reading will usually fluctuate due to turbulence. Therefore, the air speed should be read once every 10 seconds to obtain six readings, and the average of the six readings should be calculated. Forms C4 through C6 contain six lines for recording the six air speed readings. The forms also have lines for recording the air temperature and barometric pressure for the air density calculations.

The results of the traverse are the average air speeds in m/s (fpm) at each point in the traverse plane. The average air speed in the duct is calculated from the measured air speeds and multiplied by the cross-sectional area of the duct to determine the volumetric airflow rate Q_{meas} in m^3/s (cfm).

4.8.3 Vane Anemometer Traverses at System Coils

While pitot tube and hot wire traverses are the preferred techniques for measuring airflow rates in ducts, vane anemometer traverses are sometimes performed at system coils when no suitable traverse locations exist in a supply air duct. While industry guidance documents and standards discourage the use of vane anemometer traverses (NEBB 1986, ASHRAE (1988), they point out that coil traverses are sometimes the only option for measuring supply airflow rates. Rotating vane anemometers are described in Section 3.7.

Industry documents do not describe the use of vane anemometers to measure airflow rates at system coils. The following technique for traversing coils is based on reports of field experience and laboratory testing (Howell et al 1984, 1986 and 1989, Sauer and Howell 1990, Suppo 1984). In this technique, the air speed is measured at multiple points across the face of a coil bank. There should be at least twenty measurement points, located in the centers of equal-area sections of the coil face. During the measurements, the anemometer should be held directly against the downstream coil face. The airflow rate is equal to the average of the measured air speeds multiplied by both the cross-sectional area of the coil face and a so-called K-factor. The K-factor accounts for airflow obstructions caused by the coil tubes and fins. Research has shown that K depends on the air speed, the number of rows of coils, the spacing of the fins and tubes, and the tube diameter (Sauer and Howell 1990). For the purposes of this manual, a value of 0.75 can be used for K, resulting in a measurement uncertainty in the airflow rate of 20% for uniform airflows. When the airflow is very nonuniform, the uncertainty can be 30% or more.

4.8.4 Flow Hoods at Outlets and Inlets

Accurate airflow rate measurements at air outlets and inlets are difficult due to irregular airflow patterns and because the measurement device can affect the airflow rate. Among the instruments available for measuring these airflow rates, flow hoods require less training and experience than the alternatives. However, flow hood measurements can be subject to large errors in the field (ASHRAE 1988). Flow hoods are described in Section 3.8.

To measure the airflow rate, the hood is held against a supply outlet or return/exhaust inlet, and the airflow rate is read directly from the display. It is important that the vent is completely covered by the hood and that the hood is positioned securely against the surface containing the vent. If the hood has multiple measurement ranges, then the user should select a range so that the readings are near the middle of the range. After a large number of measurements with a flow hood, the user may experience fatigue, making it more difficult to position the device properly.

4.8.5 Vane Anemometers at Outlets and Inlets

Airflow rates at supply outlets and at return and exhaust inlets can also be measured with rotating vane, swinging vane and hot-wire anemometers. The use swinging vane and hot-wire anemometers requires a great deal of care and experience, and measurement errors in the field have not been well characterized (ASHRAE 1988). Rotating vane anemometers have been shown to produce reliable results if used properly. Therefore, this section discusses only vane anemometer measurements.

Vane anemometers can be used to measure airflows only at grilles and openings, and not at high velocity, induction diffusers. The opening is divided into equal area squares, approximately 10 cm (4 in.) on a side. An air speed reading is taken in the center of each square, and the measured air speeds are averaged. The airflow rate is determined by multiplying the average air speed by both the "designated" opening area and by an application factor provided by the manufacturer of the opening or grille. For exhausts, the designated opening area equals the area of the grille, uncorrected for the percent of open area. For supplies, the designated opening area equals the average of the uncorrected grille area and the free-open area. If no application factor is available from the manufacturer, ASHRAE Standard 111 (1988) recommends an application factor of 1.03 for supplies and 0.85 for returns and exhausts for air speeds between 2 to 7.5 m/s (400 to 1500 fpm). These application factors are limited to rectangular exhaust openings more than 10 cm (4 in.) wide, with an area up to 3700 cm^2 (600 in^2), and with at least 60% free open area. The same restrictions apply to supply openings, except that the free open area must be at least 70% and there must be no directional vanes. Application factors for lower air speeds and larger openings are discussed in an appendix to ASHRAE Standard 111 (1988).

The accuracy of airflow rate measurements at openings and grilles with rotating vane anemometers is 5% under good conditions, i.e., with a long, straight length of duct upstream or downstream of the opening. When there is no such duct, system effects can lead to measurement errors as large as 15 to 30% (ASHRAE 1988).

4.8.6 Multiplicative Method of Determining Outdoor Airflow Rates

Outdoor airflow rates can be measured in air distribution systems with the so-called multiplicative method. In this technique, the supply airflow rate Q_S is measured with a pitot tube, a flow hood or a hot wire or vane anemometer traverse, and the percent outdoor air intake %OA is measured as described in Section 4.7. The supply airflow rate is then multiplied by the percent outdoor air intake to determine the outdoor air intake rate Q_O according to:

$$Q_O = \%OA \times Q_S.$$

Q_S can be measured at an air handler, supply air duct or supply air outlet to determine the outdoor airflow rate at that point in the air distribution system. In most cases, the percent outdoor air intake %OA can be measured at the air handler and used at any point in the air distribution system. In induction systems, the value of %OA must be adjusted as discussed later in this section. This technique is more complex in dual-duct systems, and its use in these systems is not discussed in this manual.

The uncertainty in Q_O is estimated from the uncertainty in %OA (referred to in Section 4.7 as $\Delta\%$) and the uncertainty in the measurement of Q_S, referred to as ΔQ_S. When the supply airflow rate is measured with a pitot tube or hot-wire traverse, then ΔQ_S is equal to Q_S divided by 10. When Q_S is measured with a vane anemometer traverse or flow hood, then ΔQ_S is equal to Q_S divided by 5. The uncertainty in Q_O, referred to as ΔQ_O, is given by:

$$\Delta Q_O = \sqrt{(Q_S \times \Delta\%)^2 + (\Delta Q_S \times \%OA)^2}$$

The following example shows the uncertainties in Q_O determined with the multiplicative method. This example is based on the percent outdoor air intake determined with temperature (Section 4.8.1) and with tracer gas (Section 4.8.2) and uses the values from the examples in these two previous sections.

Example: Uncertainty in Outdoor Airflow Rate Determined with Multiplicative Method

In this example the measured supply airflow rate at an air handler is 50 m^3/s with an uncertainty of 5 m^3/s. The uncertainty in the outdoor airflow rate is presented first for the percent outdoor air determined from a temperature balance using the values in the example in Section 4.8.1. The uncertainty in the outdoor airflow rate is presented for three values of the outdoor air temperature.

	T_o = -5 °C	T_o = 10 °C	T_o = 20 °C
%OA = 20%, Q_o = 10 m^3/s	2.6 m^3/s	4.9 m^3/s	14.5 m^3/s
%OA = 50%, Q_o = 25 m^3/s	3.6 m^3/s	5.8 m^3/s	16.0 m^3/s
%OA = 80%, Q_o = 40 m^3/s	5.0 m^3/s	7.2 m^3/s	18.5 m^3/s

The uncertainty in the outdoor airflow rate is presented below for the percent outdoor air determined from a tracer gas balance using the values in the example in Section 4.8.2. The uncertainty in the outdoor airflow rate is presented for three values of the CO_2 concentration measurement uncertainty.

	ΔC = 10 ppm	ΔC = 25 ppm	ΔC = 50 ppm
%OA = 20%, Q_o = 10 m^3/s	1.6 m^3/s	3.4 m^3/s	6.6 m^3/s
%OA = 50%, Q_o = 25 m^3/s	4.4 m^3/s	9.3 m^3/s	18.1 m^3/s
%OA = 80%, Q_o = 40 m^3/s	7.7 m^3/s	16.9 m^3/s	33.0 m^3/s

In this example, when temperature is used to determine percent outdoor air intake, the uncertainty in the outdoor airflow rate is about 25% or less for an outdoor air temperature of -5 °C. When T_o is 20 °C, the uncertainty in the outdoor airflow rate ranges from about 50% to 150%. The measurement uncertainty is also greater for larger values of percent outdoor air than for lower values. When tracer gas (CO_2) is used to determine percent outdoor air intake, the uncertainty in the outdoor airflow rate is primarily a function of the uncertainty in the CO_2 concentration measurement. For a concentration measurement uncertainty of 10 ppm, the uncertainty in the outdoor airflow rate is about 20% of its value. For an uncertainty of 25 ppm, the uncertainty in the outdoor airflow rate is around 35% of its value. At 50 ppm uncertainty, the uncertainty in the outdoor airflow rate is close to 75%.

To determine the outdoor airflow rate downstream of an induction type terminal unit, the percent outdoor air intake at the air handler %OA must be corrected to account for the induced return or room air. To make this correction, the fraction of the terminal unit supply air that is primary air, referred to as %PA, must be measured. The design and accessibility of the terminal unit affects the ability to measure %PA. Figure 4.8 is a schematic of an induction unit, showing the quantities used to determine %PA. The primary airflow rate is referred to as Q_p, the airflow rate of induced air is Q_i and the supply airflow rate out of the terminal unit is Q_{tu}. %PA is equal to Q_p divided by Q_{tu}.

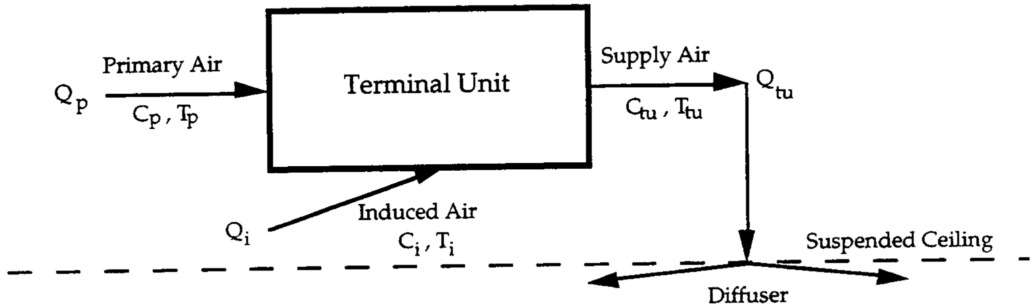

Figure 4.8 Schematic of an Induction Unit

To determine %PA, the tracer gas concentration (or temperature) of the primary air C_p (T_p), induced air C_i (T_i), and supply air leaving the terminal unit C_{tu} (T_{tu}) are measured. %PA is calculated with the following equation:

$$\%PA = \frac{C_i - C_{tu}}{C_i - C_p}$$

This equation uses tracer gas concentrations. If temperatures are used to calculate %PA, then the concentrations are replaced by air temperatures. The percent outdoor air intake in the supply airstream leaving the terminal unit $\%OA_{TU}$ equals %PA multiplied by the percent outdoor air intake measured at the air handler %OA.

$$\%OA_{TU} = \%OA \times \%PA$$

The following equation gives the uncertainty in $\%OA_{TU}$, referred to as $\Delta\%_{TU}$, based on the uncertainty in %OA, referred to as $\Delta\%$, and the uncertainty in %PA, referred to as $\Delta\%_{PA}$. The uncertainty in %PA is determined from the equation given in Section 4.7 to calculate the uncertainty in %OA.

$$\Delta\%_{TU} = \sqrt{(\%OA \times \Delta\%_{PA})^2 + (\Delta\% \times \%PA)^2}$$

The supply airflow rate at some point downstream of the induction unit Q_s is multiplied by the percent outdoor air intake at the terminal unit $\%OA_{TU}$ to determine the outdoor airflow rate.

$$Q_o = \%OA \times Q_s.$$

Q_s can be measured, and Q_o can be determined, at the terminal unit outlet or at a supply air diffuser.

The uncertainty in the outdoor airflow rate downstream of an induction unit is estimated from the measurement error in $\%OA_{TU}$ and the uncertainty in Q_s, referred to as ΔQ_s. When the supply airflow rate is measured with a pitot tube or hot wire traverse, then ΔQ_s is equal to Q_s divided by 10. When Q_s is measured with a vane anemometer traverse or flow hood, then ΔQ_s is equal to Q_s divided by 5. The measurement error in Q_o, referred to as ΔQ_o, is determined with the following equation:

$$\Delta Q_o = \sqrt{(Q_s \times \Delta\%_{TU})^2 + (\Delta Q_s \times \%OA_{TU})^2}$$

4.9 Air Change Rate

The outdoor air change rate of a building is the rate of outdoor air entering the building in L/s (cfm) divided by the building volume and is usually expressed in air changes per hour (ach). The air change rate includes both outdoor air intake via the air handling system and infiltration through leaks in the building envelope. This section describes two techniques for determining building air change rates, tracer gas decay and equilibrium carbon dioxide analysis.

4.9.1 Tracer Gas Decay

In the tracer gas decay technique (ASTM 1993), a tracer gas is released to obtain a uniform tracer concentration throughout the building. The decay in tracer gas concentration is monitored over time, and the air change rate is determined from the rate of concentration decay. If the air change rate is constant, then the tracer gas concentration C(t) decays according to

$$C(t) = C_0\, e^{-It}$$

where t is time, C_0 is the tracer gas concentration at t = 0 and I is the air change rate. The air change rate I is equal to the outdoor airflow rate into the building Q (intake and infiltration) divided by the building volume V. Tracer gas concentrations measured in decay tests are usually analyzed by taking the natural logarithm of each side of the equation above.

$$\ln C(t) = \ln C_0 - It$$

where lnC(t) is the natural logarithm of the tracer gas concentration at time t. The value of I is determined with least squares linear regression to calculate the slope of this line for a series of concentration readings over time.

The average air change rate can also be determined over a time period, from t_1 to t_2, using the following equation

$$I = \frac{\ln C(t_1) - \ln C(t_2)}{t_2 - t_1}$$

where $C(t_1)$ is the tracer gas concentration at the beginning of the time period and $C(t_2)$ is the concentration at the end of the period.

The tracer gas decay technique requires that the tracer gas concentration in the building can be represented by a single value, i.e., the tracer gas concentration is uniform. This assumption will be valid if the tracer is carefully released into the building and if the building airflow patterns keep the concentration uniform. In mechanically ventilated buildings, the tracer gas can be injected into the building air handlers so that the air distribution system distributes the tracer. Recirculation of building return air helps achieve a uniform concentration. These equations also require that the outdoor tracer gas concentration is zero. If the outdoor concentration is not zero but constant, the technique can still be used with slightly modified equations.

4.9.2 Equilibrium Carbon Dioxide Analysis

Air change rates can be estimated from peak carbon dioxide concentrations, but the applicability of this technique is limited (Persily and Dols 1990). If the CO_2 generation rate inside a building and the outdoor airflow rate into the building are both constant, then the indoor concentration builds up to an equilibrium concentration C_{eq}. In practice, a constant CO_2 generation rate means a constant number of building occupants. If the outdoor CO_2 concentration C_{out} is constant and the average CO_2 generation rate per person G_p is known, then the outdoor airflow rate into the building is given by the following equation

$$Q = \frac{\text{Number of building occupants} \times G_p}{\left(C_{eq} - C_{out}\right)}$$

The value of G_p depends on a person's age and activity level. A typical value of G_p for office buildings is 5.3×10^{-6} m^3/s (0.011 cfm) per person. The value of Q is divided by the building volume V to determine the air change rate.

Example: Air Change Rate Calculated from Equilibrium Carbon Dioxide Analysis

A building has 100 occupants. The measured equilibrium CO_2 concentration is 850 ppm, and the outdoor concentration is 400 ppm. Using 5.3×10^{-6} m^3/s (0.011 cfm) as the CO_2 generation rate per person, the outdoor airflow rate into the building is 1180 L/s (2500 cfm). Based on a building volume of 10,000 m^3 (353,000 ft^3), the air change rate is 0.42 ach.

The indoor CO_2 concentration must be at equilibrium to obtain reliable results with this technique, and the time required to achieve equilibrium depends on the building air change rate. Figure 4.9 is a graph of CO_2 build-up over time for different values of the outdoor air change rate. At lower air change rates, it takes longer to reach equilibrium. To reach equilibrium, the CO_2 generation rate (building occupancy) must be constant for the amount of time required for the concentration to reach equilibrium, which depends on the air change rate. In most office buildings, the CO_2 generation rate is constant for only a few hours a day. Unless the building air change rate is very high, at least 1 ach, the indoor concentration may never reach equilibrium. If the CO_2 concentration prior to equilibrium is used in the equation given above, the building air change rate will be overestimated.

This technique also requires that the CO_2 concentration is uniform throughout the building interior. This requirement means it is inappropriate to use this technique to estimate air change rates of individual rooms when the rooms are at different concentrations. This technique can only be used to estimate whole building air change rates.

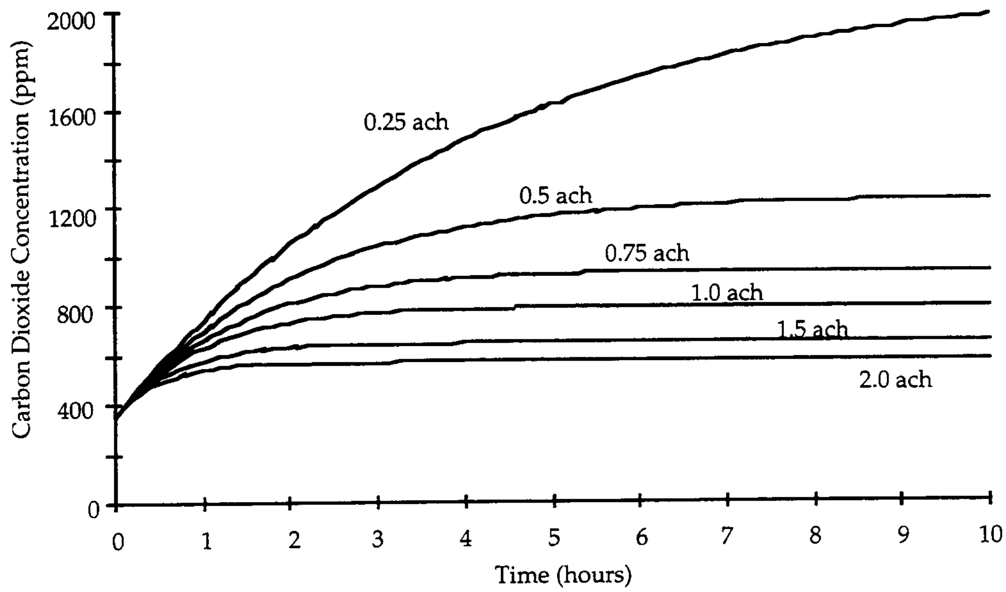

Figure 4.9 Carbon Dioxide Build-Up Over Time

Field studies have demonstrated the potential for large errors when using CO_2 concentrations measured before equilibrium to calculate air change rates (Dols and Persily 1992). This technique is not reliable for determining building air change rates unless the CO_2 concentration is at equilibrium and the other assumptions of the technique are valid.

4.10 References

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5 ASSESSMENT PROCEDURES

When evaluating the performance of a ventilation system, the building layout, the system design and the purpose of the evaluation must be considered in order to determine which system parameters are measured and the procedures used in these measurements. There is no set list of parameters and evaluation procedures that can be used in all buildings under all circumstances. This section describes several ventilation evaluation procedures and their application in the field.

5.1 Procedures

This section describes three types of ventilation evaluation procedures: space use analysis, system design evaluation, and performance measurement. Space-use analysis provides information on the ventilation requirements of a building. System design evaluation is used to determine how the ventilation system is intended to perform. Performance measurements provide information on how the system is actually operating using the techniques in Section 4.

5.1.1 Space-Use Analysis

This section describes procedures to determine the ventilation requirements of a building based on the building layout, the activities in the building and the occupancy levels. This information is important because the current occupancy levels and activities in a building can be very different from what was assumed when the ventilation system was originally designed. The space-use analysis uses information from the building plans, the building management and operating staff, and an on-site inspection.

The first step in a space-use analysis is to list the major spaces or zones in the building. This list is created by considering each building floor and identifying the major types of spaces on each floor. These space types include, but are not limited to, the following:

- **Offices**
- **Reception areas**
- **Telecommunication centers and data entry areas**
- **Duplicating and printing rooms**
- **Classrooms**
- **Auditoriums**
- **Cafeterias**
- **Conference rooms**
- **Lobbies**
- **Restrooms**
- **Smoking lounges**
- **Libraries**
- **Retail stores**
- **Kitchens**

The zones on each floor can be broken down further by considering additional factors that affect ventilation requirements and thermal loads. These additional factors include:

- **Occupant density**
- **Office equipment**
- **Lighting levels**
- **Exterior exposure (perimeter or interior; north, south, east or west)**

Table 5.1 is an example list of the major zones of a building, containing each zone's activity, number of occupants (if appropriate) and floor area. Restrooms are described by the number of water closets.

Building Floor	Activity	Number of Occupants	Floor Area	
4	Office space	250	5000 m^2	(54,000 ft^2)
	Conference room	20	50 m^2	(540 ft^2)
	Restrooms, 6 water closets		10 m^2	(110 ft^2)
3	Office space	500	5000 m^2	(54,000 ft^2)
	Smoking lounge	30	50 m^2	(540 ft^2)
	Restrooms, 6 water closets		10 m^2	(110 ft^2)
2	Office space	175	2500 m^2	(27,000 ft^2)
	Data entry area	600	2500 m^2	(27,000 ft^2)
	Conference room	20	50 m^2	(540 ft^2)
	Restrooms, 6 water closets		10 m^2	(110 ft^2)
1	Office space	210	3000 m^2	(32,000 ft^2)
	Reception area	60	2000 m^2	(22,000 ft^2)
	Duplicating center		50 m^2	(540 ft^2)
	Restrooms, 6 water closets		10 m^2	(110 ft^2)

Table 5.1 Example List of Zones for Space-Use Analysis

The next step in the analysis is to determine the ventilation requirements of the building zones based on the space type and the number of occupants. The ventilation requirements can be based on a ventilation standard, guideline or building code. ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality, is often used to determine ventilation requirements. Table 5.2 lists the minimum outdoor air requirements in Standard 62-1989 for a variety of space types. The table also contains default values for occupant density, which are useful when design or actual occupancy levels are unavailable.

	Occupancy	Outdoor Air Requirement			
Activity	people/100 m^2 or 1000 ft^2	L/s-person	cfm/person	L/s-m^2	cfm/ft^2
Office space	7	10	20		
Reception areas	60	8	15		
Telecommunication centers and data entry areas	60	10	20		
Conference rooms	50	10	20		
Smoking lounges	70	30	60		
Restrooms (exhaust flow per water closet or urinal)		25	50		
Duplicating and printing				2.50	0.50
Classrooms	50	8	15		
Laboratories	30	10	20		
Libraries	20	8	15		
Auditoriums	150	8	15		
Retail					
Basement and street level	30			1.50	0.30
Upper floors, malls, arcades	20			1.00	0.20

Table 5.2 Outdoor Air Requirements (ASHRAE Standard 62-1989)

Using outdoor air ventilation requirements, such as those contained in Table 5.2, the ventilation requirements for each zone are determined. Table 5.3 contains the ventilation requirements for the example zones in Table 5.1 based on ASHRAE Standard 62-1989.

Building Floor	Zone	Outdoor Air Requirement L/s (cfm)	
4	Office space	2500	(5000)
	Conference room	200	(400)
	Restrooms	150	(300)
3	Office space	5000	(10,000)
	Smoking lounge	900	(1800)
	Restrooms	300	(150)
2	Office space	1750	(3500)
	Data entry area	6000	(12,000)
	Conference room	200	(400)
	Restrooms	300	(150)
1	Office space	2100	(4200)
	Reception area	480	(900)
	Duplicating center	125	(250)
	Restrooms	300	(150)

Table 5.3 Outdoor Air Requirements for Example Zones

Some zones have additional requirements such as exhaust ventilation or a pressure relationship to adjoining spaces. For example, restrooms are often required to have mechanical exhaust that is not recirculated and to be at a lower pressure than the adjoining rooms.

The next step in the space-use analysis is to assess the thermal loads in the building. Thermal loads are important in evaluating the ability of the ventilation system to maintain acceptable thermal comfort in the occupied space. Interior zones in commercial buildings are usually dominated by cooling loads from people, lights and office equipment. Exterior zones have both heating and cooling loads that depend on climate, season, time of day, and exposure and construction of the building envelope. The analysis of thermal loads is beyond the scope of this manual. Thermal load calculations are described in the ASHRAE Handbook of Fundamentals (1993) and in the ASHRAE Cooling and Heating Load Calculation Manual (1992).

There is no simple approach for estimating thermal loads, but the following parameters should be considered for each zone:

- **Floor area, m^2 (ft^2)**
- **Number of occupants**
- **Number of personal computers**
- **Number of printers and photocopiers**
- **Lighting levels, W/m^2 (W/ft^2)**
- **Perimeter or interior zone**
- **Number of exterior walls, including exposure (north, south, east or west)**
- **Window treatments to reduce solar loads**

Typical cooling loads for interior zones of office buildings are about 55 W/m^2 (17.5 $Btu/hr\text{-}ft^2$). Assuming a supply air temperature of 13 °C (55 °F) and a room air temperature of 22 °C (72 °F), this cooling load requires 5 $L/s\text{-}m^2$ (1 cfm/ft^2) of supply air to remove the heat generated in the space. Spaces with a high density of occupants or office equipment and high lighting levels will have larger cooling loads. Computer rooms can have cooling loads ranging from 250 to 500 W/m^2 (80 to 160 $Btu/hr\text{-}ft^2$) and will require supply airflow rates from 23 to 46 $L/s\text{-}m^2$ (5 to 9 cfm/ft^2). In addition to the cooling loads in interior zones, heating and cooling loads in perimeter zones can also be significant. For zones with exterior walls, one needs to consider the solar exposure, the amount of window area, the type of glazing, and window treatments that decrease solar loads, e.g., overhangs, shades and films. While this information is far short of a detailed thermal load calculation, it can provide an indication of the expected range of thermal loads in the zone.

5.1.2 System Design Evaluation

This section describes how to determine how the ventilation system is intended to perform based on the design documentation. The results of a design evaluation can be compared with the ventilation requirements from the space-use analysis to determine if the system is adequate for the current use of the building. Such a comparison is important because building occupancy and activities can change without a corresponding change in the ventilation system. In some buildings the design documentation needed for a design evaluation is inadequate or missing. In such situations, the design evaluation should be completed to the extent possible.

5.1.2.1 Design Parameters

The objective of the system design evaluation is to determine how the ventilation system is intended to be operated and to obtain the values of important design parameters. An evaluation of the sequence of operations of the system is an important part of this procedure. The *sequence of operations* or *operating sequence* is a description of how the system is intended to operate under difference conditions and includes the following items: when the fans operate; how they are turned on and off; how the outdoor air intake and supply air temperatures are controlled; how the outdoor air, exhaust and return air dampers are controlled; and descriptions of different modes of system operation such as morning warm-up.

A list of the parameters that are considered in a design evaluation is presented below. This list includes both numerical values, such as airflow rates, and descriptive information, such as operating sequences.

- Design conditions
  Outside
    Winter: Outside dry bulb temperature
    Summer: Outside dry bulb and wet bulb temperatures
  Inside (may vary among the zones)
    Heating: Temperature, relative humidity
    Cooling: Temperature, relative humidity
  Thermal loads: Heating and cooling loads in the zones

- Supply fans
  System identification number and area served
  Description of system: type; location
  Filters: type; efficiency
  Operating sequence: controls; schedule; relationship to operation of other fans
  Supply airflow rate (In a variable air volume system this will be the system capacity, i.e., the maximum supply airflow rate.)
  Minimum outdoor air intake rate

- Return fans
  System identification number and area served
  Description of system: type; location
  Operating sequence: controls; schedule; relationship to operation of other fans
  Airflow rate (In a variable air volume system this will be the system capacity, i.e., the maximum return airflow rate.)

- Outdoor air fans
  System identification number and area served
  Description of system: type; location
  Filters: type; efficiency
  Operating sequence: controls; schedule; relationship to operation of other fans
  Airflow rate

- Air distribution system
  Duct branches: airflow rates in various supply ducts
  Terminal units
    System identification number and area served
    Description of system: type; location
    Operating sequence: controls; schedule; relationship to operation of other fans
    Supply airflow rates (In a variable air volume system this will be the supply air capacity, i.e., the maximum supply airflow rate.)
  Diffusers
    Description: type; location
    Supply airflow rates (In a variable air volume system this will be the supply air capacity, i.e., the maximum supply airflow rate.)

- Exhaust systems
  System identification number and area served
  Description of system: type; location
  Airflow rate
  Operating sequence: controls; schedule; relationship to operation of other fans

5.1.2.2 Sources of Design Information

The sources and amount of system design documentation varies among buildings. This section discusses some sources of design information.

<u>Construction or contract documents</u>: These documents, also called bid documents, are developed during the building design. They describe the HVAC system requirements, and the contractor uses them to prepare a bid on the building construction contract. Bid documents include building plans and the "M series" mechanical drawings. The mechanical drawings usually contain fan schedules and specifications, and may describe the sequence of operations for the controls.

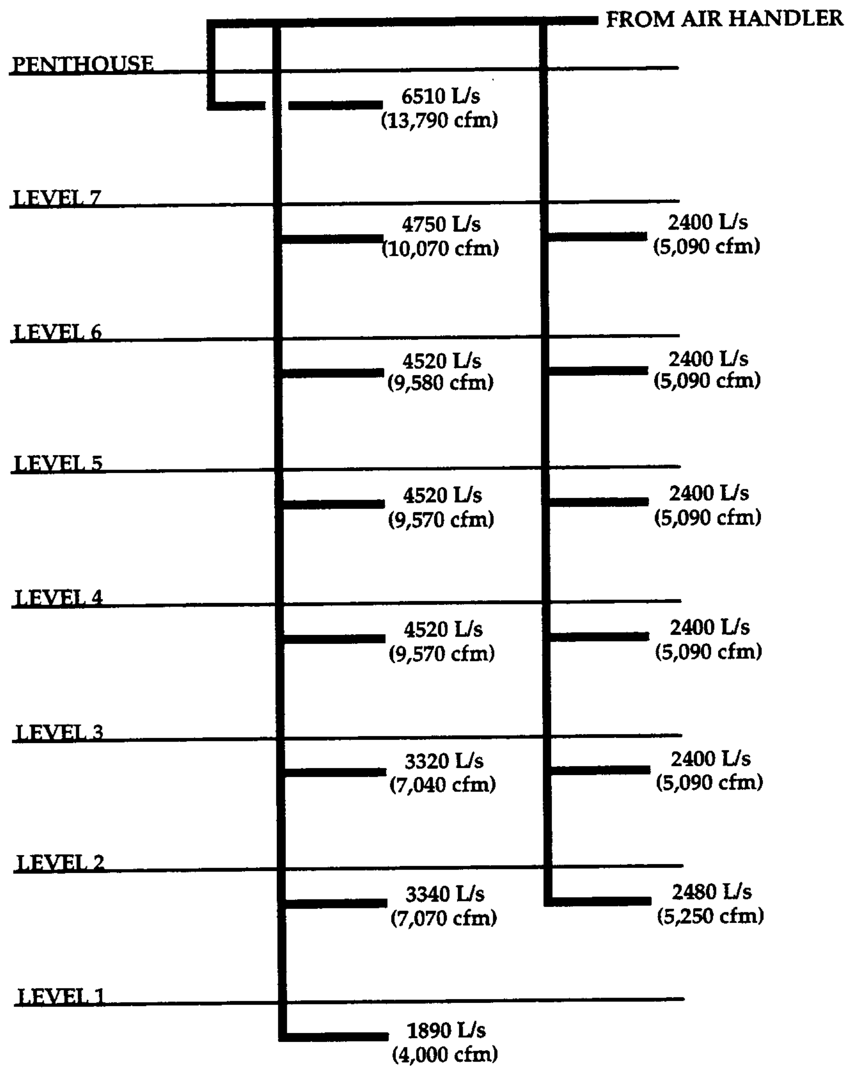

Figure 5.1 Example of Mechanical Drawing - Riser Diagram

Figure 5.1 is a riser diagram from a set of mechanical drawings, showing the design supply airflow rate to each building floor from an air handler. In constant volume systems, these are the airflow rates when the system is operating. In variable air volume systems, these are the supply airflow rate capacities that occur at the design thermal loads.

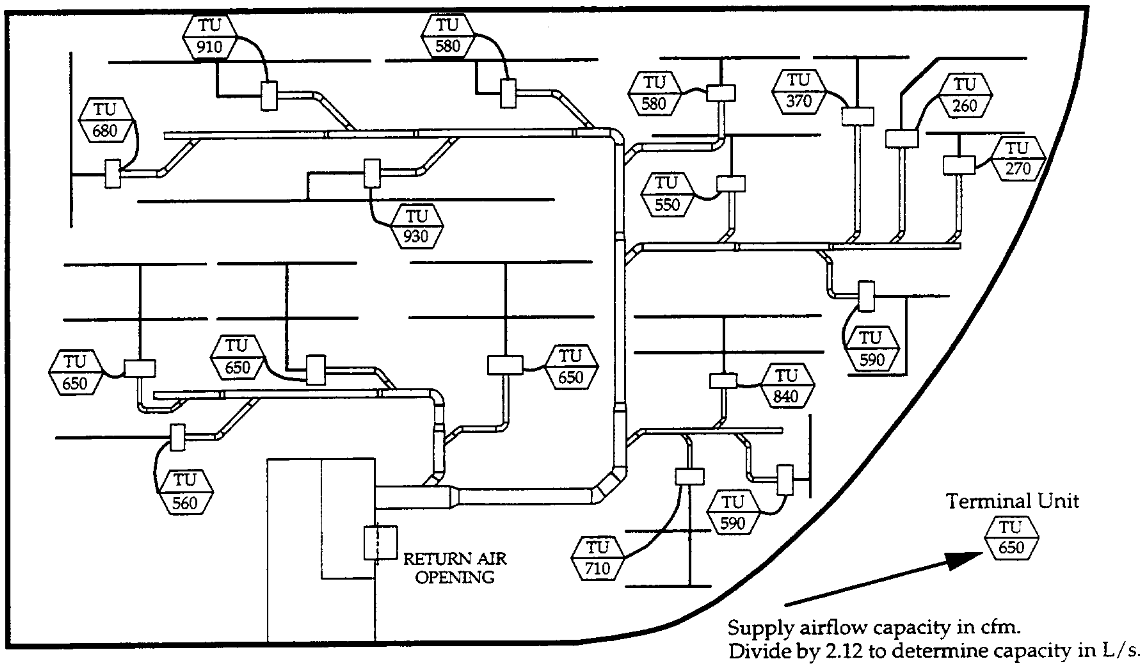

Figure 5.2 Example of Mechanical Drawing - Floor Plan

Figure 5.2 is a mechanical drawing showing the supply air ductwork and the terminal units on an individual floor. The supply airflow rate is specified for each terminal unit, in this case the design capacity in a VAV system. In addition to the information in this drawing, additional specifications exist for the airflow rate out of each supply air diffuser.

UNIT DESIGNATION		A-1
CONTROL SEQUENCE		A
OUTSIDE AIR CFM - MAXIMUM		11415
OUTSIDE AIR CFM - MINIMUM		11415
GRAND TOTAL HEAT (BTU/HR)		2405000
FAN	AIR CAPACITY (CFM)	55730
	MAXIMUM OUTLET VELOCITY (FPM)	3295
	INTERNAL STATIC PRESSURE (IN. W.G.)	3.24
	EXTERNAL STATIC PRESSURE (IN. W.G.)	3.95
	MOTOR HORSEPOWER / RPM	100/1,750
FACE FILTERS	AIR CAPACITY (CFM)	41800
	MAX FACE VELOCITY (FPM)	500
	AIR FRICTION ALLOWANCE (IN. W.G.)	1.4
	TYPE / SIZE	A/15'-0" X 7'-0"
	MINIMUM EFFICIENCY (%)	93
BYPASS FILTERS	AIR CAPACITY (CFM)	27865
	MAX FACE VELOCITY (FPM)	500
	AIR FRICTION ALLOWANCE (IN. W.G.)	1.4
	TYPE / SIZE	B/11'-0" X 6'-0"
	MINIMUM EFFICIENCY (%)	93

Figure 5.3 Example of Mechanical Drawing - Fan Schedule

Figure 5.3 is a fan schedule from a set of mechanical drawings. These are specifications for a single fan with a constant outdoor air intake rate, i.e., the minimum and maximum outdoor airflow rates are the same.

The following is a portion of an operating sequence from a set of bid documents.

Example: Sequence of operations

Mixed air temperature control: Provide an averaging bulb sensor and controller in the mixed air plenum set to maintain 14 °C (57 °F) by modulating the maximum outdoor air, exhaust and return dampers in unison. Interlock the maximum mixed air dampers to go to the 100% return air position when the outdoor air temperature raises to 27 °C (81 °F) or above and during warm-up operation. Arrange the dampers to go to the 100% return air position upon fan shutdown.

The designer wants the mixed air temperature controlled to 14 °C (57 °F) by modulating the outdoor air, exhaust and return dampers. When the outdoor air temperature exceeds 27 °C (81 °F), the dampers should be positioned so that all of the return air is recirculated. The dampers will also be positioned for 100% return air during morning warm-up and when the fan is off.

Controls submittals: These are shop drawings developed by the controls contractor to demonstrate their understanding of the system design. The controls submittals describe how the contractor plans to implement the design and may repeat or expand on the description of the sequence of operation. A portion of an operating sequence description follows.

Example: Sequence of operations

Cooling Supply Fans 1&2: The supply and return fans are normally started and stopped through an ORT program in the JC/85/40. In addition a warm-up and morning cooling down program may also start them. They may also be started via manual control from either the 85/40 or the DSC-8500. The mixed air control modulates the outdoor air, exhaust and return air dampers to maintain the mixed air set point. The dampers will go to 100% return when night low limit or warm-up are called for and when the fan is shut down. Anytime the OSA temperature exceeds the mixed air set point the dampers will be in a 100% return position. When the OSA exceeds the economizer lockout the dampers are closed.

This description discusses how the supply and return fans of a cooling deck will be controlled by the building automation system. When the outdoor air temperature exceeds the mixed-air setpoint temperature, the dampers will be positioned such that all of the return air is recirculated. When the fan is off, the dampers will be positioned for 100% return air. The control submittals are usually more specific than the bid documents.

Mechanical equipment submittals: These submittals are information supplied by equipment manufacturers that contain specifications of the equipment installed in the building, such as fans and chillers. Equipment submittals are also referred to as "cut sheets."

Operation and maintenance manuals: Operation and maintenance manuals exist in some buildings and describe how the building equipment is intended to be operated and to perform. The information in these manuals varies among buildings and may include information on the time of operation, outdoor air intake controls, indoor temperature and humidity controls, and relationships between the operation of different fan systems. An outline of one building's operation and maintenance manual is presented below.

Example: Operations and maintenance manual outline

- Design conditions
  - Outdoor temperature, winter and summer; summer design wet bulb temperature.
  - Inside temperature and relative humidity for individual areas within the building, with separate values for heating and cooling.
- Description of systems and design intent (one section for each system)
  - Air conditioning apparatus - Central systems
    - System number and area served.
    - Description of systems including air handler type (e.g., low velocity terminal reheat), outdoor air control (e.g., minimum when outdoor air temperature exceeds 21 °C (70 °F), economizer control intake below 21 °C (70 °F)), and reference to contract document drawing number containing control diagram and sequence of operations.
    - List of equipment manufacturers for each system component and reference to section of the operations and maintenance manual describing each item.
  - Exhaust fan systems
    - System number and area served.
    - Description of systems including associated make-up air fans.
    - List of equipment manufacturers for each system component and reference to section of the operations and maintenance manual describing each item.
  - Air distribution systems
    - Description of equipment that distributes air including diffusers, grills and light troffers, and description of volume dampers used to adjust airflow rates.
    - List of equipment manufacturers for each system component and reference to section of the operations and maintenance manual describing each item.
  - Interior and perimeter fan coil systems
    - System types and reference to relevant contract document drawing numbers.
    - Description of operation schedules and control of water valves.
    - List of equipment manufacturers for each system component and reference to section of the operations and maintenance manual describing each item.
  - Filters
    - Filter types described for each air handling unit including efficiency and appropriate guidance on replacement.

TAB report: The testing, adjusting and balancing, TAB or air balance report has information on the design and performance of air handling systems. Air balancing is performed late in the building construction process, but may also be performed at other times in the life of a building. Air balance reports should contain design supply airflow rates for fans, terminal units and diffusers. The minimum outdoor air intake rate of the fans and other design information related to outdoor air intake may also be included. The report also contains the airflow rates measured during the TAB process. If many months or years have passed since the air balancing, or if the air balance contractor did not use accepted industry practice, the measurements may not be useful in the current ventilation assessment.

The following example is an outline of some of the information that is found in air balance reports.

Example: Air balance report

Air balance contractor
Person(s) performing test
Date of test
Test equipment employed
Test data
Fans: Identification number, type, design values of airflow rate, test conditions, measured values of static pressure, total pressure, rpm and airflow rate
Terminal units: Identification number, type, location, design airflow rate, test instrument, measured airflow rate
Supply air outlets: Identification number, type, room, design airflow rate, test instrument, measured airflow rate

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"Walk-and-talk": The building engineers and mechanics who operate and maintain ventilation systems are valuable sources of design, operation and performance information. These people often know things about the system and the equipment that are never documented. The term "walk-and-talk" refers to the process of inspecting the system with the building operation staff. During the inspection, the design and actual operation of the system are discussed. To maximize the usefulness of the effort, the persons performing the ventilation evaluation should study all available documentation on the system before meeting with the building operators. It is important to remember that building operators are very busy people with many demands on their time, and the "walk-and-talk" session should be scheduled at the operators convenience.

5.1.3 Comparing Space-Use and Design

After the ventilation requirements of the building have been determined and the ventilation system design has been evaluated, the ventilation requirements and the system design are compared. As stated earlier, building space-use can change without changes in the ventilation system design or operation. The system may no longer be able to provide adequate outdoor air ventilation or good thermal comfort. This section describes how to compare the results of the space-use analysis and the system design evaluation.

5.1.3.1 Minimum Outdoor Air Intake

One of the most important aspects of the comparison between ventilation requirements and system design is the outdoor air intake rate. The outdoor air requirements of the zones are obtained from the space-use analysis described in Section 5.1.1. The requirements for all of the zones served by an air handler are added together, and the total is compared to the minimum outdoor air intake specification for the air handler. If the total outdoor air requirement for the zone is more than the design value of the minimum outdoor air intake for the air handler, then the system design does not meet the recommendations of the ventilation standard or guideline used to determine the requirement.

The ventilation requirement of an air handler is the sum of the requirements of several individual zones. In some cases the outdoor air requirements of these zones will be different. Even if the total outdoor air intake of the system is adequate, the ventilation requirements of some of the zones may not be met by the system. ASHRAE Standard 62-1989 describes a procedure for adjusting the outdoor air requirements of an air handler in these situations. Another approach is to divide the ventilation requirements for each individual space by the floor area of the space to determine the outdoor airflow rate in units of L/s-m^2 (cfm/ft^2). Similarly, the design outdoor air intake rate for the air handler is divided by the total floor area served by that air handler. If the design outdoor air intake of the air handler by floor area is less than the ventilation requirement of a zone in the same units, then the design outdoor airflow to that zone does not meet its requirements.

5.1.3.2 Supply Airflow Rate

The supply airflow rate of a system determines whether it will be able to handle the thermal loads in the space and provide acceptable thermal comfort. As discussed in Section 5.1.1, the determination of thermal loads is beyond the scope of this document. Instead, the thermal loads based on occupant densities, activities and envelope exposure are considered qualitatively. To evaluate the adequacy of the design supply airflow rate, convert the supply airflow rate from units of L/s (cfm) to units of L/s-m^2 of floor area (cfm/ft^2). This conversion should be done for each of the zones in the space-use analysis. If design supply airflow rates are not available for these zones, they can be determined by summing the design values for the terminal units or diffusers in the space. If terminal unit or diffuser design values are not available, then the total supply airflow rate for the air handler can be divided by the total floor area served by that air handler. The design supply airflow rates of a space should be evaluated based on the occupant density, the amount of office equipment and the exposure of any exterior walls. The evaluation of design supply airflow rates in relation to the thermal loads in the space requires experience with ventilation and thermal loads in commercial buildings. If the supply airflow rate capacity for a space is 2 L/s-m^2 (0.4 cfm/ft^2) and the space has an occupant density of 15 people per 100 m^2 (1000 ft^2) with a personal computer on every desk, then the supply airflow rate is probably too low to remove the heat generated in the space. The consideration of perimeter zones is more difficult, but one should still determine the supply airflow rate per m^2 (ft^2) of floor area and consider the ability to heat and cool the perimeter zones given the envelope construction and climate.

Evaluating the adequacy of the supply airflow rate to maintain thermal comfort must consider the supply air temperature. While most systems in commercial buildings have a supply air temperature of about 13 °C (55 °F), some ventilation systems use "low temperature" supply air closer to 7 °C (45 °F). In these systems, the supply airflow rate required to remove heat will be less than for a more typical supply air temperature.

5.1.3.3 Exhaust Airflow Rates

For spaces that contain an exhaust air system, the design exhaust airflow rate is compared to the exhaust air requirements for the space. Exhaust air design values are expressed as airflow rates in units of L/s (cfm) for individual rooms or for a system serving several rooms. Exhaust air ventilation requirements are airflow rates for individual rooms or zones, based on an airflow rate recommendation per room or per unit floor area. Exhaust airflow rate requirements are determined based on the space-use analysis described in Section 5.1.1. The requirements are compared to the design values for each room. If design values are not available for the individual rooms, the exhaust air requirements for the rooms served by an exhaust system should be added together and compared to the design value for the whole system. If the exhaust air requirement is significantly above the system design value, then the design does not meet the recommendations of the ventilation standard or guideline used to determine the requirement.

5.1.3.4 Pressure Relationships Between Spaces

Ventilation standards and guidelines sometimes recommend that a pressure difference be maintained between certain space types and adjacent spaces. Common sense is also useful for deciding when there should be a pressure difference. For example, toilets or smoking lounges should be at a negative pressure to prevent the movement of contaminants to adjoining rooms. The ventilation system design may document this requirement. In some cases, e.g. clean rooms, the system may have pressure sensors and airflow rate controls to maintain this pressure difference. More typically, the space will have exhaust ventilation only or the exhaust ventilation rate will be larger than the supply airflow rate. To determine whether the space is being exhausted as recommended, one should evaluate the design ventilation rates to and from the space. The total exhaust airflow rate should be greater than the total supply airflow rate. If only exhaust airflow is specified for the space, the design should include provision for make-up air, such as a transfer grille in a doorway. The system design should also be studied to determine if there is return airflow from the space, since recirculation of return air probably violates the ventilation recommendations.

5.1.4 Performance Measurements

This section describes procedures for measuring ventilation system performance using the techniques in Section 4. Measurement procedures are presented for air handlers, air distribution systems, ventilated spaces, and exhaust systems. Procedures are also presented to evaluate pressure relationships in the building and to measure building air change rates.

Scheduling these measurements can be difficult because ventilation system operation can vary from day to day and from hour to hour. Because of the variety and complexity of ventilation system designs, there is no general approach to scheduling performance measurements. If one understands the system controls and the timing of the internal loads, times and conditions can be identified when the measurements will be most useful.

Whether the system is constant volume or variable air volume (VAV) affects the scheduling of the measurements. Scheduling is more difficult in VAV systems because the airflow rates of terminal units and air handlers vary in response to changes in thermal loads. Operation under minimum and maximum loads are of most interest, but it can be difficult to predict when these conditions will occur without being familiar with the system and the building activities. Also, the timing of these loads depends on the season of the year and the daily weather patterns. The minimum and maximum thermal loads may not occur on the days when testing is performed. Minimum cooling loads will usually occur early in the morning when the building has not been occupied for very long. Maximum cooling loads usually exist in the mid-afternoon, after the building has been occupied for several hours and the internal and weather-induced (i.e., outdoor temperature and solar) loads have built up. The timing of the measurements is less difficult in constant volume systems.

Even though there is no general approach to scheduling ventilation system performance measurements, the following suggestions are useful. In constant volume systems, measurements should be made with minimum and with maximum outdoor air intake. Measurements in VAV systems should also be made under minimum and maximum outdoor intake, and these two conditions should be evaluated under minimum and maximum thermal loads. Therefore, there are four sets of measurement conditions for VAV systems. The building should be occupied normally during these measurements, under typical building loads, and with all ventilation equipment operating.

The procedures in this section discuss measurements of different parts of the ventilation system, i.e., air handlers, air distribution ducts, terminal units and diffusers. When making ventilation performance measurements under one set of system operation conditions, all of these system components should be measured over as short a time period as possible. Some discussions of ventilation system performance evaluation suggest that control setpoints and system components should be adjusted to make the system operate in a certain way, such as minimum outdoor air intake. However, such adjustments can cause the system to operate in a way in which it would never actually operate. If the system operation is modified, it should always be done by someone who understands the system. If such modifications are performed, they must be reported with the measurement results.

When measuring ventilation system performance, it is important to record the conditions during the measurement. These conditions include date, time of day, outdoor weather conditions (air temperature, relative humidity and solar radiation or cloudiness index from a nearby weather station) and building occupancy. In some systems the building automation system can provide useful information, such as damper positions, airflow rates and supply fan static pressures.

5.1.4.1 Air Handlers

Air handler performance is evaluated through measurement of the following parameters:

- **Supply airflow rate**
- **Supply air temperature and humidity**
- **Outdoor air intake rate**
- **Operating sequence**
- **Pressure differences within the system**

The **supply airflow rate** of an air handler is measured with a pitot tube or hot-wire anemometer traverse as described in Sections 4.8.1 and 4.8.2. The traverse plane location must meet the velocity profile requirements discussed in these sections. The traverse plane should be located where the velocity in the duct is relatively uniform, avoiding obstructions such as elbows and transitions. If no acceptable traverse plane can be identified, or if the supply air ductwork of the air handler is inaccessible, a vane anemometer traverse can be performed at the coils (see Section 4.8.3).

The **supply air temperature and relative humidity** are measured using the techniques described in Sections 4.1 and 4.4. The measurements should be performed downstream of the cooling and heating coils and should employ a "traverse" across the duct.

The **outdoor air intake rate** is measured at the air intake with a pitot tube or hot-wire anemometer traverse as described in Sections 4.8.1 and 4.8.2. However, acceptable traverse locations usually do not exist at outdoor air intakes. These intakes are often only a louvered opening in the plenum where the outdoor air and recirculation air mix. A vane anemometer traverse of the intake dampers is not acceptable.

The multiplicative method, as described in Section 4.8.6, can be used to measure the outdoor air intake rate when a traverse of the intake is not possible. The supply airflow rate Q_S is measured as discussed above, and the percent outdoor air intake %OA at the air handler is measured as described in Section 4.7. The supply airflow rate Q_S is multiplied by the percent outdoor air intake %OA to determine the outdoor air intake rate Q_O according to:

$$Q_O = \%OA \times Q_S.$$

The uncertainty in the determination of Q_O using the multiplicative method should be estimated as described in Section 4.8.6.

Comparing the operation of an air handler to the design intentions as described in the system **operating sequence** is an important part of air handler performance assessment. Because operating sequences vary from building to building, there is no simple procedure that can be used in all buildings. Assessing the operating sequence includes evaluation of the supply airflow rate modulation, the outdoor air intake control, and the control of the exhaust, return and intake dampers. The positions of these dampers can be inspected to determine the approximate level of outdoor air intake. Based on the outdoor air temperature and an understanding of the system controls, one determines if the air handler is bringing in the fraction of outdoor air that would be expected under the existing conditions. In a VAV system, one can evaluate the supply airflow rate relative to its capacity. The positions of the inlet vanes are inspected, or a value of the supply airflow rate or static pressure is obtained from the building automation system. Based on an understanding of the system design and controls, the supply airflow rate can be compared to the expected value based on the thermal loads in the space.

There are two approaches to evaluating whether the system operation is consistent with the operating sequence, examining the operation under the conditions that exist during the evaluation and adjusting the controls to induce other modes of operation. Assessing the operation of the air handler under the current conditions of building use and weather provides a realistic picture of its operation. However, it does not provide information for other conditions of interest. The system can be made to operate under other modes of operation by modifying control system setpoints and other control elements. However, such an adjustment may cause the system to operate in a mode in which it would never actually operate and therefore may provide information of questionable value. If such adjustments are made, they should always be done by someone who understands the system.

Static pressure measurements within an air handler are also useful in evaluating performance. Establishing a complete static pressure profile through a system is sometimes used in testing and balancing, but is beyond the scope of this manual. The static pressure difference across filters banks or system coils, measured using the techniques discussed in Section 4.3, can be compared to the design value. The pressure difference across the filters can be compared to the design value, and a larger measured pressure difference can indicate an installation problem or that the filters are heavily loaded and need to be changed. Of course, the loading of the filters should also be evaluated visually. A significant difference between the measured pressure difference across the coils and the design value can also indicate a problem, such as dirty coils.

5.1.4.2 Air Distribution System

Air distribution system performance involves the supply airflow rate to the building zones, the supply air temperature and relative humidity, and the outdoor airflow rate to the zones. The zones of interest within the distribution system vary from building to building and include individual floors, portions of floors served by individual terminal units, rooms and work stations. The performance at individual diffusers is also of interest. The following list contains performance parameters for three scales of the air distribution system:

Ducts
- **Supply airflow rate**
- **Air temperature and humidity**
- **Outdoor airflow rate**

Terminal Units
- **Supply airflow rate**
- **Air temperature and humidity**
- **Outdoor airflow rate**
- **Airflow rate modulation**

Supply Air Outlets
- **Supply airflow rate**
- **Air temperature and humidity**
- **Outdoor airflow rate**

Ducts

Supply airflow rates in the air distribution system are measured at various locations in the distribution ductwork. The person(s) performing the assessment must understand the distribution system layout before deciding where to make these measurements. Useful measurement locations include where the supply air duct enters a floor, major branches on a floor, supply ducts serving critical areas, and ducts serving areas that are being investigated because of indoor air quality complaints. The availability of design values for supply airflow rates can also affect decisions on where to perform the measurements.

Supply airflow rates in ducts are measured with a pitot tube or hot-wire anemometer traverse as described in Sections 4.8.1 and 4.8.2. The earlier discussion on the traverse plane location applies to measurements in ducts, as do the comments on scheduling the measurements. The airflow rate in the ducts should be measured close in time to the airflow rate measurements in the corresponding air handler.

The **supply air temperature and relative humidity** should be measured using the techniques described in Sections 4.1 and 4.4. A traverse of across a duct is usually not necessary, since these supply air properties are likely to be uniform in air distribution ducts.

The multiplicative method should be used to measure the **outdoor airflow rate** within a supply air duct. In this method, described in Section 4.8.6, the airflow rate measured in the supply duct is multiplied by the percent outdoor air intake measured at the air handler serving the duct. The supply airflow rate and percent outdoor air intake must be measured close to one another in time, under the same system operating conditions.

Terminal Units

It is also useful to measure the **supply airflow rate** at individual terminal units. The terminal units where these measurements are made depend on the goals of the assessment. Options include a random selection of terminal units in the building and terminal units serving spaces that are subject to indoor air quality complaints.

Supply airflow rates in terminal units are measured with a pitot tube or hot-wire anemometer traverse as described in Sections 4.8.1 and 4.8.2. The earlier discussion on the traverse plane location applies to measurements in terminal units, as do the comments on scheduling the measurements. In terminal units that do not induce any plenum or room air into the supply airstream, the supply airflow rate can be measured either upstream of downstream of the unit, whichever is more convenient. In induction units, the airflow upstream of the unit is referred to as the primary air and the downstream airflow is referred to as the supply air, and both airflow rates are of interest. The primary airflow rate is used to determine the amount of outdoor air delivered to the space served by the terminal unit. The supply airflow rate is related to the ability of the terminal unit to meet the thermal loads of the space. The following discussion distinguishes between primary and supply airflow rates as appropriate. In non-inducing units, the distinction between these two airflow rates in not necessary.

When measuring the supply airflow rate, the **supply air temperature and relative humidity** should also be measured in the supply airstream (as opposed to the primary airstream) using the techniques described in Sections 4.1 and 4.4. A traverse is usually not necessary, since these properties are likely to be uniform across the ducts.

The multiplicative method should be used to measure the **outdoor airflow rate** at a terminal unit. In this method, described in Section 4.8.6, the airflow rate of the primary air measured upstream of the terminal unit is multiplied by the percent outdoor air intake measured at the air handler serving the duct. The primary airflow rate and percent outdoor air intake must be measured close to one another in time, under the same system operating conditions. The application of the multiplicative method to dual-duct terminal units is beyond the scope of this manual.

The **airflow rate modulation** of the terminal unit should be evaluated. Depending on the type and design of the terminal unit, it will modulate the airflow rates of the primary, induced and supply air. The control of these airflow rates in response to thermal loads and time of day should be compared to the design intentions. This comparison requires a good understanding of the system design and experience with the design and operation of terminal units.

Supply Air Outlets

The **supply airflow rate** delivered by an individual supply air outlet is measured with a flow hood as described in Section 4.8.4. If the location of the outlet prohibits the use of a flow hood, then one can use a rotating vane anemometer as described in Section 4.8.5. Vane anemometers can only be used at openings and grills, and not at high velocity, induction diffusers. The outlets at which these measurements are made depend on the goals of the assessment. Options include a random selection of outlets within the building and outlets serving spaces with indoor air quality complaints.

The **supply air temperature and relative humidity** in the supply air leaving the outlet are measured using the techniques described in Sections 4.1 and 4.4. This measurement can be made in the duct just upstream of the outlet or by inserting the measurement probe into the outlet.

The multiplicative method should be used to measure the **outdoor airflow rate** at an outlet. In this method, described in Section 4.8.6, the airflow rate measured at the outlet is multiplied by the percent outdoor air intake measured in the supply airstream. In systems that do not induce return or room air at the terminal unit, the percent outdoor air intake measured at the air handler is used in the calculation. If the diffuser is served by an induction unit, the percent outdoor air intake at the air handler must be corrected to account for the induced return or room air. This correction is described in Section 4.8.6. The application of the multiplicative method to supply air outlets in dual-duct systems is beyond the scope of this manual.

5.1.4.3 Ventilated Space

Evaluation of ventilation system performance within ventilated spaces involves thermal comfort and air mixing. The acceptability of thermal comfort depends on air temperature, radiant temperature, relative humidity and air speed. Only air temperature and relative humidity are measured in these evaluations. Air mixing is more complex, and no standardized procedures exist to evaluate the mixing or distribution of ventilation air within a space.

Air temperature and relative humidity are measured in the ventilated space using the techniques described in Sections 4.1 and 4.4. The measurement locations should be based on the layout of the space, the zoning of the ventilation system, and the objectives of the tests. If the purpose of the assessment is to evaluate the system performance throughout the building, the locations should be well distributed within the building. Measurements should be made on different floors and in different temperature control zones on a floor. There is no simple rule for the number of measurement points, the minimum amount of floor area per measurement point or the locations of the measurement points. In general, there should be one measurement location for each temperature control zone. If this approach results in too many measurements, one can make one measurement for every 500 to 1000 m^2 (5000 to 10,000 ft^2) of floor area. Another guideline is to make 6 to 10 measurements per floor. The locations should be well distributed between different floors of the building, in both core and perimeter areas, and in spaces with different occupant densities. The air temperature and relative humidity should be measured within the ventilated space at heights corresponding to the occupant locations.

If the assessment is intended to address a portion of the building subject to complaints, then more measurements should be made within the complaint area. Measurements should also be conducted in portions of the building that are not associated with complaints.

The measurements should be scheduled based on an understanding of the ventilation system control and operation and on the timing of the building thermal loads. Because of the variety and complexity of ventilation system controls and thermal loads, it is difficult to generalize on the scheduling of these measurements. Ideally, the measurements should be scheduled under minimum, typical and maximum loads. Given constraints on time and resources, these measurements may only be possible under some of the conditions of interest.

Mixing of ventilation air and room air affects the delivery of the ventilation air to the occupants and the removal of internally-generated contaminants. Assessing **airflow patterns** may be important in some situations, but no standardized evaluation procedures are available. Tracer gas techniques are being developed to address room air mixing in the field, sometimes described as "ventilation effectiveness," but the measurement procedures are still at the research stage. The only technique available for assessing airflow patterns is to release smoke in the space using smoke tubes (see Section 3.10) and observe the speed at which the smoke disperses. No standardized procedure has been developed for the use of smoke, but it is generally thought that if smoke is released within the occupied space and disperses in a few seconds, then the circulation is probably good. If the smoke does not move or moves very slowly, then the air mixing is a potential problem. Smoke can also be released at supply outlets to qualitatively determine the direction and speed of air delivery.

5.1.4.4 Exhaust Systems

Exhaust systems remove pollutants near their source before they can move to other locations in the building. Assessing the performance of exhaust systems involves the following parameters:

- **Airflow rate**
- **Operation**
- **Make-up air**

Exhaust system **airflow rates** can be measured at the fan itself to obtain the total exhaust airflow rate of the system and at individual exhaust vents. The airflow rate at the fan is measured with a pitot tube or hot-wire anemometer traverse as described in Sections 4.8.1 and 4.8.2. The earlier discussion on the traverse plane location applies to measurements in exhaust systems.

The exhaust airflow rate at individual exhaust vents can be measured with flowhoods, as described in Section 4.8.4. In some cases, the location of the vent and the configuration of the room may make it impossible to position a flowhood. A rotating vane anemometer can then be used, as described in the Section 4.8.5.

Since exhaust systems are usually constant volume, the airflow rate can be measured at any time. However, because the operation of other building ventilation systems can affect exhaust system performance, these other systems should be operating normally during the exhaust measurements.

The **operation** of the exhaust system relative to its intended operation is an important performance issue. Exhaust fans are controlled according to a variety of control strategies including continuous operation, time schedules, interlocks with other equipment, and manual control. The intended operation of the exhaust system needs to be understood in order to evaluate its control, and the system should be inspected at critical times to determine if the actual operation agrees with the design. If a system is supposed to operate continuously, it should be checked during both occupied and unoccupied hours. If a system is suppose to operate on a specific schedule, it should be checked both during the intended hours of operation and at times when it is supposed to be off. An exhaust system interlocked with another piece of equipment, e.g., an emergency generator, should be checked when the associated equipment is on and when it is off. If the exhaust system is controlled manually, the on/off switch should be tested for proper operation.

The provision of adequate **make-up air** to an exhaust system is critical to its proper operation. Make-up air is air brought into a building from the outdoors to replace the air that is exhausted. Exhaust systems without an adequate source of make-up air will not be able to move sufficient amounts of air. Make-up air can be provided by a variety of sources: dedicated make-up air fans that supply air to the space being exhausted, an oversupply of air by the central air handling system, transfer grilles, and vents to the outdoors. If there is no intentional source of make-up air, the exhaust fan will pull whatever air it can into the space from openings in interior and exterior walls. Such uncontrolled make-up air sources can cause undesirable airflows within the building.

Evaluating the adequacy of make-up air involves design evaluation and building inspection to determine the intended source of make-up air and the actual source. Measured exhaust airflow rates that are significantly below design can be a sign of inadequate make-up air. If the make-up air is provided by a separate air handler, the performance of the make-up air system should be evaluated to determine if it can provide adequate make-up air. If there is no mechanical source of make-up air, pressure relationships to adjoining spaces should be assessed as described in the next section. The size and sign of pressure differences between the space being exhausted and adjoining spaces can indicate the source of the make-up air.

5.1.4.5 Building Pressure Relationships

Pressure differences within buildings are important aspects of ventilation system performance. Based on the balance of supply and exhaust or return airflow rates for different spaces, and the relative airtightness of exterior walls and interior partitions, significant pressure differences can exist at exterior openings and between building zones. These pressure differences can cause pollutant movement within and into buildings and can affect the operation of ventilation systems. As discussed earlier, the ventilation requirements for a space may specify that a pressure difference of a certain direction, and perhaps a certain size, be maintained between the space and the surrounding spaces. Assessing building pressure relationships involves the two categories of pressure differences:

- **Pressure differences across the exterior envelope**
- **Pressure differences between adjoining spaces**

Assessing **pressure differences across the exterior envelope** involves a determination of the sign and sometimes the size of pressure differences across openings where outdoor air and pollutants could enter the building, such as loading docks and entrances. Although buildings are often designed to maintain a positive interior pressure relative to the outdoors, air leakage through interior partitions and poor control of ventilation system airflow rates can lead to negative pressures relative to the outdoors. Negative pressures can bring in motor vehicle exhaust, odors from trash dumpsters and other contaminants from the outdoors.

Pressure difference are measured with differential pressure gauges (Section 3.3), with one port connected to the building interior and the other to the outdoors. Smoke tubes, described in Section 3.10, can also be used to determine the direction of the pressure difference at an opening. The sign of the pressure difference should be determined at all entrances, loading docks and other locations where air can enter a building.

Pressure differences are affected by ventilation system operation and weather, and therefore there is no ideal time to measure these pressures. Low outdoor air temperatures increase the strength of the stack effect and the possibility that air will flow into the building at lower floors. Therefore, pressure difference measurements should be made during cold weather if possible. Pressure differences are also of interest in different modes of ventilation system operation. System operation varies between buildings, and optimal timing of these measurements requires a good understanding of the system design and its actual operation.

The size and sign of **pressure differences between adjoining spaces** are also measured with differential pressure gauges. The sign alone can be determined with smoke tubes. The specific locations at which these pressure differences are measured depend on the building layout.

Pressure differences should be evaluated from special-use spaces (such as bathrooms, smoking lounges and copy rooms) to adjoining spaces. In addition, pressure differences should be evaluated at points where air can enter a space from elsewhere in the building, such as stairwell and elevator doors. Air in stair and elevator shafts can contain pollutants from other locations in building such as underground garages. Internal pressure differences are affected by ventilation system operation, and the discussion on scheduling exterior pressure difference measurements relative to system operation also applies to the measurement of internal pressure differences.

5.1.4.6 Whole Building Air Change Rate

The whole building air change rate is the total rate at which outdoor air enters the building, including both outdoor air intake through the air handler and air infiltration through the building envelope. There are two techniques to determine whole building air change rates:

- **Tracer gas decay**
- **Equilibrium carbon dioxide analysis**

The **tracer gas decay** technique for determining whole building air change rates (ASTM Standard E741) is described in Section 4.9.1 and determines the whole building air change rate over a period of several hours. Air change rates are affected by outdoor air temperature, wind speed and direction, and ventilation system operation, and a measured air change rate only provides information for the conditions during the test. To fully characterize air change rates in a mechanically ventilated building, many measurements are needed for a range of weather and ventilation system operation. A complete characterization requires many months of testing, and is usually not possible given the resources available for most ventilation assessments. When only a small number of measurements are possible, they should be conducted under representative operating conditions, such as minimum and maximum outdoor air intake.

Accurate measurements of the whole building air change rates using tracer gas decay requires that the tracer gas concentration in the building can be characterized by a single value, i.e., there are no significant variations in concentration within the building. This requirement is sometimes referred to as the "single zone" or "well mixed" assumption, and can be met by injecting the tracer gas uniformly throughout the building. The airflow patterns within the building may also help keep the concentration uniform. It will not be possible to meet this requirement in some buildings, and these buildings can not be tested with this technique.

It is not appropriate to measure tracer gas decay rates for individual floors, rooms or other zones. If different concentrations and decay rates are measured in different zone of a building, the test results are not valid.

Equilibrium carbon dioxide analysis is another way to estimate whole building air change rates and is discussed in Section 4.9.2. As discussed earlier, the technique is generally inappropriate in office buildings. If it can be used, it determines the whole building air change rate under the conditions of weather and system operation during the test. As discussed for the tracer gas decay technique, many measurements under different conditions are needed to characterize the air change rate of a building.

This technique is based on assumptions that limit its usefulness in mechanically ventilated commercial buildings. First, the carbon dioxide concentration in the building must be at equilibrium. This requires that the carbon dioxide generation rate (building occupancy) and the building air change rate are constant for several hours, with the number of hours dependent on the air change rate. As discussed in Section 4.9.2, the time to reach equilibrium is long relative to typical occupancy patterns in office buildings. Also, the air change rate will change during the test if the outdoor air intake is modulated by the ventilation system, violating another critical assumption. The single-zone requirement discussed for the tracer gas technique also applies to equilibrium analysis, i.e., the carbon dioxide concentration must be uniform throughout the building.

5.2 Applications

This section contains a general discussion of how space-use analysis, design evaluation and performance measurement are used in preventive maintenance programs, indoor air quality diagnosis and energy assessment.

5.2.1 Preventive Maintenance

An important application of ventilation system evaluation is preventive maintenance. Building and equipment maintenance is crucial for controlling building energy use, thermal comfort and indoor air quality, and a preventive maintenance program is a sound strategy for ensuring good performance and avoiding future problems in new and existing buildings.

A preventive maintenance program should include an initial ventilation evaluation when the program is first put into place. This initial evaluation should include a space-use analysis, design evaluation and a comparison between the space-use and design. These efforts will serve to document all available design information. The initial evaluation should also include performance measurements on the ventilation system. In new buildings, performance measurements are conducted during the system testing and balancing.

After the initial phase of the preventive maintenance program, periodic follow-up assessments should be performed roughly once a year. These follow-ups should include an update of the space-use analysis to determine if there have been any significant changes affecting the building's ventilation requirements. The follow-up assessments should also include performance measurements.

Follow-up assessments should also be performed after major space-use changes or ventilation equipment modifications. In the case of a space-use change, an updated space-use analysis is needed, followed by a comparison of the new ventilation requirements with the ventilation system design. In cases of ventilation system modifications, these changes will be documented as part of the system design evaluation. Performance measurements are also appropriate and should be done as part of the testing and balancing of the modified system. Even if the modification takes place in only one part of the building, it can affect ventilation in the rest of the building. Therefore, measurements are needed in more than just the zones affected directly by the modification.

5.2.2 Air Quality Diagnosis

Ventilation system performance assessment is an important tool when diagnosing the causes of indoor air quality problems in buildings, and the role of ventilation evaluation in the diagnosis process is described in *Building Air Quality* (EPA/NIOSH 1991). The application of ventilation assessment procedures in indoor air quality diagnosis depends on whether a preventive maintenance program is in place. If there is no such program, then one needs to perform all of the procedures: space-use analysis, system design evaluation and performance measurements. If there is a preventive maintenance program, then the space-use analysis and system design evaluation should have already been performed. However, it is important to make sure that the space-use analysis and system evaluation are up to date, i.e., to confirm that there have been no space-use changes or ventilation equipment modifications that could be linked to the indoor air quality complaints. The ventilation evaluation should focus on the portion of the building where the complaints are occurring. However, other areas in the building must be considered because pollutants can move within buildings. Also, performance measurements should be conducted in non-complaint areas to provide a reference point for comparison.

5.2.3 Energy Assessment

Building ventilation performance is important when evaluating building energy use. The goal of energy-related ventilation evaluations is to determine if excessive amounts of outdoor air are being brought into the building by the ventilation system or if there are large envelope infiltration rates. A current space-use analysis and system design evaluation is needed to determine the building ventilation requirements. These requirements are then compared to the measured values of the outdoor air intake rates to determine if these rates are excessive. If there is a preventive maintenance program in place, a space-use analysis and system evaluation is not needed. The performance measurements should focus on the whole building air change rate and the outdoor air intake rates through the ventilation systems. The difference between the whole building air change rate and the total of the outdoor air intake rates for all the air handling systems is equal to the building envelope infiltration rate. If the building envelope is very leaky, or the ventilation system airflow rates are poorly controlled, the infiltration rate can be large, i.e., 0.5 ach or more. Pressure differences across the exterior envelope should also be measured, since these pressures induce envelope leakage

5.3 References

ASTM, 1993, Standard Test Method for Determining Air Change in a Single Zone by Means of Tracer Gas Dilution, E741, American Society for Testing and Materials, Philadelphia.

ASHRAE, 1989, Ventilation for Acceptable Indoor Air Quality, Standard 62-1989, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta.

EPA/NIOSH, 1991, Building Air Quality. A Guide for Building Owners and Facility Managers, EPA/400/1-91/033, U.S. Environmental Protection Agency, DHHS (NIOSH) Publication No. 91-114, National Institute for Occupational Safety and Health.

APPENDIX A TERMINOLOGY

The terms defined in this section are used in this manual and in other discussions of ventilation performance.

air:
- exhaust: air discharged from the building to the outdoors by mechanical ventilation.
- makeup: air brought into a building from the outdoors to replace air that is exhausted.
- outdoor: air taken from outdoors and, therefore, not previously circulated through the system.
- primary: supply air discharged by an air outlet or terminal unit prior to the entrainment of air from the conditioned space or return air plenum.
- recirculated: return air passed through the air conditioning unit before being again supplied to the conditioned space.
- return: air returned from the conditioned space.
- secondary: air from the conditioned space or return air plenum that is entrained by the primary air discharged by an air outlet or terminal unit.
- spill: return air that is exhausted from the building
- supply: air delivered by the air conditioning unit to the conditioned space, generally containing both outdoor and recirculated air.

air change rate: airflow rate into or out of a building or room in volume units per hour divided by the building or room volume in identical volume units. The air change rate can refer to either outdoor or supply airflow.

air conditioner, unitary: an evaporator, compressor, and condenser combination designed in one or more assemblies, the separate parts designed to be assembled together.

air conditioning unit: an assembly of equipment for the treatment of air so as to control, simultaneously, its temperature, humidity, cleanliness and distribution to meet the comfort requirements of a conditioned space.

air diffuser: a circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary air with secondary room air.

air inlet: device or opening through which air is withdrawn from a ventilated space.

air outlet: device or opening through which air is discharged into a ventilated space.

anemometer: device to measure velocity of airflow at a point.

vane: device consisting of rotating propeller-type vanes; the air velocity is indicated from the rotational speed of the vanes.

thermal: device that relies on the cooling effect of the airflow to change the temperature of a heated body in proportion to the air speed.

arrestance: a measure of the ability of a filter to remove test dust from air as described in ASHRAE Standard 52.1-1992; in this test, the weight of test dust captured by the filter in the test airstream divided by the weight of dust in a control airstream that bypasses the filter, expressed as a percentage.

atmospheric dust spot efficiency: a measure of the ability of a filter to remove atmospheric dust from air as described in ASHRAE Standard 52.1-1992; in this test, the ratio of the light-transmission of stains on paper targets upstream and downstream of the filter, expressed as a percentage.

barometer: device for measuring atmospheric pressure.

building envelope: elements of the building that enclose the interior space, including all external building materials, windows and walls.

ceiling plenum: space above the suspended ceiling and below the floor above that is used as part of the return air system.

cold deck: the cooling section of a dual-duct air conditioning system.

cooling coil: an arrangement of pipe or tubing that uses a refrigerant or secondary coolant to cool, and in some cases dehumidify, air.

damper: a device used to vary the volume of air passing through an air outlet, air inlet or duct.

density: mass per unit volume.

diffuser: a circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary supply air with secondary room air.

duct: a passageway made of sheet metal or other suitable material, not necessarily leaktight, used for conveying air.

economizer: a system of dampers, temperature and humidity sensors and motors which maximizes the use of outdoor air for cooling.

entrainment: the capture of part of the surrounding air by the airstream discharged from an outlet (sometimes called secondary air motion).

fan coil unit: a heating and cooling unit, containing a small fan and heating and coiling coils, that provides supplementary heating and cooling to perimeter zones of buildings, generally mounted below exterior windows.

fan performance curve: a graphical presentation of static or total pressure and power input over a range of air volume flow rate at a stated inlet air density and fan speed. The range of air volume flow rate which is covered generally extends from shutoff (zero air volume flow rate) to free delivery (zero fan static pressure).

filter: a device to remove particulates from air.

grille: a louvered or perforated covering for an air inlet or outlet which can be located in a sidewall, ceiling or floor.

heating coil: coil that uses a heat transfer fluid, condensing refrigerant or direct electrical resistance elements to heat air.

hot deck: the heating section of a dual-duct air conditioning system.

humidity:

absolute: ratio of the mass of water to the total mass of a moist air sample.

relative: ratio of the mol fraction of water vapor present in the air to the mol fraction of water vapor present in saturated air at the same temperature and barometric pressure.

inch of water (in. w.g.): in the I-P system, a unit of pressure equal to the pressure exerted by a column of liquid water 1 inch high at a temperature of 39.2 °F.

induction: the capture of part of the ambient air by the jet action of the primary airstream discharging from an air outlet or terminal unit.

induction unit: a heating and cooling unit that delivers a small quantity of conditioned air (primary), from a remote central air handler, through high-velocity jets to induce a large quantity of room or plenum air (secondary) into the supply airstream. Heating and cooling coils may be located in the primary or secondary airstream.

infiltration: air flow into a space through unintentional openings such as cracks and joints.

louver: an assembly of sloping vanes intended to permit air to pass through and to inhibit transfer of water droplets.

manometer: an instrument for measuring pressure.

outlet: device or opening through which air is discharged into a ventilated space.

ceiling: a round, square, rectangular or linear air diffuser located in the ceiling, which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied space.

slotted: a long, narrow air distribution outlet, comprised of deflecting members, located in the ceiling, sidewall or sill, with an aspect ratio greater than 10, designed to distribute supply air in varying directions and planes, and arranged to promote mixing of primary air and secondary room air.

vaned: a register or grille equipped with vertical and/or horizontal adjustable vanes.

outlet velocity: the average velocity of air emerging from an opening, fan or outlet, measured in the plane of the opening.

pitot tube: double-tube instrument from which the flow velocity can be calculated with one opening in the tube facing the flowing stream to measure total pressure and another set of openings perpendicular to the stream to measure static pressure.

pressure: the normal forced exerted by a homogeneous fluid per unit of area.

absolute: pressure above a perfect vacuum; the sum of gage pressure and atmospheric pressure.

atmospheric: pressure of the outdoor atmosphere; standard atmospheric reference pressure is defined as 101.325 kPa, 1013.25 millibars, 14.696 psi or 29.921 inches of mercury at 32 °F.

differential: difference in pressure between two points.

drop: decrease in fluid pressure, as from one end of a duct to the other, due to friction, dynamic losses and changes in velocity pressure.

static: pressure exerted by a fluid at rest; in a flowing fluid, the total pressure minus the velocity pressure.

total: in fluid flow, the sum of the static pressure and the velocity pressure.

velocity: in moving fluid, the pressure due to the velocity and density of the fluid, expressed by the velocity squared times the fluid density, divided by two.

psychrometer: instrument for measuring relative humidity with wet- and dry-bulb thermometers.

aspirated: psychrometer having mechanical means for rapidly circulating air to be tested over dry and wet bulbs.

sling: psychrometer having two matched thermometers, one with its bulb wetted and the other dry, capable of being whiled rapidly on a sling to indicate the temperature differences related to relative humidity.

register: a grille equipped with an integral damper or control valve.

secondary air: the air surrounding an outlet that is captured or entrained by the initial outlet discharge airstream.

set point: the value of the controlled condition at which a controlled device is set to operate.

system curve: a graphic presentation of the pressure versus volume flow rate characteristics of a particular system.

system effect: fan inlet restrictions, fan outlet restrictions or other conditions that influence installed fan performance.

temperature:

dewpoint: the temperature at which moist air becomes saturated (100% relative humidity) with water vapor when cooled at constant pressure.

dry-bulb: the temperature of air indicated by an ordinary thermometer.

wet-bulb: temperature indicated by a psychrometer when the bulb of one thermometer is covered with a water-saturated wick over which air is caused to flow at approximately 4.5 m/s (900 ft/min).

thermometer: instrument for measuring temperature.

variable air volume (VAV): use of varying airflow to control the condition of air in a space in response to varying thermal loads, in contrast to the use of constant flow with varying temperature.

ventilation: the process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

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APPENDIX B ALTITUDE CORRECTION FACTORS FOR BAROMETRIC PRESSURE (from ASHRAE Standard 111-1988)

Altitude (m)	Correction Factor
0	1.000
30	0.996
60	0.993
90	0.989
120	0.986
150	0.982
180	0.979
210	0.975
240	0.971
270	0.969
300	0.965
330	0.962
360	0.958
390	0.955
420	0.951
450	0.948
480	0.945
510	0.941
540	0.938
570	0.934
600	0.931
630	0.927
660	0.924
690	0.921
720	0.917
750	0.914
780	0.910
810	0.907
840	0.905
870	0.901

Altitude (ft)	Correction Factor
0	1.000
100	0.996
200	0.993
300	0.989
400	0.986
500	0.982
600	0.979
700	0.975
800	0.971
900	0.968
1000	0.964
1100	0.961
1200	0.957
1300	0.954
1400	0.950
1500	0.947
1600	0.944
1700	0.940
1800	0.937
1900	0.933
2000	0.930
2100	0.926
2200	0.923
2300	0.920
2400	0.916
2500	0.913
2600	0.909
2700	0.906
2800	0.903
2900	0.899

Altitude (m)	Correction Factor
900	0.898
960	0.892
1020	0.885
1080	0.879
1140	0.872
1200	0.866
1260	0.859
1320	0.853
1380	0.847
1440	0.841
1500	0.835
1560	0.829
1620	0.823
1680	0.817
1740	0.810
1800	0.804
1950	0.789
2100	0.776
2250	0.761
2400	0.747
2550	0.733
2700	0.719
2850	0.706
3000	0.693
4500	0.571
6000	0.469
7500	0.381
9000	0.308
10500	0.247
12000	0.198

Altitude (ft)	Correction Factor
3000	0.896
3200	0.890
3400	0.883
3600	0.877
3800	0.870
4000	0.864
4200	0.857
4400	0.851
4600	0.845
4800	0.838
5000	0.832
5200	0.826
5400	0.820
5600	0.814
5800	0.807
6000	0.801
6500	0.786
7000	0.772
7500	0.757
8000	0.743
8500	0.729
9000	0.715
9500	0.701
10000	0.688
15000	0.564
20000	0.460
25000	0.371
30000	0.297
35000	0.235
40000	0.185

Instructions

To determine the correction factor at an altitude A that is not on the chart, find the altitudes below and above A, A_1 and A_2. The correction factor at A is given by:

$$CF(A) = CF_1 + \left[\frac{(CF_2 - CF_1)}{(A_2 - A_1)}\right] \times (A - A_1)$$

CF_1 and CF_2 and are the corrections factors at A_1 and A_2.

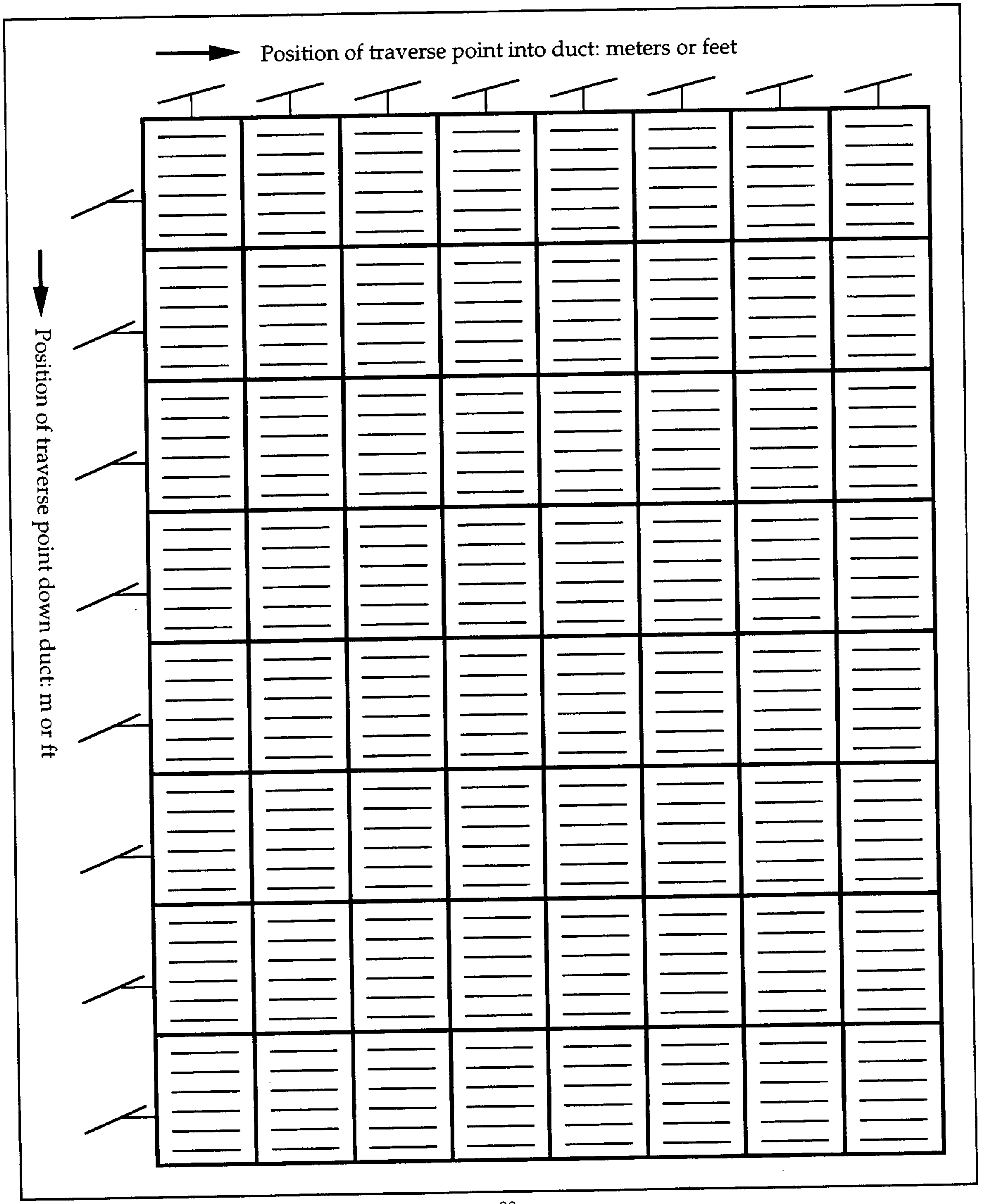
Example

If the altitude A = 1231 m (4038 ft), then A_1 = 1200 m (4000 ft) and A_2 = 1260 m (4200 ft). CF_1 = 0.866 (0.864) and CF_2 = 0.859 (0.857). Using the equation, the correction factor at 1231 m (4038 ft) equals 0.862.

FORM C1: DATA FROM PITOT TRAVERSE OF RECTANGULAR DUCT

Air Handler Number: ____________________
Supply, Intake or Exhaust Duct: __________
Date of Test: ____________________________
Time of Start of Test: ______________________
Time of End of Test: ______________________

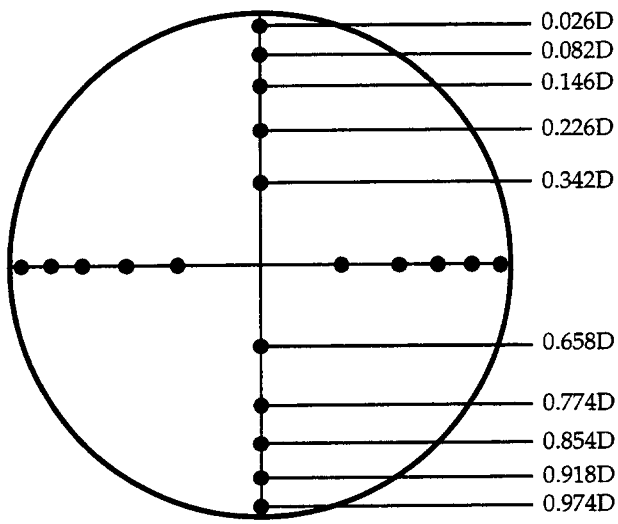
Air temperature: ________________________ °C/°F
Barometric pressure: ____________________ kPa/in. Hg
Duct width: _____________________________ m/in.
Duct height: ____________________________ m/in.
Velocity pressure units - Pa or in. w.g.: ______________



FORM C2: DATA FROM PITOT TRAVERSE OF ROUND DUCT

Air Handler Number: ________________
Supply, Intake or Exhaust Duct: ________
Date of Test: ________________________
Time of Start of Test: ________________
Time of End of Test: _________________

Air temperature: ______________________°C/°F
Barometric pressure: __________________kPa/in. Hg
Duct diameter: ________________________m/in.
Velocity pressure units - Pa or in. w.g.: ____________



Horizontal Traverse

0.026D	0.082D	0.146D	0.226D	0.342D

0.658D	0.774D	0.854D	0.918D	0.974D

Vertical Traverse

0.026D	0.082D	0.146D	0.226D	0.342D

0.658D	0.774D	0.854D	0.918D	0.974D

FORM C3: DATA FROM PITOT TRAVERSE OF OVAL DUCT

Air Handler Number: ________________

Supply, Intake or Exhaust Duct: ________

Date of Test: ________________

Time of Start of Test: ________________

Time of End of Test: ________________

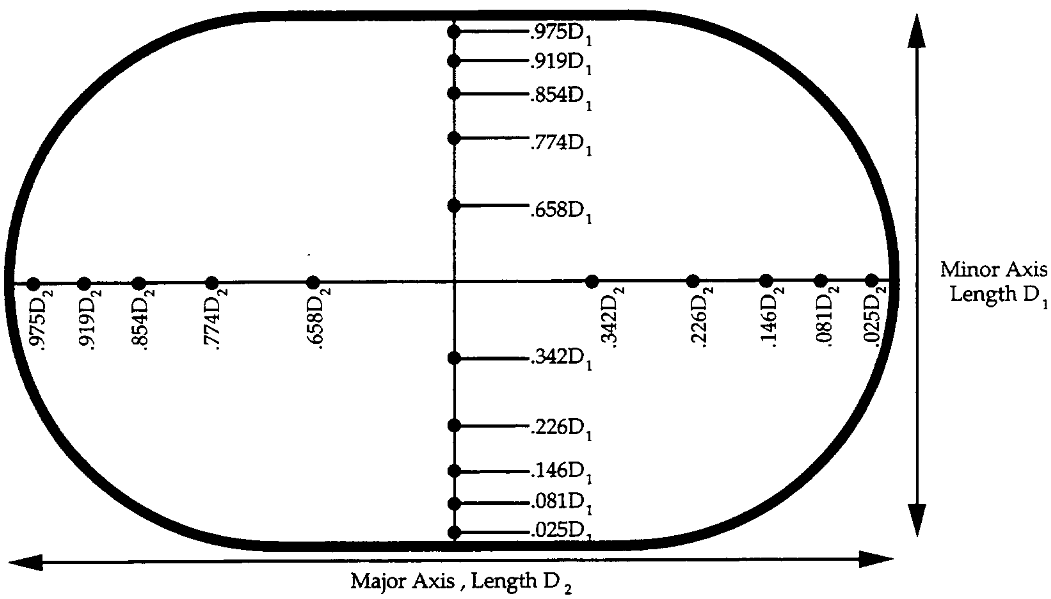
Air temperature: ________________ °C/°F

Barometric pressure: ________________ kPa/in. Hg

Major axis of duct: ________________ m/in.

Minor axis of duct: ________________ m/in.

Velocity pressure units - Pa or in. w.g.: ____________



Major Axis Traverse

$0.025D_2$	$0.081D_2$	$0.146D_2$	$0.226D_2$	$0.342D_2$

$0.658D_2$	$0.774D_2$	$0.854D_2$	$0.919D_2$	$0.975D_2$

Minor Axis Traverse

$0.025D_1$	$0.081D_1$	$0.146D_1$	$0.226D_1$	$0.342D_1$

$0.658D_1$	$0.774D_1$	$0.854D_1$	$0.919D_1$	$0.975D_1$

FORM C4: DATA FROM HOT WIRE TRAVERSE OF RECTANGULAR DUCT

Air Handler Number: ____________________

Supply, Intake or Exhaust Duct: __________

Date of Test: ____________________

Time of Start of Test: ____________________

Time of End of Test: ____________________

Air temperature: ____________________ °C/°F

Barometric pressure: ____________________ kPa/in. Hg

Duct width: ____________________ m/in.

Duct height: ____________________ m/in.

Velocity units - m/s or fpm: ____________________

→ Position of traverse point into duct: meters or feet

↓ Position of traverse point down duct: m or ft

FORM C5: DATA FROM HOT WIRE TRAVERSE OF ROUND DUCT

Air Handler Number: ____________________

Supply, Intake or Exhaust Duct: __________

Date of Test: ____________________

Time of Start of Test: ____________________

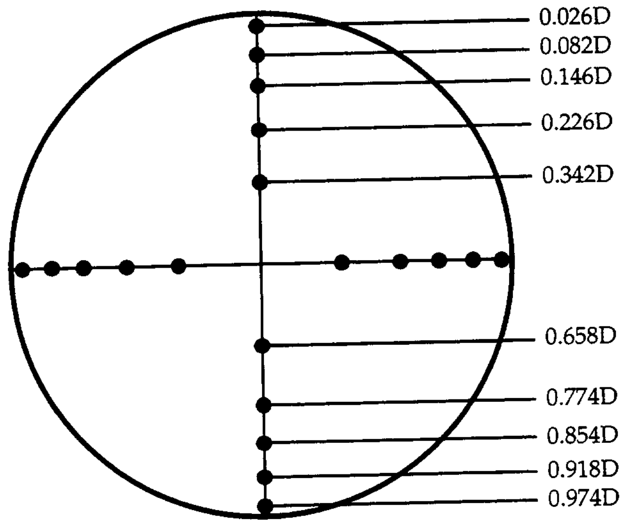
Time of End of Test: ____________________

Air temperature: ____________________ °C/°F

Barometric pressure: ____________________ kPa/in. Hg

Duct diameter: ____________________ m/in.

Velocity units - m/s or fpm: ____________________



Horizontal Traverse

0.026D	0.082D	0.146D	0.226D	0.342D

0.658D	0.774D	0.854D	0.918D	0.974D

Vertical Traverse

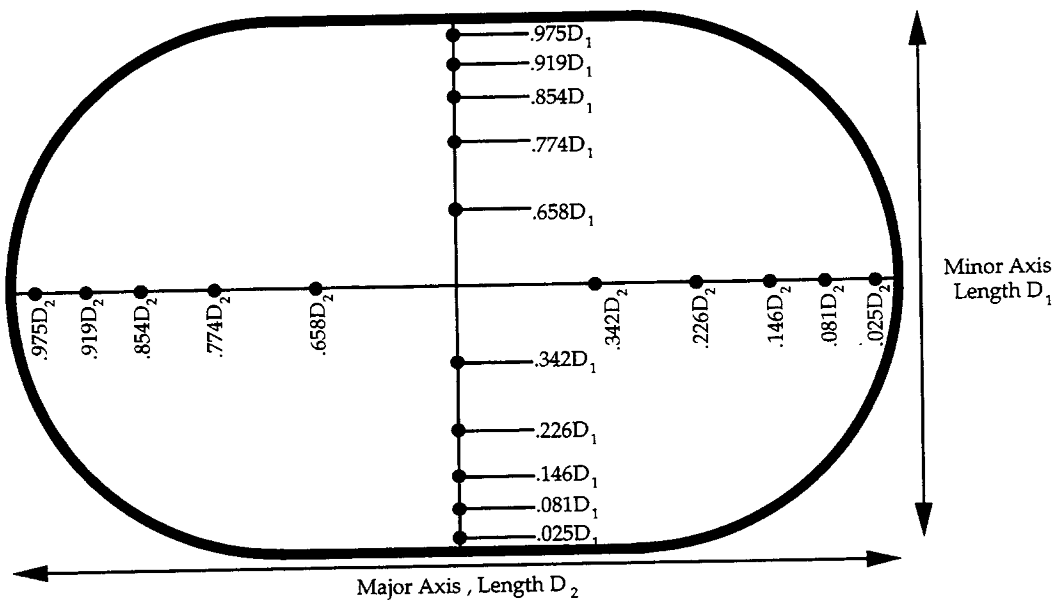
0.026D	0.082D	0.146D	0.226D	0.342D

0.658D	0.774D	0.854D	0.918D	0.974D

FORM C6: DATA FROM HOT WIRE TRAVERSE OF OVAL DUCT

Air Handler Number: ____________________
Supply, Intake or Exhaust Duct: ____________
Date of Test: ______________________________
Time of Start of Test: ____________________
Time of End of Test: ______________________

Air temperature: ______________________ °C/°F
Barometric pressure: __________________ kPa/in. Hg
Major axis of duct: ____________________ m/in.
Minor axis of duct: ____________________ m/in.
Velocity units - m/s or fpm: ____________________



Major Axis Traverse

$0.025D_2$	$0.081D_2$	$0.146D_2$	$0.226D_2$	$0.342D_2$

$0.658D_2$	$0.774D_2$	$0.854D_2$	$0.919D_2$	$0.975D_2$

Minor Axis Traverse

$0.025D_1$	$0.081D_1$	$0.146D_1$	$0.226D_1$	$0.342D_1$

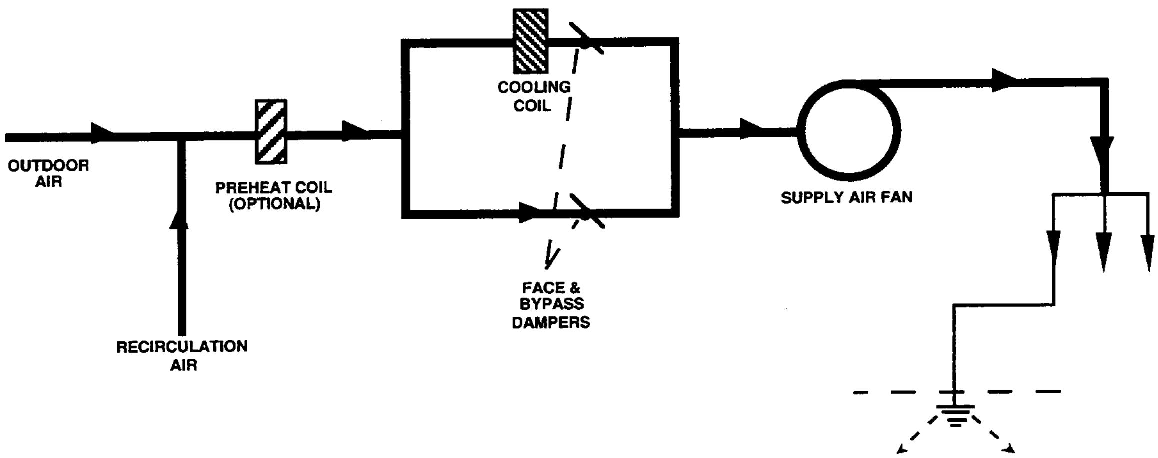
$0.658D_1$	$0.774D_1$	$0.854D_1$	$0.919D_1$	$0.975D_1$

APPENDIX D CLASSIFICATION OF CENTRAL VENTILATION SYSTEMS

This appendix describes several types of central, all-air ventilation systems.

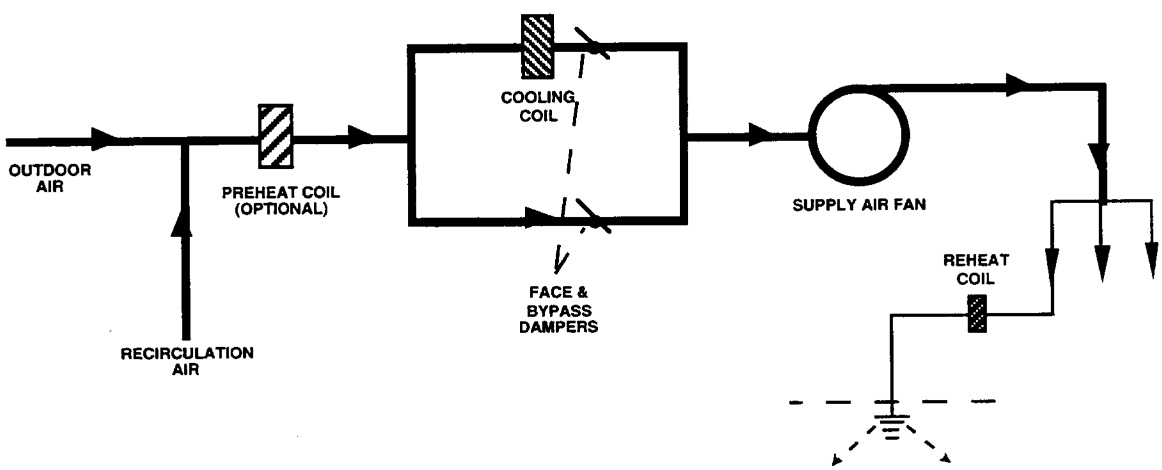
A Single Duct, Constant Volume

The air handler provides a constant supply airflow rate to multiple zones with small thermal load variations. The thermal load is met by varying the supply air temperature. The supply air temperature is controlled by varying the amount of heating or cooling at the air handler, the relative rates of outdoor air intake and recirculation, the position of face and bypass dampers in the air handler, or a combination of these approaches.



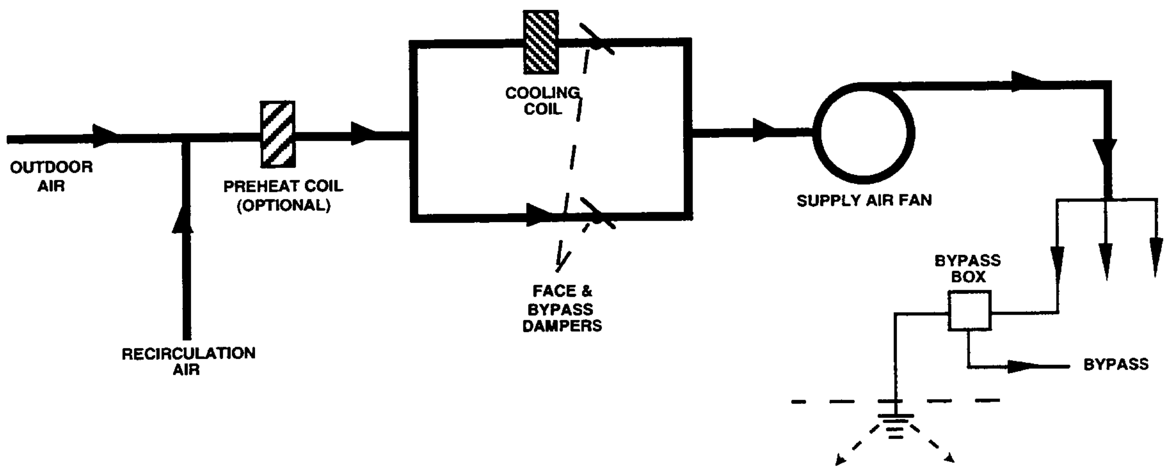
B Single Duct, Constant Volume, Multiple Zone Reheat

The air handler provides a constant supply airflow rate to multiple zones with different thermal loads. The loads in the zones are met by varying the supply air temperature. The supply air temperature is controlled by varying the amount of heating or cooling at the air handler, the relative rates of outdoor air intake and recirculation, the position of face and bypass dampers in the air handler, or a combination of these approaches. Further temperature control is provided by reheat coils in the zones.



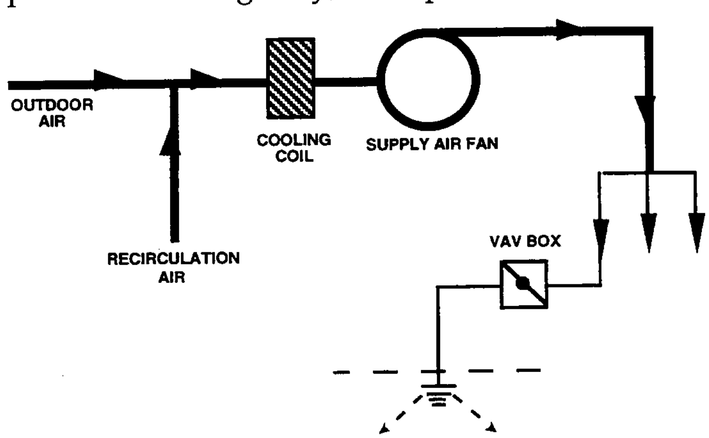
C Single Duct, Constant Volume, Multiple Zone Bypass

The air handler provides a constant supply airflow rate to multiple zones with different thermal loads. The loads in the zones are controlled by varying the supply air temperature and the supply airflow rate to each zone. The supply air temperature is controlled by varying the amount of heating or cooling at the air handler, the relative rates of outdoor air intake and recirculation, the position of face and bypass dampers in the air handler, or a combination of these approaches. Further temperature control in individual zones is provided through the use of a bypass box which dumps some of the supply air into the return air plenum.



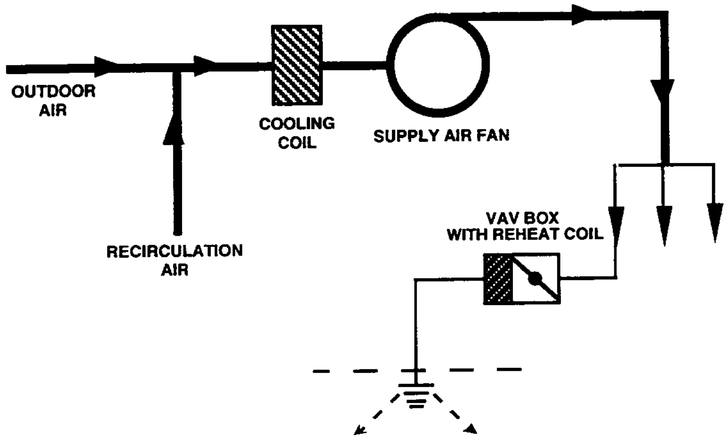
D Single Duct, Variable Air Volume

The air handler provides constant temperature supply air at about 10 °C (50 °F) to VAV units located in the ceiling plenum. In each zone, the VAV unit controls the supply airflow rate to meet the cooling loads in the zone. The supply airflow rate of the air handler therefore varies in response to space load variations in the building. A true VAV system provides cooling only, with perimeter zones heated by some other system.



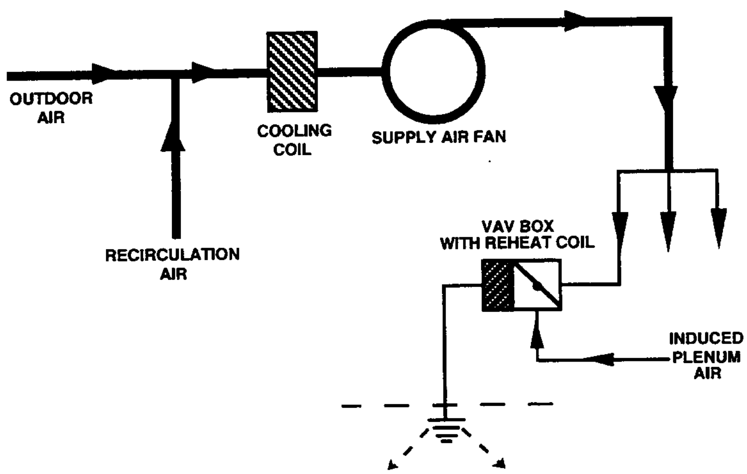
E Single Duct, Variable Air Volume, Reheat

This system is a modification of a true VAV system capable of providing both heating and cooling. Heat is provided in or near the terminal units after the supply airflow rate has been reduced to a minimum amount.



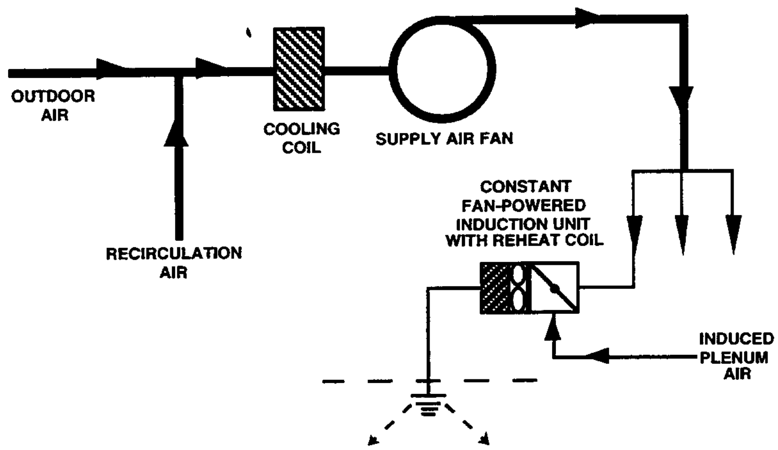
F Single Duct, Variable Air Volume, Induction

A VAV air handler provides primary air to unpowered terminal units that induce plenum or room air into the supply airstream. The total airflow rate of the primary and induced air are roughly constant. Variations in space load are met by varying the relative proportions of the primary and induced air. Reheat coils or some other form of auxiliary heat are required when heat gain in the room and ceiling can not balance envelope losses and cooling loads from the primary supply air.



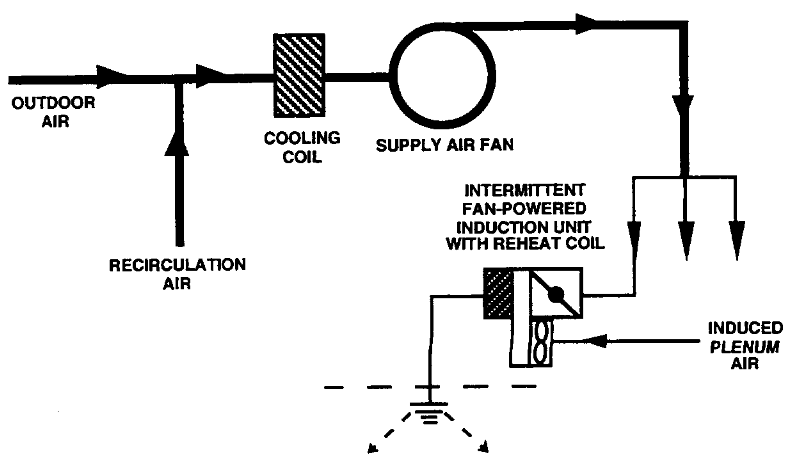
G Single Duct, Variable Air Volume, Fan Powered, Constant Fan

A VAV air handler supplies primary air to fan-powered induction units that are installed in series with the primary supply airflow. The fan-powered units run continuously and operate at a relatively constant volume. In each zone, the unit mixes the required quantity of primary supply air with induced return air from the plenum. Terminal units in exterior zones have heating coils for winter heating requirements. The heating coil is not activated until the primary air volume is reduced to a minimum value.



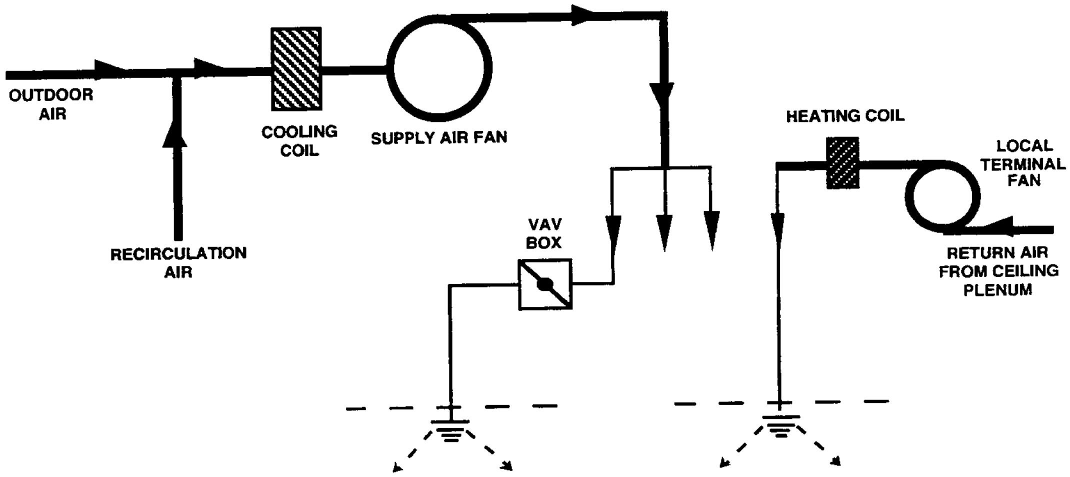
H Single Duct, Variable Air Volume, Fan Powered, Intermittent Fan

A VAV air handler supplies primary air to fan-powered induction units that are installed in parallel with the primary supply airflow. The unit modulates the primary supply air in response to the cooling needs of the zone and operates the fan-powered unit when induced air is needed to meet the heating requirements. The primary air and the induced air mix within a common plenum within the fan-powered unit.



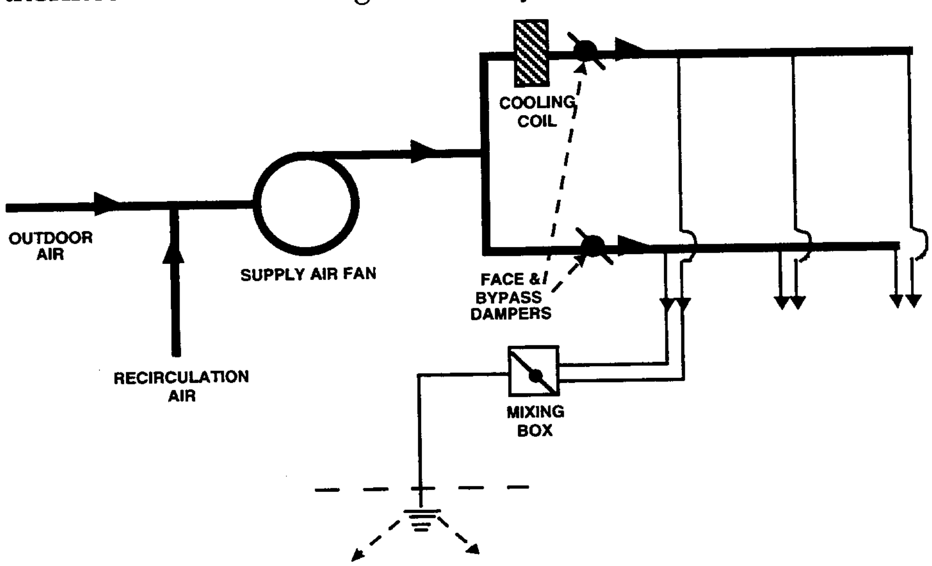
I Single Duct, Variable Air Volume, Dual Conduit

Two ventilation airstreams are used to condition the space, one system to meet year round cooling loads and a second to offset transmission losses. The first system is a conventional central VAV system that provides year round cooling to meet space cooling loads. The second system, located in perimeter zones, operates at constant volume with the air temperature varied to meet transmission losses. In some systems, this local terminal fan only operates during peak loads.



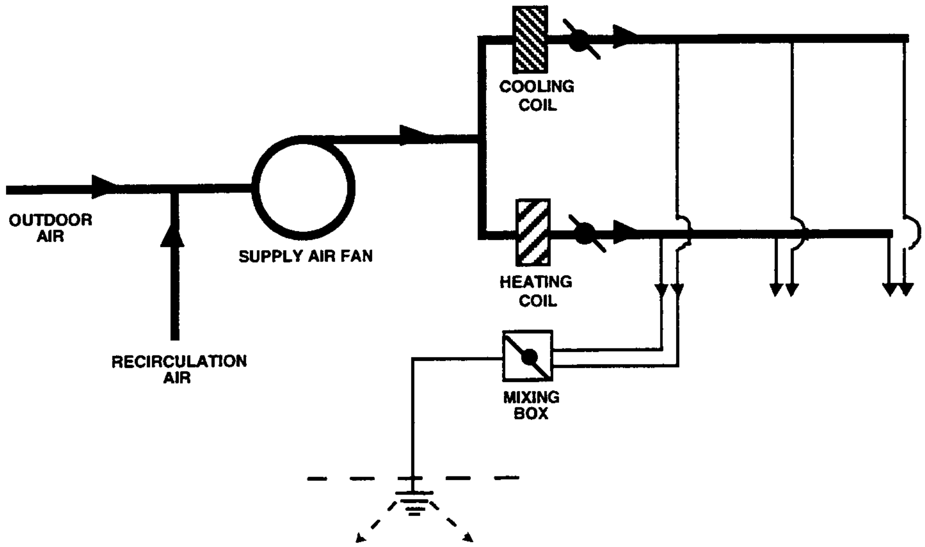
J Dual Duct, Constant Volume

The air handler supplies a constant volume of supply air to multiple zones, with the supply fan blowing through cooling coil and bypass sections connected to cold and hot ducts respectively. These two ducts run through the building to unpowered mixing boxes in the ceiling plenum, which mix the warm and cold air in proper proportions to meet the loads in the zone. The dampers in the mixing boxes are controlled by zone thermostats. The mixing boxes may contain reheat coils.



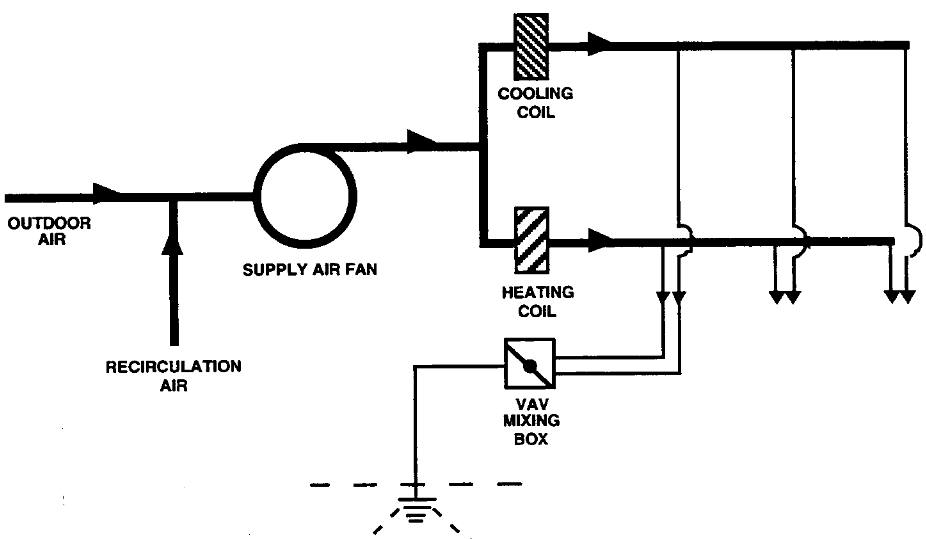
K Dual Duct, Constant Volume, Reheat

The air handler provides a constant supply airflow rate to multiple zones. The supply airstream is split into two flows, one blowing through cooling coils and the other through heating coils. The hot and cold air decks are connected to unpowered mixing boxes in the ceiling plenum, which mix the hot and cold air to meet the loads in the zone. Interior zone mixing boxes may only be connected to the cold deck.



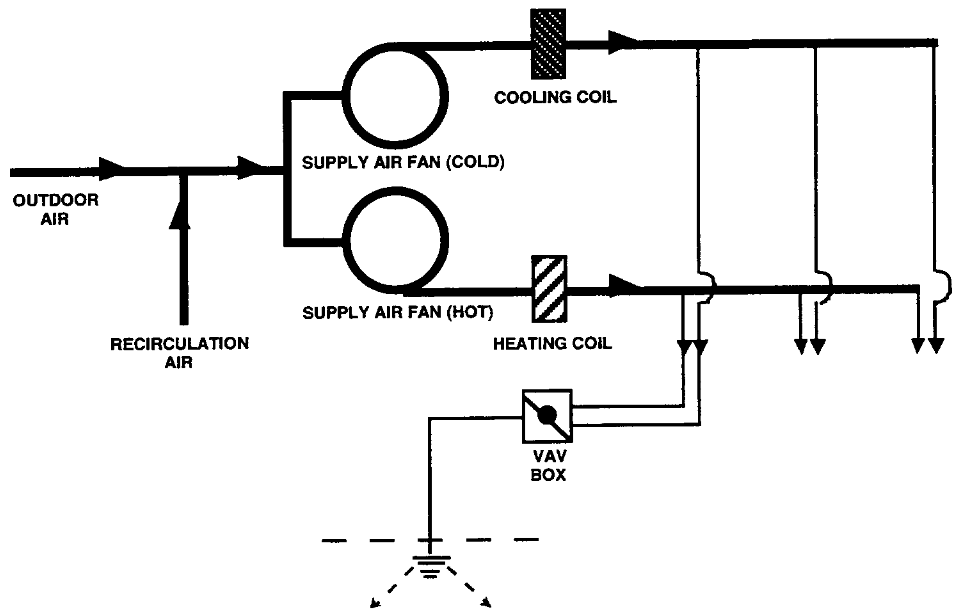
L Duct Duct, Variable Air Volume, Single Fan

A VAV air handler supplies air to multiple zones, with the supply fan blowing through cooling and heating coil sections connected to cold and hot decks. The two decks run through the building to VAV mixing boxes in the ceiling plenum, which mix the hot and cold air to meet the loads in the zone. Interior zone boxes may be connected to only the cold deck.



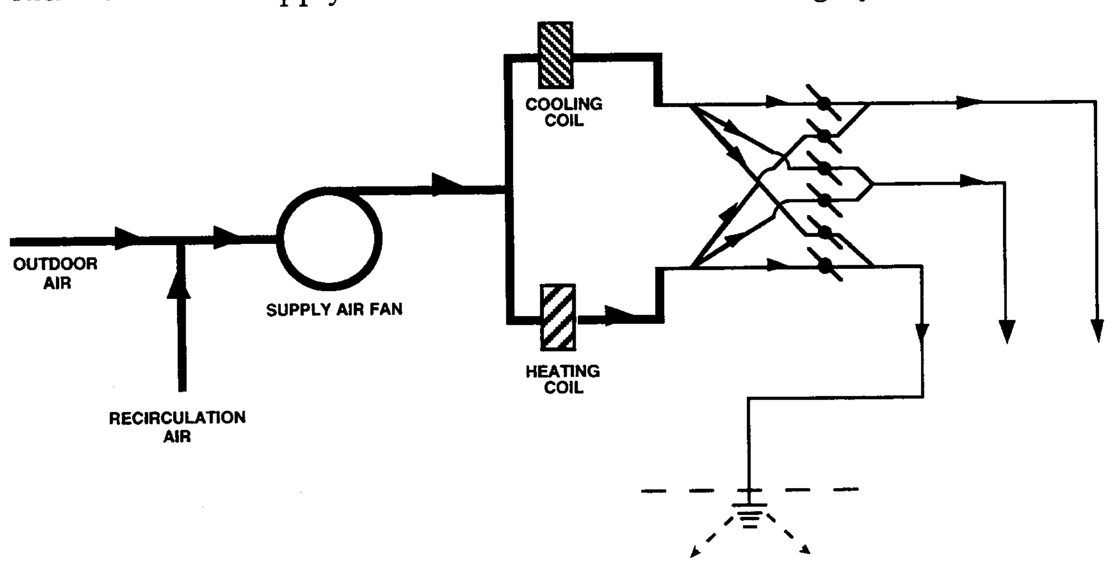
M Dual Duct, Variable Air Volume, Dual Fan

In this system, separate supply fans serve the cold and hot decks. The two duct systems run through the building to VAV mixing boxes in the ceiling plenum, which mix the hot and cold air to meet the loads in the zone. The dampers in the boxes are controlled by zone thermostats. Interior zone boxes may be connected to only the cold duct, while exterior zones will be connected to both the hot and cold ducts.



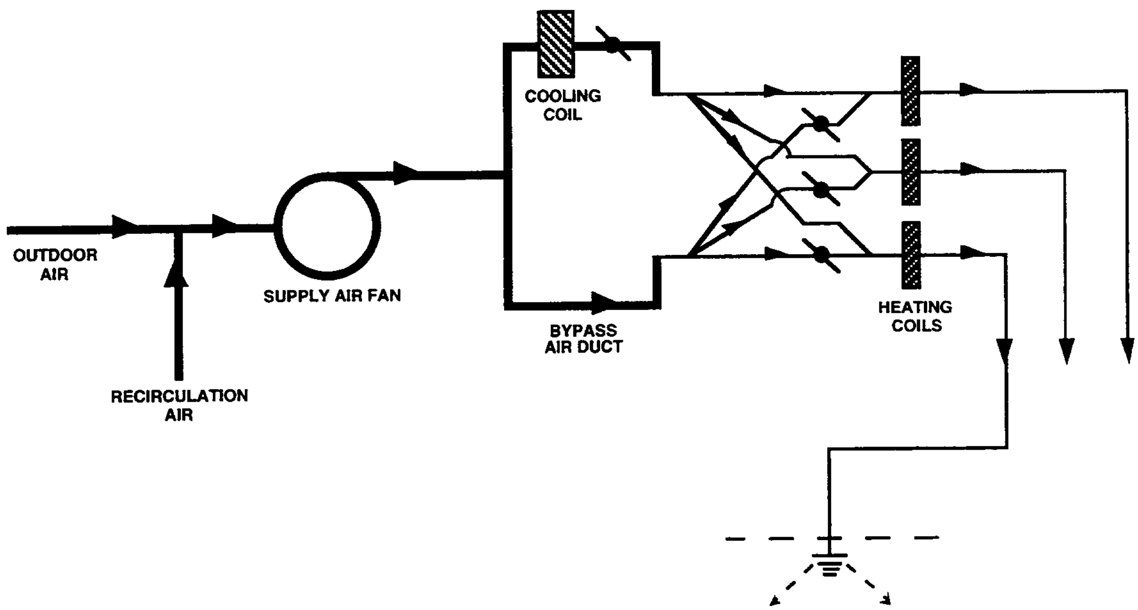
N Multizone, Constant Volume

The air handler supplies a constant volume of supply air to multiple zones. The space load of each zone is met through a mixture of the hot and cold airstreams carried to the zone by a single duct. The hot and cold airstreams for each zone mix at the air handler, with a set of dampers for each zone. The supply airflow rate to each zone is roughly constant.



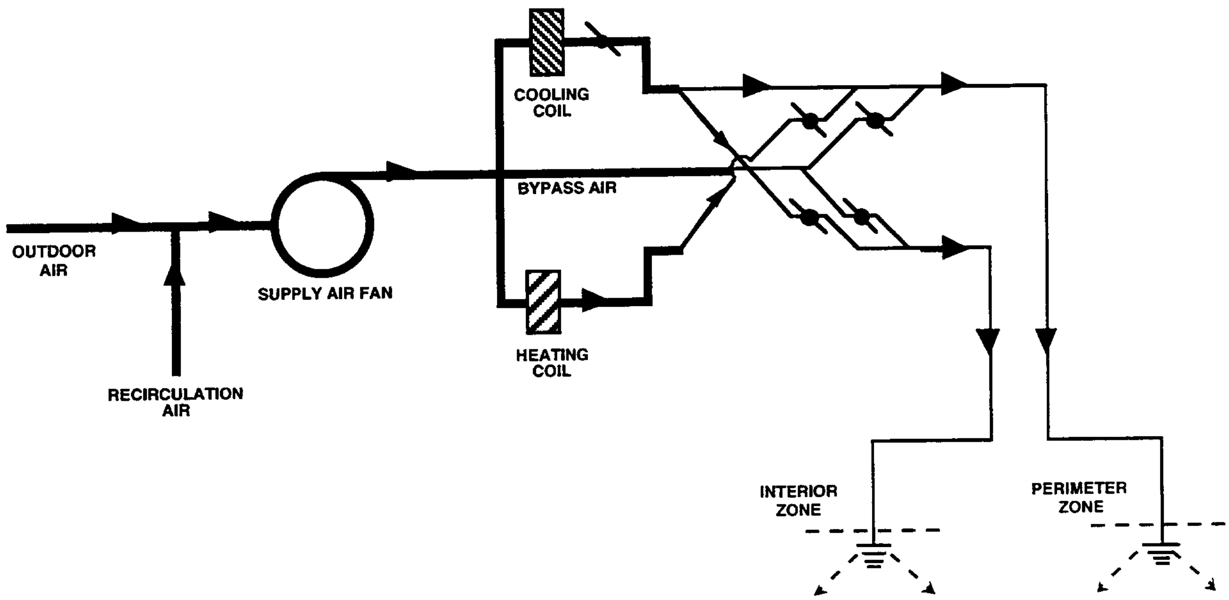
O Constant Volume, Blow-Through Bypass

The air handler provides a constant supply airflow rate to multiple zones, with the supply fan blowing air through cooling coils or through a bypass section around the coils. The cold and bypass decks are split so that there is a cold duct and bypass air duct for each zone. The two supply airflows mix within the mechanical room, with a damper in the bypass air duct and a heating coil downstream of where the two flows merge. A constant quantity of air is supplied to each zone, and the supply air temperature to each zone is varied to meet cooling or heating loads by modulating the bypass damper and using of the heating coil. The heating coil is not used unless all of the zone's supply air is bypass air. Interior zones may not have a heating coil in their ducts.



P Texas Multizone, or Three-Deck Multizone

The air handler provides a constant supply airflow rate to multiple zones, with the supply fan blowing through a cooling coil, a bypass section or a heating coil. The thermal load of each zone is met through a mixture of the cold, bypass and hot airstreams that is carried by single duct to the zone. The three airstreams for each zone mix at the unit, employing dampers in the three decks. Interior zones are generally not connected to the hot deck. The heating coils are activated only if the bypass air can not meet the loads. The supply airflow rate to each zone is roughly constant depending on the pressure drop through each coil and the position of the mixing dampers.



APPENDIX E VERIFICATION OF ASHRAE STANDARD 62-1989 VENTILATION RATE RECOMMENDATIONS

This section describes how to verify that a building is being ventilated according to the ventilation rate recommendations in ASHRAE Standard 62-1989, *Ventilation for Acceptable Indoor Air Quality*. While Standard 62-1989 is a design standard, the forward of the standard states that "the conditions specified by this standard must be achieved during the operation of buildings ... if acceptable indoor air quality is to be achieved." The performance of a ventilation system can be different from its design due to improper system installation, operation and maintenance. Therefore, it is important to verify that a system is performing as intended through measurement.

The verification procedures described in this section can be used to determine if the outdoor air ventilation rate of a building or a portion of a building is consistent with the recommendations in the Ventilation Rate Procedure of Standard 62-1989. The Ventilation Rate Procedure contains other requirements including the existence of acceptable contaminant levels in the intake air and no unusual indoor contaminants or sources. These other requirements are not covered in this appendix. The procedures described here can be used to determine if a building, or a portion of a building, is being ventilated at a rate that agrees with the minimum outdoor air ventilation recommendations in Table 2 of the standard. A portion of this table, applicable to spaces in commercial buildings, is given in Table E.1.

	Occupancy	Outdoor Air Requirement			
Activity	people/100 m^2 or 1000 ft^2	L/s-person	cfm/person	L/s-m^2	cfm/ft^2
Office space	7	10	20		
Reception areas	60	8	15		
Telecommunication centers and data entry areas	60	10	20		
Conference rooms	50	10	20		
Smoking lounges	70	30	60		
Restrooms (exhaust flow per water closet or urinal)		25	50		
Duplicating and printing				2.50	0.50
Classrooms	50	8	15		
Laboratories	30	10	20		
Libraries	20	8	15		
Auditoriums	150	8	15		
Retail					
Basement and street level	30			1.50	0.30
Upper floors	20			1.00	0.20
Malls and arcades	20			1.00	0.20

Table E.1 Outdoor Air Ventilation Requirements (ASHRAE Standard 62-1989)

In order to determine if the outdoor air ventilation rates in a building are in compliance with the minimum ventilation recommendations of Standard 62-1989, one performs the following five steps. This procedure can be performed for a whole building, the portion of a building served by a single ventilation system, or an individual office or workstation within a building.

Step 1 Analyze Space-Use

The first step is to perform a space-use analysis as described in Section 5.1.1. The objective of this analysis is to develop a list of building zones containing the information needed to calculate the recommended ventilation rate based on Standard 62-1989. This list should contain the following items: the zone location, the air handler(s) serving each zone; the number of occupants in the zone (based on an analysis of the current space use); the number of water closets and urinals in bathrooms; and the floor area of the zone. Table E.2 is an example of such a list based on the example discussed in Section 5.1.1.

Building Floor	Zone	Air Handler	Number of Occupants	Number of Water Closets	Floor Area m^2 (ft^2)
4	Office space, east	S-1	125		2500 (27,000)
	Office space, west	S-2	125		2500 (27,000)
	Conference room	S-1	20		50 (540)
	Restroom, men	EX-1		3	10 (110)
	Restroom, women	EX-1		3	10 (110)
3	Office space, east	S-1	250		2500 (27,000)
	Office space, west	S-2	250		2500 (27,000)
	Smoking lounge	EX-2	30		50 (540)
	Restroom, men	EX-1		3	10 (110)
	Restroom, women	EX-1		3	10 (110)
2	Office space, east	S-1	175		2500 (27,000)
	Data entry area, west	S-2	600		2500 (27,000)
	Conference room	S-1	20		50 (540)
	Restroom, men	EX-1		3	10 (110)
	Restroom, women	EX-1		3	10 (110)
1	Office space, east	S-1	210		3000 (32,000)
	Reception area, west	S-2	60		2000 (22,000)
	Duplicating center	EX-3			50 (540)
	Restroom, men	EX-1		3	10 (110)
	Restroom, women	EX-1		3	10 (110)

Table E.2 Example List of Building Zones

Step 2 Determine Ventilation Requirements for Building and Zones

Based on the list of building zones, the required ventilation rates are calculated for each of these zones based on the recommendations in Standard 62-1989. Outdoor air requirements are determined by multiplying the number of people in the zone by the recommended ventilation rate per person for that type of space. Exhaust ventilation requirements for restrooms are based on the number of water closets multiplied by the recommended exhaust airflow rate per water closet. Exhaust air requirements for printing and duplicating areas, and other space types, are calculated by multiplying the floor area by the recommended airflow rate per unit floor area. Table E.3 contains the outdoor air ventilation rates for each zone in the example.

Building Floor	Zone	Air Handler	Ventilation Rate L/s (cfm)	Comments
4	Office space, east	S-1	1250 (2500)	Outdoor air
	Office space, west	S-2	1250 (2500)	Outdoor air
	Conference room	S-1	200 (400)	Outdoor air
	Restroom, men	EX-1	75 (150)	Exhaust airflow, no recirculation
	Restroom, women	EX-1	75 (150)	Exhaust airflow, no recirculation
3	Office space, east	S-1	2500 (5000)	Outdoor air
	Office space, west	S-2	2500 (5000)	Outdoor air
	Smoking lounge	EX-2	900 (1800)	Exhaust airflow, no recirculation
	Restroom, men	EX-1	75 (150)	Exhaust airflow, no recirculation
	Restroom, women	EX-1	75 (150)	Exhaust airflow, no recirculation
2	Office space, east	S-1	1750 (3500)	Outdoor air
	Data entry area, west	S-2	6000 (12,000)	Outdoor air
	Conference room	S-1	200 (400)	Outdoor air
	Restroom, men	EX-1	75 (150)	Exhaust airflow, no recirculation
	Restroom, women	EX-1	75 (150)	Exhaust airflow, no recirculation
1	Office space, east	S-1	2100 (4200)	Outdoor air
	Reception area, west	S-2	480 (900)	Outdoor air
	Duplicating center	EX-3	125 (250)	Exhaust airflow, no recirculation
	Restroom, men	EX-1	75 (150)	Exhaust airflow, no recirculation
	Restroom, women	EX-1	75 (150)	Exhaust airflow, no recirculation

Table E.3 Ventilation Requirements for the Building Zones

The ventilation requirements for each of the air handlers are determined by adding the requirements for all of the spaces served by each air handler. Table E.4 contains the outdoor air ventilation rates for the air handlers in the example.

Air Handler	Ventilation Requirement L/s (cfm)	Comments
S-1	8000 (16,000)	Minimum outdoor air
S-2	10,230 (20,400)	Minimum outdoor air
EX-1	600 (1200)	Exhaust air to outdoors
EX-2	900 (1800)	Exhaust air to outdoors
EX-3	125 (250)	Exhaust air to outdoors

Table E.4 Ventilation Requirements for the Building Air Handlers

ASHRAE Standard 62-1989 contains a procedure for adjusting the total outdoor air requirement of an air handler that serves multiple zones with different outdoor air requirements and for adjusting the ventilation rate for spaces with variable occupancy patterns.

The outdoor air ventilation requirement of an individual office or workstation is determined by multiplying the number of occupants by 10 L/s (20 cfm) per person.

Step 3 Configure Ventilation Systems for Measurement

The ventilation rate measurements should be performed when the building is normally occupied and all ventilation equipment is being operated. If it is not possible to make the measurements when the building is occupied, it is still essential that all ventilation equipment is being operated as if it were occupied. The operation of all building ventilation equipment should be verified by visual inspection before, during and after the measurements. The air handlers should be operated at their minimum level of outdoor air intake. The building operator should adjust the system to induce minimum outdoor air intake in a manner that is consistent with the normal operation of the system. The operation of the system at minimum outdoor air should be verified by visually inspecting the outdoor air intake dampers before, during and after the measurements.

Step 4 Measure the Ventilation Rate

The ventilation rates should be measured using the procedures described in Section 5.1.4. The measurements are described below for a single air handler, an exhaust fan, a whole building and an individual office or workstation.

Air Handler

The outdoor airflow rate of an air handler should be measured with a velocity traverse of the outdoor air intake if possible. If a traverse can not be performed at the intake, then the multiplicative method should be used in which the measured supply airflow rate of the air handler is multiplied by the measured percent outdoor air intake.

Exhaust Fan

The airflow rate of an exhaust fan should be measured with a velocity traverse. If a duct with an acceptable configuration for a velocity traverse is not accessible, then the airflow rates of the duct branches connected to the exhaust fan should be measured individually and added together. The airflow rates of these ducts should be measured with a velocity traverse if possible, or alternatively with a flowhood at the individual exhaust vents.

Whole Building

The outdoor airflow rate of a whole building is measured by making individual measurements of all of the air handlers serving the building and adding them together. A tracer gas measurement of the whole building air change rate is not acceptable since it includes infiltration, and the requirements of Standard 62-1989 should not be met in a mechanically ventilated building with infiltration air.

Office or Workstation

The outdoor air ventilation rate to an individual office or workstation is determined with the multiplicative method. The supply airflow rate at the outlet(s) serving the office is measured using a flowhood, vane anemometer or a traverse of the duct serving the outlet(s). The percent outdoor air intake is measured at the air handler serving the outlet with a tracer gas or temperature balance. If the ventilation system has induction terminal units, then the percent outdoor air at the air handler must be corrected for induction air.

For workstations in an open-plan office, it is not always possible to identify the specific supply air outlets serving an individual workstation. In this case, the following procedure should be used to estimate the supply airflow rate to the workstation.

Determine the total floor area of the open-plan office A.

Determine the total number of supply air outlets in the office area N_o.

Divide the number of supply air outlets N_o by the floor area A to determine the average number of outlets per floor area N_A.

Multiply the average number of outlets per floor area N_A by the floor area of the workstation A_{ws} to determine the number of outlets serving the workstation N_{ws}. N_{ws} may not be a whole number.

Measure the supply airflow rate at the four outlets closest to the workstation.

Determine the average supply airflow rate for these four outlets Q_4.

The supply airflow rate to the workstation Q_{ws} equals Q_4 multiplied by N_{ws}.

Q_{ws} is the supply airflow rate that should be multiplied by the percent outdoor air intake to determine the outdoor air ventilation rate of the workstation.

Room with Exhaust

The exhaust airflow rate from an individual room should be measured at the exhaust air inlet(s) serving the room using a flowhood, vane anemometer or a traverse of the duct serving the inlet(s).

Step 5 Compare Design and Measurements

The measured outdoor airflow rates are then compared to the ventilation requirements determined in Step 2. The measured outdoor air and exhaust ventilation rates should be greater than or equal to these requirements to comply with the minimum ventilation rate recommendations in Standard 62-1989.

APPENDIX F RESOURCES

This section contains a bibliography on ventilation performance assessment and a list of organizations that can provide additional information on the subject.

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F.2 ORGANIZATIONS

Air Movement and Control Association (AMCA)
30 West University Drive
Arlington Heights, IL 60004
708 394-0150

American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE)
1791 Tullie Circle NE
Atlanta, GA
404 636-8400

Associated Air Balancing Council (AABC)
1518 K Street NW
Washington, DC 20005
202 737-0202

American Industrial Hygiene Association (AIHA)
2700 Prosperity Avenue Suite 250
Fairfax, VA 22031
703 849-8888

American Society for Testing and Materials (ASTM)
1916 Race Street
Philadelphia, PA 19103
215 299-5400

Building Owners and Managers Association International (BOMA)
1201 New York Avenue NW Suite 300
Washington, DC 20005
202 408-2662

U.S. Environmental Protection Agency (EPA)
Indoor Air Quality Information Clearinghouse
P.O. Box 37133
Washington, DC 20013
800 438-4318

National Environmental Balancing Bureau (NEBB)
1385 Piccard Drive
Rockville, MD 20850
301 977-3698

Sheet Metal and Air-Conditioning Contractors National Association (SMACNA)
4201 Lafayette Center Drive
Chantilly, VA 22021
703 803-2980