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Measurements of Outdoor Air Distribution in an Office Building

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ABSTRACT

The National Institute of Standards and Technology (NIST) has performed a study of outdoor air distribution in an office building. This study, performed in the Portland East Federal Office Building in Portland, Oregon, is a follow-up to a study in which outdoor airflow rates to the whole building were measured. This report focuses on the delivery of outdoor air to smaller sections of the building.

The technique used to measure these "local" outdoor airflow rates is referred to as the multiplicative method. It consists of measuring the supply airflow rate and the percentage of outdoor air in the supply air, and then multiplying them together to obtain the outdoor airflow rate. Outdoor airflow rates were measured to various zones of the building ranging in size from an individual workstation or office cubicle to the entire space served by an air handler. In addition, both automated and manual sampling techniques were demonstrated for measuring local age of air to determine air change effectiveness and to provide information on the distribution and mixing of ventilation air.

Some of the major findings of this study are as follows. When performing supply airflow rate measurements, the selection of the measurement location and the use of recommended guidelines were important for obtaining reliable results. Measurements of the same supply airflow rate made at different locations in the system were generally within 20% of each other. Also, while appropriate levels of outdoor air were brought in by the main air handling system, this outdoor air did not always reach the individual diffusers in the occupied space. In this study, the measured outdoor airflow rates per person, when considered on the scale of an air handler, were consistent with the recommendations of 10 L/s per person given in ASHRAE Standard 62-1989. However, measured outdoor airflow rates per person on smaller scales, i.e., in spaces served by individual terminal units and at individual workstations, were sometimes below the recommended levels of the current ASHRAE Standard 62-1989 and ASHRAE Standard 62-1981 to which the building was designed to conform. Several instances were observed when terminal units were completely shut off, thus eliminating the flow of outdoor air to as many as fifteen diffusers at a time. Measured values of air change effectiveness based on tracer gas decay measurements of local age of air were consistent with good mixing of the ventilation air in the occupied space.

KEY WORDS: age of air, airflow, building performance, commercial building, measurement, office building, tracer gas, ventilation, ventilation effectiveness

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INTRODUCTION

Outdoor air delivery is a critical function of mechanical ventilation systems in office buildings. Mechanical ventilation systems are designed with outdoor air intake specifications based on a building code or ventilation standard, for example ASHRAE Standard 62 "Ventilation for Acceptable Indoor Air Quality" [1]. The Ventilation Rate Procedure in this standard requires levels of outdoor air intake for a variety of indoor spaces. A minimum of 10 L/s (20 cfm) per person of outdoor air is required for office spaces. Verification of compliance to recommended and design levels of outdoor air intake is important for building operators and indoor air quality diagnosticians.

In order to verify that appropriate levels of outdoor air are being provided, measurement techniques are required which are straightforward, cost effective, and relatively easy to apply by building engineering practitioners of various skill levels [2]. Techniques demonstrated in a previous study by NIST to measure the total outdoor air intake rate of a large office building are reported in reference [3]. These techniques are useful to verify that a given amount of outdoor air is being provided to the entire building. However, other techniques are necessary to measure the delivery of outdoor air to the smaller zones of the building such as whole floors and individual offices and workstations. This report presents the results of a study in which these "local" outdoor air delivery rates were measured.

Measurements of local outdoor air delivery rates were performed in the Portland East Federal Building occupied by the Bonneville Power Administration in Portland, Oregon. The basic technique used in this study is to measure the supply airflow rate and the percentage of outdoor air intake to a specific zone of the building and multiply them together to obtain the outdoor airflow rate. This technique is referred to as the multiplicative method of determining outdoor air delivery. Another technique was demonstrated in this study, i.e., the use of the tracer gas decay technique to determine the local age of air within the occupied zones of the building. Measurement of the local age of air is currently more appropriate for research purposes as opposed to the more immediate and practical requirements of building engineers and investigators in the field. The applicability of local age of air measurements in the field is still being investigated, but it may provide a practical tool in the future.

BUILDING DESCRIPTION

Building Layout and HVAC Specifications

The Portland East Federal Office Building of the Bonneville Power Administration (BPA) was constructed as part of the General Services Administration project to build advanced technology office buildings in the 1980s. Construction of the BPA building was completed in 1987. The BPA building is a seven-story office building with a one-story basement and a two-story underground parking garage. A breezeway connects this building to another office building on the first floor, and a kitchen and dining room are also located on this level (floor plans are contained in reference [4]). Most of the building consists of open office space which is divided into individual workstations by partitions which are approximately 1.5 m (5 ft) tall. There are also several conference rooms and individual offices on each floor. The conditioned office space has a floor area of approximately 37,200 m^2 (400,000 ft^2) and a volume of 136,000 m^3 (4,800,000 ft^3), assuming an average ceiling height of 3.66 m (12 ft) including the return air plenum. The penthouse level has an approximate floor area of 4,000 m^2 (43,500 ft^2) and, based on a ceiling height of 6.4 m (21 ft), a volume of about 25,600 m^3 (905,000 ft^3).

A penthouse mechanical room houses the main HVAC systems which consist of three large variable air volume (VAV) systems, one serving the center of the building and the others serving the east and west sides. There are also several smaller air handling systems located on and serving parts of the B1 level. Sketches of the three main air handling systems are shown in reference [3]. All three of the main air handling systems are basically the same, with some differences in physical layout and control parameter set-points. Each system consists of two "cold" supply fans that work in parallel, one "hot" supply fan, a return fan, and a minimum outdoor air intake fan. The design supply air capacity of each system is approximately 47,200 L/s (100,000 cfm), and the minimum outdoor air intake fan capacity is 2,000 L/s (4,200 cfm) per system. Based on the building volume, the minimum design outdoor air intake rate translates to 0.16 air changes per hour (ach) or 0.16 L/s•m^2 (0.031 cfm/ft^2) of floor area, and the maximum supply airflow capacity is 3.7 ach or 3.8 L/s•m^2 (0.75 cfm/ft^2). An estimate of 2,000 building occupants yields minimum and maximum per person design outdoor air ventilation rates of 3 L/s (6.4 cfm) per person and 70 L/s (150 cfm) per person. This building was designed to comply with ASHRAE Standard 62-1981, which contained a minimum outdoor air intake requirement of 2.5 L/s (5 cfm) per person in office space with no smoking present [5]. This requirement corresponds to an air change rate of approximately 0.13 ach for this building based on 2,000 occupants and the gross building volume. ASHRAE Standard 62-1989 contains a minimum outdoor air requirement for office space of 10 L/s (20 cfm) per person [1] which corresponds to an air change rate of about 0.53 ach for this building. Alternatively, based on an occupant density of 7 people per 100 m^2 (1,000 ft^2) and a ceiling height of 3.66 m (12 ft), the ASHRAE Standard 62-1981 and 1989 recommended ventilation rates correspond to 0.17 and 0.69 ach respectively.

An economizer system controls the outdoor air intake rate through the cold supply fan system during mild weather by modulating the outdoor air intake (mixed-air) damper positions. During building occupancy, the minimum outdoor air fans run continuously to provide the design minimum of outdoor air, and the supply fans use variable-pitch fan blades to modulate the supply airflow rate based on the supply air demand of the occupied space. Supply air demand is controlled by VAV boxes, or terminal units, located above the suspended ceiling in the return air plenum. The terminal units modulate supply airflow rates to supply air diffusers depending on the temperature in the HVAC control zone being served by the terminal unit. The system includes both single-duct and dual-duct terminal units. Single-duct terminal units are served only by the cold fan system, whereas dual-duct units are served by both the hot and cold fan systems. Dual-duct units are generally located around the perimeter of the building. As more terminal unit dampers open, requiring additional supply airflow, the associated supply fan increases the airflow to maintain a supply static pressure set point in the main supply duct.

All of the terminal units served by the three main air handlers are non-induction units, which means that return air from the plenum is not mixed with the supply air prior to delivery to the space. Each terminal unit was factory calibrated to provide the supply airflow rate based on the velocity pressure measured at pressure taps on the terminal unit. A plot of airflow rate versus velocity pressure is provided on a label on the side of each terminal unit. The diffusers are linear slot diffusers which are approximately 0.9 m (3 ft) long and are spaced in a regular grid pattern in the ceiling. Between each row of diffusers is a row of return air slots which are staggered so as not to be located directly between two supply air diffusers.

Air Distribution System

This section describes the air distribution system of the HVAC system and introduces some terminology used in this report to describe various parts of the system. This terminology was developed in order to more easily refer to different sections of the air distribution system and the zones of the building with which they are associated. *Air distribution zone* refers to a volume of the building to which a given section of the air distribution system provides supply air. The air distribution zones do not necessarily correspond to HVAC control zones. The figures used to present this terminology refer to sections of the building which were investigated in this study.

A schematic of the air distribution system associated with the west cold supply fan system (SFC-5&6) is shown in Figure 1. This figure illustrates some of the terminology used to describe the air distribution system and provides design airflow capacities for selected sections of the system. Outdoor air and return air mix together in a large fan box which houses filters, cooling coils and the two supply air fans. Supply air is distributed from the fan box to three submain ducts referred to as A, B, and C. Submain A serves the entire west side of the seventh floor, and submains B and C serve the north and south sections of the west side of floors one through six. Floor branch ducts branch off of the submains on each floor of the building. The air distribution zones served by the fan boxes are referred to as the *air handler zones*, of which there are three in this building. The submain ducts serve smaller zones of the building, referred to as *submain zones*, which are subsets of the air handler zones. There are nine submain zones in this building, three in each air handler zone. The next smaller size zones are the *floor branch zones*. Each of these zones is served by a floor branch duct. There are six floor branch zones on each of floors 1 through 7, two from each air handler.

Figure 2 is a schematic of the sixth floor air distribution system associated with SFC-5&6, which consists of two floor branch zones referred to as 6-North and 6-South. Each floor branch serves a set of terminal units, and each terminal unit serves anywhere from 1 to 12 supply air diffusers. Figure 2 shows only a few of the terminal units and diffusers in the two floor branch zones. Floor branch zone 6-North consists of 5 terminal units and 38 diffusers, and zone 6-South consists of 15 terminal units and 101 diffusers. The air distribution zone associated with a terminal unit is referred to as a *terminal unit zone*. A *diffuser zone* is the building volume associated with a single diffuser. A *workstation zone* is the volume associated with the work area of a single building occupant. This could be a cubicle within the open office space or an individual office with floor-to-ceiling partitions. A workstation zone may be served by a single diffuser or a set of diffusers which serve no other workstation zones, as in the case of an enclosed office. Alternatively, within open office space, a workstation zone may be served by diffusers that also serve other workstation zones. Table 1 lists the different air distribution zones of the building and the components of the air distribution system which serve them.

Varying degrees of difficulty are encountered when establishing boundaries of air distribution zones, depending on the zone in question. The air distribution zone of a floor branch duct is a section of an entire floor and does not have distinct physical boundaries separating it from other floor branch zones. The boundaries of the terminal unit zones are defined by the diffusers associated with the terminal units, and these zones do not have distinct physical boundaries either. Based on the regular spacing of the diffusers, each diffuser can be associated with a specific amount of floor area, but there is no clear boundary to a diffuser zone. A workstation zone boundary is more easily defined because it does have a physical boundary; however, the specific diffusers that impact a workstation zone are not uniquely defined except in the case of an enclosed office.

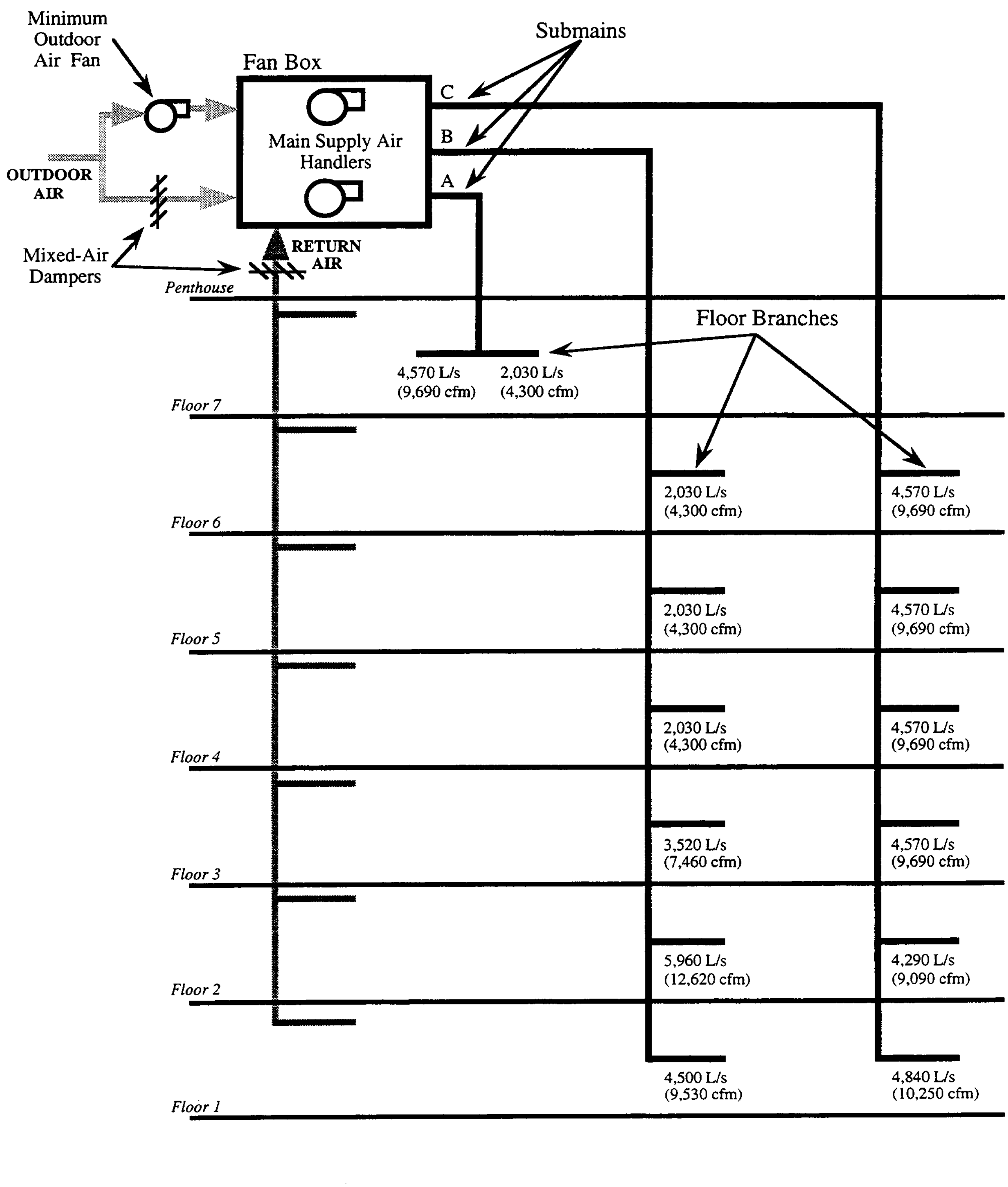

Figure 1: Schematic of West (SFC-5&6) Air Handling System

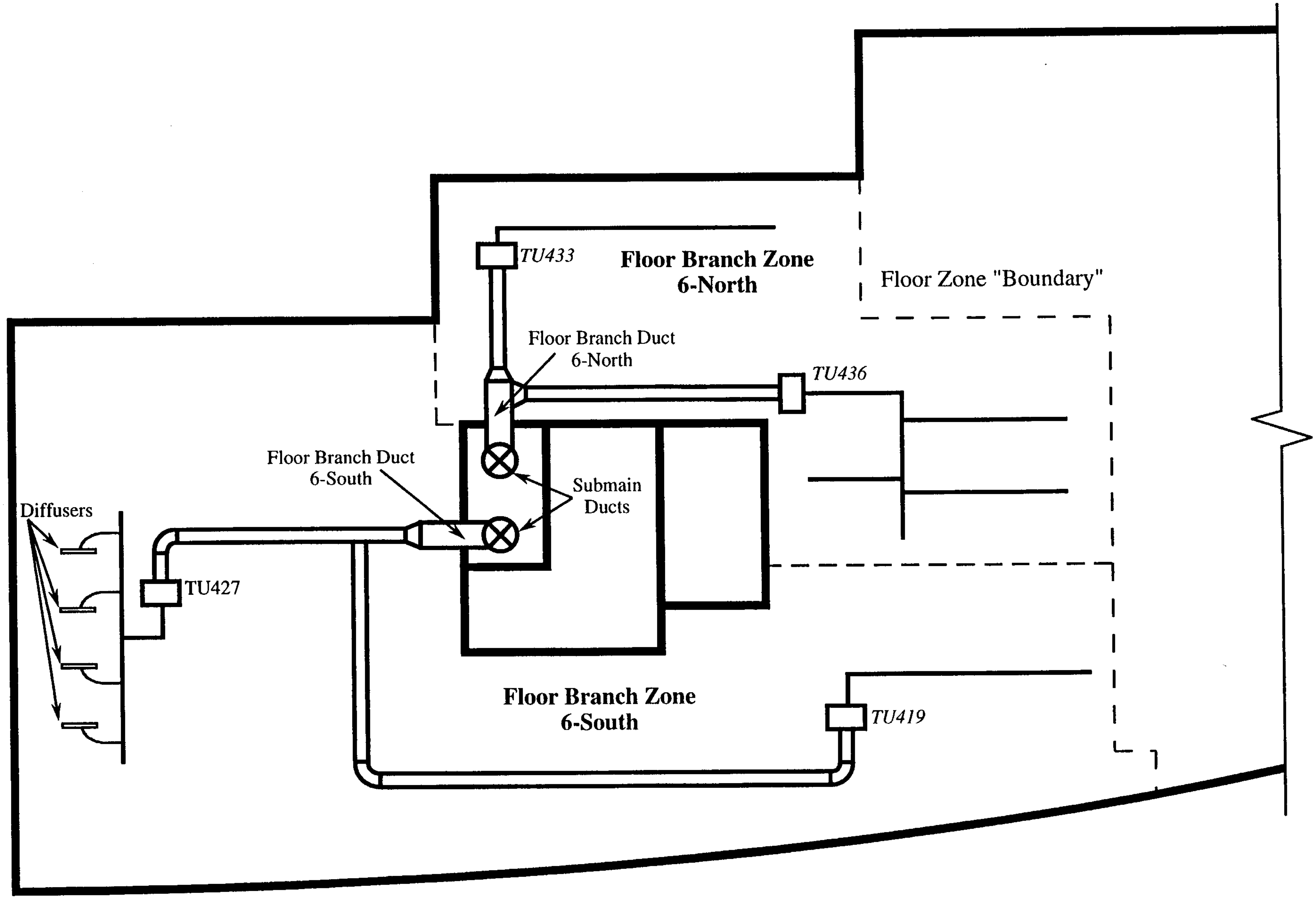

Figure 2: Schematic of Sixth Floor Air Distribution System

Air Distribution Zone	Air Distribution Component (Example)	Description
Air Handler Zone	Two supply air fans (SFC-5&6)	Three air handler zones Center, East, and West. Each zone consists of approximately one-third of floors 1 through 7.
Submain Zone	Submain duct (Submain B)	Two or three submain zones per air handler. Some zones consist of one third the floor area of the seventh floor, and others consist of about one sixth the total floor area of floors 1 through 6.
Floor Branch Zone	Floor branch duct (Floor Branch 6-North)	Two floor branch zones per air handler per floor. Each floor branch zone consists of about one sixth the floor area of a single floor.
Terminal Unit Zone	Terminal unit (TU418)	There are anywhere from 5 to 15 terminal units per floor branch. Zone boundaries are loosely defined by the diffusers served by the terminal unit.
Diffuser Zone	Supply air diffuser (TU418-2)	There are anywhere from 2 to 16 diffusers per terminal unit. Diffusers are spaced in a regular pattern in the ceiling, such that each diffuser "serves" a floor area of approximately 10 m^2 (108 ft^2).
Workstation Zone	Supply air diffuser(s) (TU418-2)	A workstation zone consist either of a cubicle of office system furniture or a wall-to-ceiling partitioned office space. There are about 300 workstations on floors 1 through 7, each with a floor area of approximately 6.7 m^2 (72 ft^2) or 10 m^2 (108 ft^2).

Table 1: Description of Air Distribution Zones of the Building

MEASUREMENT EQUIPMENT

This section describes the measurement devices and systems used in this study. Each device is described along with its associated measurement uncertainty and a brief description of how each device was used in this study. The uncertainties are either given by the manufacturer or determined by in-house calibrations.

Airflow Measurement

Airflow measurements were performed with three different devices: hot-wire anemometers, hooded velometers and a digital manometer. The hot-wire anemometers were used to perform velocity traverses in fan boxes, submains, floor branches, and terminal unit ducts. The anemometer calibrations are NIST traceable and have an uncertainty of 2.5% of the indicated reading. Hooded velometers were used to provide direct measurements of airflow rates from the supply air diffusers. The measurement range of the hooded velometer is 0 to 120 L/s (0 to 250 cfm) with an uncertainty of 3% of full scale. The digital manometer was used to measure the velocity pressure at terminal unit pressure taps. This velocity pressure was then converted to the airflow rate through the terminal unit using a conversion chart attached to each terminal unit. The manometer uncertainty is obtained by adding the following three values:

1% of the indicated reading

Resolution of the device (1Pa or 0.004 inch w.g.)

0.03% of span (5000 Pa or 20 inches w.g.) per °C difference of the ambient air from the reference temperature of 25°C.

The measurement uncertainties of these devices are based on the manufacturers' specifications.

Tracer Gas Measurement

Carbon dioxide (CO_2) and sulfur hexafluoride (SF_6) were used as tracer gases. Automated and manual sampling methods were used with both tracer gases. The automated sampling systems utilized a network of sampling tubes and a set of air sampling pumps which are centrally located in an indoor air quality and ventilation system diagnostic center in the building. The pumps draw air from up to twenty different sample locations through the sampling tubes and deliver the air to the CO_2 and SF_6 measurement equipment. This building diagnostic center and the associated measurement equipment are described in detail in reference [3].

Measurements of CO_2 concentrations were used to obtain percent outdoor air intake rates in the supply airstream. The automated CO_2 system continuously monitored the supply, return, and outdoor air CO_2 concentrations at the three main air handlers to determine the percent outdoor air. Portable CO_2 monitors were used to determine the CO_2 concentrations at other locations within the air distribution system. The automated CO_2 monitor has an uncertainty of 0.5% of full scale, which is 2500 ppm (parts per million), and the portable monitor has an uncertainty of 5.0% full scale over the range of 0 to 2000 ppm. Both of these measurement uncertainties are based on manufacturers' specifications. Calibrations of the CO_2 monitors were performed using zero gas and three calibration gases with concentrations of 350, 1029 and 2010 ppm.

SF_6 was used to measure local age of air. The automated system collected data at the main air handlers and other locations in the office space at which sample tubing was installed. Air sample bags and pumps were used to collect air samples at locations at which there were no automated sample locations. After the tests were completed, the air sample bags were analyzed using the detector employed by the automated system.

The SF_6 system employs a gas chromatograph equipped with an electron capture detector (GC-ECD). Calibration gases with concentrations of SF_6 ranging from 5 to 300 ppb were used to calibrate the GC-ECD. The natural logarithm of the frequency response of the detector is linearly related to the natural logarithm of the concentration of SF_6, and a regression analysis of the calibration data is used to determine this relationship. Based on the regression analysis of the calibrations, the uncertainty in SF_6 concentrations measured with this system is approximately 5% of the indicated concentration.

MEASUREMENT PROCEDURES

This section describes the measurement procedures used in this study and the uncertainty associated with each procedure. Procedures are presented for determining supply airflow rates, percent outdoor air intake rate, percent outdoor air uniformity, outdoor air delivery rate, and local age of air. An uncertainty analysis was performed for each procedure based on the uncertainties of the measurement devices and the propagation of uncertainty involved in performing calculations with the measured values. These uncertainties are reported in the section on measurement results as the *combined standard uncertainty* which is the estimated standard deviation of the result obtained by combining the standard uncertainties (estimated standard deviations) of each parameter used to determine the result as described in reference [6]. The standard uncertainties of measurement parameters are obtained from calibrations of measurement equipment against standard references and from accuracies quoted by equipment manufacturers.

It is important to make the distinction between measurement error and uncertainty. Measurement error or bias refers to the difference between the measured and the actual values, and uncertainty refers to the variation in results due to the repeatability of measurement equipment and the performance of calculations using the measured values. For example, an instrument may yield results which are very repeatable (low uncertainty), but the technique used to perform the measurement may cause improper readings of the desired value resulting in a large measurement error.

Supply Airflow Rate

Supply airflow rates at various locations throughout the air distribution system were measured using several different procedures. The procedures used to measure supply airflow rates are differentiated by the measurement technique used and the air distribution zone to which the supply air is delivered. In some cases different procedures can be used to measure the supply airflow rate to a given air distribution zone. Following are descriptions of measurement techniques and test protocols used to measure the supply airflow rate to specific air distribution zones of the building. Supply airflow rates were measured for air handler zones, submain zones, floor branch zones, terminal unit zones, workstation zones, and diffuser zones. Table 2 summarizes the airflow measurement procedures used for each of the air distribution zones.

Procedures

The measurement techniques used to determine supply airflow rates include velocity traverses of ducts, hooded velometer measurements of diffuser airflow rates, and velocity pressure measurements at terminal unit pressure taps. Velocity traverses were performed using a hot-wire anemometer to measure the air speed at multiple points in a cross-sectional area of a duct in order to determine the average velocity of air through the duct. ASHRAE Standard 111 [7] describes duct traverse techniques for measuring airflow rates in mechanical ventilation systems. An average velocity based on five individual velocity readings was determined at each traverse location. Upon completion of a velocity traverse, the average velocity in the duct was calculated and multiplied by the inside cross-sectional area of the duct at the traverse plane to obtain the volumetric airflow rate.

Airflow rates through terminal units were determined by measuring the velocity pressure at the pressure taps provided on the terminal unit by the manufacturer. This pressure was converted to a volumetric airflow rate using an equation determined from a conversion chart for each terminal unit. Each velocity pressure measurement consisted of an average of five readings made with a digital manometer. The velocity pressure to airflow rate conversion charts are based on factory calibrations, not on calibrations of the installed equipment. Therefore, the airflow rates determined from the terminal unit pressure taps are of unknown reliability and accuracy, which is not reflected in the combined uncertainty presented later. Airflow rates from supply air diffusers were measured using hooded velometers which were held up to a diffuser to obtain a single reading of volumetric airflow rate in cubic feet per minute (cfm), which was converted to liters per second (L/s).

There are three air handler zones in this building, Center, East and West, which are served respectively by the three main supply fan systems SFC-1&2, SFC-3&4 and SFC-5&6. Each air handler zone extends from the first to the seventh floor and contains approximately one third of the floor area on each floor. Figure 1 is a schematic of the air handler zone served by the West mechanical system. In this study, airflow measurements were performed on the West air handler zone using the three procedures in Table 2. All of the air handler zone measurements involved hot-wire traverses, but the traverses were performed in different locations. In the first procedure, traverses were performed inside the fan box downstream of the supply fans. In the second procedure traverses were performed in the three submain ducts and these airflow rates were added together. The third procedure consisted of measuring and adding together the airflow rates through the floor branches served by the air handler. The fan box measurements consisted of an 18 point traverse. Each submain measurement consisted of either a 10 or 20 point traverse, depending on whether the entire cross section or only a half cross section was traversed. Half traverses were sometimes used to decrease the measurement time in the submain ducts. Full cross section traverses were initially performed in all submains and compared to those of the half cross section traverses in order to verify the acceptability of the half traverses. Floor branch traverses included 16 to 32 points depending on the size of the duct.

Airflow rates to the submain zones of the West system were measured by a traverse of the individual submains and by summing the results of traverses performed in the floor branch ducts served by the submain. These measurements were performed on all three submains of the West air handler system. Submain A has two floor branch ducts on the seventh floor and submains B and C each have a single floor branch duct on each of floors one through six as illustrated in Figure 1.

Floor branch airflow rates were measured using two procedures. The first procedure was a traverse of the floor branch duct with at hot-wire anemometer. The second procedure was to measure the airflow rates of all the diffusers served by the floor branch duct and add them together.

Selected terminal unit zones were measured using three different techniques: hot-wire traverses of the ducts downstream from the terminal units, velocity pressure measurements converted to airflow rate using the terminal unit conversion charts, and summation of diffuser flow rates measured with a hooded velometer. Airflow rates to diffuser zones were measured with hooded velometers.

Supply airflow rates to individual workstation zones are difficult to define. The boundaries of the workstation zones are defined by the physical partitions between the workstations; however, individual diffusers are not uniquely associated with individual workstations in open plan areas. For this study, the supply airflow rate to a workstation was measured using two different procedures. One procedure involved measuring the airflow rate of the diffuser closest to the workstation. The other procedure used the average of the measured supply airflow rates of the four diffusers closest to the workstation. In both cases, the diffuser airflow rate is then converted to an estimate of the airflow rate per workstation (per person) by multiplying by the ratio of diffusers to workstations.

Air Distribution Zone	Measurement Procedures		
Air Handler	Hot-wire traverse inside fan box	Summation of hot-wire traverse of submain ducts	Summation of hot-wire traverse of floor branch ducts
Submain	Hot-wire traverse of submain duct	Summation of hot-wire traverse of floor branch ducts	
Floor Branch	Hot-wire traverse of floor branch duct	Summation of diffuser airflow rates measured with hooded velometer	
Terminal Unit	Hot-wire traverse of duct downstream from terminal unit	Velocity pressure measurement at terminal unit pressure taps converted to airflow	Summation of diffuser airflow rates measured with hooded velometer
Diffuser	Airflow rate measured with hooded velometer		
Workstation	Airflow rate of the diffuser closest to the workstation measured with a hooded velometer	Average airflow rate of the four diffusers closest to the workstation measured with a hooded velometer	

Table 2: Airflow Measurement Procedures

Uncertainty Analysis

Specific guidelines for minimizing measurement error and uncertainty in performing velocity traverses are given in ASHRAE Standard 111-1988 [7]. These guidelines contain recommendations for the selection of traverse locations which are as far away as possible from obstructions, changes in duct size, and flow redirection. It is difficult to avoid these regions of change in flow characteristics when performing airflow measurements in HVAC systems, and the standard specifies a criterion for the uniformity of the velocity distribution in the traverse plane. This criterion requires that at least 80% to 90% of the velocity measurements be greater than 10% of the maximum measured velocity (80%/10% rule). It is stated in ASHRAE Standard 111-1988 that this method of airflow rate measurement can yield results within 5% to 10% of the actual flow rate under good field conditions. However, if less suitable conditions exist for velocity traverses, then measurement errors can exceed 10%. In these situations uncertainty and sometimes error can be reduced by performing a more detailed traverse of the duct, i.e., increasing the number of traverse points.

A rigorous determination of the error associated with velocity traverse measurements of airflow rate in the field is not generally possible due to the wide variety of measurement configurations encountered in the field. In this study the 80%/10% rule was observed, and the combined uncertainty of a given measurement was determined based on the manufacturers' stated accuracy of the hot-wire anemometers and the propagation of uncertainty in performing the airflow rate calculations [6]. The propagation of uncertainty includes the calculations of the mean velocity and the multiplication by the cross sectional area, which has its own measurement uncertainty. This method of calculating measurement uncertainty is based on the assumption that only random measurement errors are occurring due to the accuracy of the measurement devices and does not account for systematic errors which may be occurring due to improper measurement techniques. Additional research is needed in order to better characterize errors involved in performing traverse measurements of airflow rates under field conditions.

Uncertainty in airflow rates measured using the flow hood and velocity pressure at the terminal unit devices are based on the accuracy of the measurement device used and the propagation of uncertainty based on the associated calculations. The terminal unit airflow rate measurements performed using the velocity pressure technique have uncertainties associated with the velocity pressure measurement made with the digital manometer and the use of the velocity pressure to airflow conversion based on the conversion chart of the terminal unit. Uncertainty in the diffuser airflow rate measurements are due solely to the uncertainty of the hooded velometers which were calibrated by the manufacturer just prior to this study. The uncertainty in the summation of diffuser airflow measurements is based on the propagation of uncertainty in performing the summation.

Percent Outdoor Air Intake Rate

Ideally, the outdoor air intake rate of an air handler can be measured directly by velocity or velocity pressure traverses in the outdoor air intake ducts. However, these locations are seldom configured such that traverse techniques are appropriate. When the outdoor air intake duct can not be traversed, or when one needs to determine the outdoor air delivery to an air distribution zone without a dedicated air handler, one must employ an indirect means of determining the outdoor airflow rate. One such procedure, referred to as the multiplicative method, involves the multiplication of the measured supply airflow rate by the measured percent outdoor air intake in the supply airstream. This method only accounts for outdoor air delivered by the air handler and does not account for infiltration of outdoor air through the building envelope.

The percentage of outdoor air in the supply airstream should be the same from the supply fan to the diffusers, except in induction systems, which employ terminal units that mix return air from the ceiling plenum or the ventilated space with the supply air from the air handler. Even in non-induction systems, conditions could conceivably exist that cause the outdoor air percentage to vary within the air distribution system. If the outdoor and return air do not mix well within the air handler before the supply airstream splits off into the submain ducts, then the outdoor air percentage could be different within each submain [8]. In addition, if return air is induced into the supply air ductwork through leaks in the system, then the outdoor air percentage downstream from the leak will be lower than the value at the air handler. Ideally, to use the multiplicative method of determining local outdoor air delivery rates, one would measure the percent outdoor air intake at all locations of interest within the air distribution system. However, this process is much more time consuming than making a single measurement at the air handler.

This section describes the measurement of percent outdoor air intake at the air handler and at other points in the air distribution system, including how these techniques were employed in this study. The adjustment of the percent outdoor air intake measured at the air handler to account for induction of return air at an induction unit is also described, even though this adjustment was not required in this study. This adjustment can also be used to account for unintentional return air induction into the supply airstream.

Procedures

A mass balance of air and tracer gas at the air handlers was employed to measure percent outdoor air intake. Carbon dioxide was used as the tracer gas, and the automated CO_2 measurement system was used to monitor concentrations in the outdoor, return, and supply airstreams of the main air handlers. Supply air concentrations were also measured inside fan boxes, submain ducts, and floor branch ducts with portable CO_2 monitors. The equation for determining percent outdoor air using a tracer gas is as follows:

$$\%OA = 100 \times \frac{C_R - C_S}{C_R - C_O} \qquad \text{Equation 1}$$

where C_R is the return air tracer gas concentration, C_O is the concentration in the outdoor air, and C_S is the concentration in the supply air.

In cases where there is induction of return air into the supply airstream, the percent outdoor air intake downstream of the induction site is determined as follows. The method presented here is for an induction type terminal unit but can be applied to any induction site. In addition to measuring the three tracer gas concentrations at the air handler to determine the percent outdoor air intake using Equation 1, the concentrations are also measured at the supply air inlet to the terminal unit, in the return air plenum near the terminal unit, and at the outlet of the terminal unit. Based on these concentrations, the percentage of outdoor air in the terminal unit outlet is determined by the following equation:

$$\%OA = 100 \times \left(\frac{C_R - C_S}{C_R - C_O}\right)\left(\frac{C_{PL} - C_{TU}}{C_{PL} - C_S}\right) \qquad \text{Equation 2}$$

where C_{PL} is the return air plenum concentration near the terminal unit, and C_{TU} is the concentration at the outlet of the terminal unit. Equation 2 is based on the assumption that the concentration at the inlet to the terminal unit is the same as that at the outlet of the supply air fan.

Uncertainty Analysis

Equation 3 is the combined standard uncertainty associated with percent outdoor air calculations using Equation 1. The uncertainty $\Delta\%OA$ is based on the uncertainty in the measured concentrations and the propagation of uncertainty in performing the calculations. The uncertainties in concentrations are indicated by the symbol Δ in front of the concentration to which the uncertainty applies, e.g., ΔC_R is the uncertainty in the return air concentration.

$$\Delta\%OA = 100 \times \left[\frac{\Delta C_R^2 + \Delta C_S^2}{(C_R - C_O)^2} + \frac{(C_R - C_S)^2(\Delta C_R^2 + \Delta C_O^2)}{(C_R - C_O)^4}\right]^{1/2} \qquad \text{Equation 3}$$

An equation similar to Equation 3 can be developed for calculating the uncertainty in the percent outdoor air downstream of return air induction locations given by Equation 2. Note that the uncertainty calculated with Equation 3 is in units of percent outdoor air.

The magnitude of the difference between the return and outdoor air concentrations is the main factor affecting the uncertainty in the percent outdoor air measurements. The uncertainty in the calculation of percent outdoor air intake will be larger for smaller differences between the return and outdoor air concentrations. To minimize uncertainties when using CO_2 as a tracer gas, measurements should be performed well into the occupied period of the day when the indoor CO_2 concentration has built up well above the outdoor air concentration.

The combined uncertainty can be reduced by performing multiple percent outdoor air measurements with a fixed mixed-air damper setting and averaging the results. In this study, the average and standard deviation of the measured percent outdoor air intake rate was calculated for a number of individual measurements taken over several hours (generally from 2 to 6 hours). During these measurements, the difference between the return and outdoor air concentrations changed significantly due to fluctuations in the indoor CO_2 concentration. However, the calculated percent outdoor air intake rate was fairly constant. Based on the propagation of uncertainty [6], the combined standard uncertainty of these averages is much less than the values obtained for individual readings based on Equation 3. The combined uncertainties of the individual measurements, given by Equation 3, ranged from 20% to 40% of the measured percent outdoor air intake rate. However, the combined uncertainties based on the average of a set of measurements ranged from 4% to 11% of the average value depending on the number of measurements included in the average and the average value.

Percent Outdoor Air Uniformity

The percentage of outdoor air can be assumed to be uniform throughout a supply air distribution system without induction type terminal units. In this study, measurements were made to verify that the percent outdoor air intake was indeed uniform. The uniformity of percent outdoor air in the distribution system was measured to support the measurements of outdoor air delivery rates to the various air distribution zones of the building. Measurements of percent outdoor air uniformity were also used to assess the mixing of outdoor air and return air within the supply airstream. If the outdoor air is well mixed within the supply air stream at the air handlers prior to reaching the submain ducts, and there is no induction of return air into the supply air system, then a single measurement of percent outdoor air at the air handler will suffice for the entire system. The use of a single value for percent outdoor air measured at a main air handler would simplify the measurement of outdoor air delivery rates throughout the air distribution system. The existence of induction requires local measurements of percent outdoor air intake rates to determine outdoor airflow rates. Because every air handler configuration is unique, such an assessment of percent outdoor air uniformity should be performed in any building where the multiplicative method is being applied.

<u>Procedures</u>

Carbon dioxide was used as a tracer gas to verify percent outdoor air uniformity within the West air distribution system. The automated CO_2 system was used to continuously monitor the supply, return and outdoor air concentrations at the main air handler, and portable monitors were used to measure the supply air concentration at various points in the air distribution system downstream of the air handler. Values of the percent outdoor air intake were then calculated at the locations at which the local supply air concentrations were measured using Equation 1, and these values were compared to the percent outdoor air intake at the air handler.

Supply air tracer gas concentration, and therefore percent outdoor air intake, were measured inside the fan box, submain ducts, and floor branch ducts. Fifteen-point concentration traverses were performed in much the same way as were velocity traverses in the fan boxes. Five-point traverses were performed in the submain ducts, and three-point traverses were made in the floor branch ducts.

<u>Uncertainty Analysis</u>

The uncertainty in performing the percent outdoor air uniformity analysis is the same as presented previously for the measurement of percent outdoor air intake rates.

Outdoor Air Delivery Rate

Outdoor air delivery rates to the different air distribution zones were measured using the multiplicative method. In this method the outdoor airflow rate at some point in the air distribution system is obtained by multiplying the supply airflow rate at that point by the percentage of outdoor air in the supply air. In order to compare these results to recommended and design ventilation rates, these outdoor air intake rates are converted to airflow rates per person.

Procedures

The techniques used to measure supply airflow rates and percent outdoor air intake rates are discussed earlier. During measurements of outdoor air intake rates, the automatic control of the mixed-air dampers was overridden to maintain the dampers in a fixed position. The supply airflow rates did modulate to maintain thermal comfort within the building. However, the measured values of percent outdoor air intake at the air handler showed little variation during the outdoor airflow rate measurements.

The air distribution zones for which outdoor air intake rates are reported include the air handler zones, floor branch zones, terminal unit zones and workstation zones. Air handler, submain, and floor branch outdoor airflow measurements were performed on the West air handler system (SFC-5&6). Terminal unit and workstation zone outdoor airflow measurements were performed in floor branch zones 6-North and 6-South, served by the West air handler system.

The comparison of the measured values of the outdoor airflow rates to ventilation standards requires an association between the air distribution zones for which the outdoor air delivery is measured and the number of occupants in the zone. The number of occupants of the air handler zones was based on the total number of building occupants, about 2000, divided by three for each of the three air handler zones. The number of occupants for the submain and floor branch zones was determined from the floor area of the zones and the average occupant density in the three air handler zones of 6 people per 100 m^2. The number of occupants for floor branch, terminal unit and workstations zones was determined by counting the number of workstations in the appropriate zone.

Uncertainty Analysis

The combined standard uncertainty in calculating outdoor airflow rates, ΔQ_{OA}, is based on the propagation of uncertainty in multiplying the values of supply airflow rate, Q_S, by the percent outdoor air intake rate, %OA, as shown in Equation 4. The combined standard uncertainty in the outdoor airflow rate will depend on the uncertainties in the techniques used to measure the supply airflow rate and on the uncertainty in the percent outdoor air intake rate.

$$\Delta Q_{OA} = \pm\left[\left(Q_S \times \frac{\Delta\%OA}{100}\right)^2 + \left(\frac{\%OA}{100} \times \Delta Q_S\right)^2\right]^{1/2} \qquad \text{Equation 4}$$

Local Age of Air

The local age-of-air is related to the rate of outdoor air delivery to a specific location in a ventilated space and can be used to determine the air change effectiveness of the ventilation system at a given location. Air change effectiveness is a measure of the uniformity of outdoor air delivery and mixing in a space [9, 10, and 11]. The local age of air at a specific location is defined as the average amount of time that has elapsed since the air molecules at that location entered the building and is denoted by τ_i. The inverse of the building air change rate is referred to as the nominal time constant of the building, τ_n. The local air change effectiveness characterizes the ventilation effectiveness at a specific location and is defined as

$$\varepsilon_i = \tau_n / \tau_i \qquad \text{Equation 5}$$

If the air within a space is perfectly mixed, then the local age of air will be the same throughout the space and equal to τ_n. The local air change effectiveness, ε_i, at all locations within the space will therefore equal one. If there is non-uniform air distribution within a space, those locations with poor ventilation air distribution will have local ages of air that are higher than the space average. Locations in these so-called "stagnant" regions will have values of τ_i that are relatively large and values of ε_i significantly less than one, a generally undesirable situation. More information on the measurement of local age of air and its application in mechanically ventilated buildings can be found in references 12 and 13.

When measuring the local age of air at a particular location, the results are influenced by the uniformity of the distribution of outdoor air to that space, the mixing of air within the space, the infiltration of outdoor air directly to the space, and airflow from adjoining spaces. As pointed out by Fisk [14], comparing the value of τ_i to the nominal time constant of the building, as in the definition of ε_i in Equation 5, does not enable one to distinguish between the effects of mixing of air within the space and outdoor air delivery to the space itself. In order to assess within-space mixing, Fisk proposes comparing the value of τ_i to the age of air measured at the return vent(s) serving the space, τ_r. The values of the age of air at return vents and within other return airstreams can also be used to assess the uniformity of outdoor air distribution within the building.

Procedures

Local age of air was measured with the tracer gas decay technique using SF_6 as the tracer gas. This technique involves establishing initial conditions of a uniform tracer gas concentration throughout the building and then monitoring the decay in tracer gas concentration at each measurement location. The procedure used in these tests was to inject tracer gas at a constant rate until a uniform equilibrium concentration is achieved throughout the building. Depending on the air change rate of the building, it takes several hours for the indoor tracer gas concentration to reach equilibrium. If the outdoor airflow rate into the building is constant, then it will take three time constants, τ_n, to reach 95% of the equilibrium concentration and four time constants to reach 98% of equilibrium. The nominal time constant of the building, τ_n, is equal to the inverse of the air change rate Q/V. Where Q is the outdoor airflow rate into the building, and V is the building volume. The nominal time constant is determined from the equilibrium concentration of the tracer gas as follows:

$$\tau_n = C_{eq}(V/q) \qquad \text{Equation 6}$$

where

C_{eq} is the equilibrium concentration,
V is the building volume, and
q is the tracer gas injection rate.

During the tests, the HVAC system controls were adjusted to maintain a constant outdoor air damper position and all other controls (e.g. supply airflow rate modulation) operated normally. During the injection, the tracer gas concentrations in the return airstreams of each of the main air handlers (Central, East and West) and about 12 locations within the occupied space were monitored every ten minutes with the automated SF_6 monitoring system. The goal of the injection procedure was to maintain tracer gas concentrations in the three main returns within 10% of their mean value for at least one hour. In most of the tests, the establishment of these conditions required about six hours.

Once a uniform tracer gas concentration was attained in the building, the tracer gas injection was stopped, and air sampling for tracer gas decay analysis was conducted at selected points in the building and the ventilation system. The age of air at a given location was then determined from the following equation:

$$\tau_i = \frac{1}{C_{i,0}} \int_0^{\infty} C_i(t)\,dt \qquad \text{Equation 7}$$

where

$C_i(t)$ is the tracer gas concentration at location i and time t, and
$C_{i,0}$ is the concentration at t=0.

During these tests, tracer gas concentrations were measured and the age of air was determined at locations that were sampled by the tracer gas system and at locations that were sampled manually. The automated sample locations of the occupied space were located at breathing level at selected workstations throughout the building. Two locations in each of the three air handler zones of the sixth floor were monitored along with one location on each of floors one through five and one location on floor seven in the West air handler zone. Air sample bags and portable air sample pumps were also used to collect samples manually at selected workstations and the return air vents located in the 6-North and 6-South floor branch zones.

The integral in Equation 7 was determined for the different sampling approaches as follows. For the automated sample locations, numerical integration of the concentrations measured at ten minute intervals were performed. Two methods of manual sampling were used to determine the integral. One method was to collect periodic grab samples at approximately twenty minute intervals during the decay to be numerically integrated. The other method was to fill a bag at a constant rate during the entire decay period. The concentration of tracer gas inside the bag is then the average concentration during the decay, which when multiplied by the duration of the sampling interval is equal to the integral in Equation 7. For both bag sampling methods, $C_{i,0}$ was obtained using a grab sample just prior to shutting off the tracer gas. Details for determining the solution to Equation 7 are given in [15].

To determine air change effectiveness, the local age of air, τ_i, is typically compared to the nominal time constant of the building, τ_n (see Equation 5). However, when local ages of air are referenced to the nominal time constant for the whole building, one can not distinguish between the effects of nonuniform outdoor air delivery to different building zones and nonuniform mixing of ventilation air within the zone. As proposed by Fisk [14], the local age of air, τ_i, was referenced to the local age of air in the return airstreams associated with the measurement location, referred to as τ_r. The local age of air in the return airstreams were measured at the main air handler serving the space and in some cases in the return vents located closest to the space location at which τ_i was measured. Samples at the return air handler were collected using the automated sampling system, and air sample pumps and bags were used to collect samples at the return air vents.

The values of the local age of air were also referenced to a so-called local nominal time constant, $\tau_{n,local}$. The value of $\tau_{n,local}$ was determined by selecting a single diffuser or group of diffusers associated with a location at which τ_i was measured. Based on the outdoor airflow rate measured at these diffusers using the multiplicative method, $\tau_{n,local}$ was determined according to the following equation.

$$\tau_{n,\,local} = \frac{N \times A_d \times \text{CeilingHeight}}{\Sigma Q_{Diffusers} \times \frac{\%OA}{100}} \qquad \text{Equation 8}$$

where N is the number of diffusers, A_d is the floor area per diffuser, $\Sigma Q_{Diffusers}$ is the total supply airflow rate through the diffusers and %OA is the percent outdoor air intake rate. The value of $\tau_{n,local}$ is approximate based on the use of a single value of A_d for all measurement locations. The local time constant is a measure of the outdoor air delivered by the ventilation system and does not account for outdoor air from other zones of the building or infiltration of outdoor air through the building envelope.

Uncertainty Analysis

The uncertainty in calculating the local age of air using Equation 7 is given by the following equation.

$$\Delta\tau_i = \tau_i \left\{ \left[\frac{\Delta\int Cdt}{\int Cdt}\right]^2 + \left[\frac{\Delta C_{i,0}}{C_{i,0}}\right]^2 \right\}^{1/2} \qquad \text{Equation 9}$$

$\Delta\int Cdt$ is the uncertainty in the integral term of Equation 7, and $\Delta C_{i,0}$ is the uncertainty in the initial concentration, $C_{i,0}$. The uncertainty in the initial concentration is based on the accuracy of the SF_6 measurement system given earlier (5% of reading). The uncertainty in the integral calculation depends on the method used to determine the integral. As mentioned previously, three different methods were used to determine the integral term of Equation 7. Numerical integration techniques were used for measurements performed using the automated sample locations and the periodic bag samples. When using these techniques the uncertainty in the integral term is associated with the integral approximation calculations and the uncertainty in the measured concentrations. The other method involved the measurement of the average concentration during the integral period using a sample bag. In using this method the uncertainty is due to the accuracy of the SF_6 measurement system used to determine the average concentration.

The uncertainty in calculating the local nominal time constants is given by the following equation.

$$\Delta\tau_{n,\,local} = \tau_{n,\,local}\left\{\left[\frac{\Delta\%OA}{\%OA}\right]^2 + \left[\frac{\Delta\Sigma Q_{Diffusers}}{\Sigma Q_{Diffusers}}\right]^2\right\}^{1/2} \qquad \text{Equation 10}$$

$\Delta\%OA$ is the uncertainty in the percent outdoor air intake rate, and $\Delta\Sigma Q_{Diffusers}$ is the uncertainty in the measurement of the total supply airflow rate through the N diffusers. Equation 10 neglects any uncertainty associated with the values of A_d and the ceiling height.

RESULTS AND DISCUSSION

This section presents the results of the measurements discussed in the previous section. Because this building has variable air volume systems, it is important to note that these measurements only reflect the conditions at the time of the measurements. However, all of the measurements were made during normal occupied hours with no unusual activities or thermal loads in the space. The most detailed measurements were made on three days: 1/13/93, 2/24/93 and 2/25/93. On these days, the measurements were made between about 8 a.m. and 6 p.m.

Supply Airflow Rate

Supply airflow rates were measured in order to compare actual supply airflow rates with design airflow rates and for use in determining outdoor airflow rates to the air distribution zones of the building using the multiplicative method. The results of the supply airflow measurements performed for this study are presented in order of decreasing size of the air distribution zone, i.e., air handler zone, submain zone, floor branch zone, terminal unit zone and workstation zone.

Air Handler Zones

The results of six separate measurements of supply airflow rates to air handler zones are presented in Table 3. All of the measurements were performed on the West air handler zone (SFC-5&6) except for one set of measurements which was performed on the East air handler zone (SFC-3&4). Supply airflow rates to air handler zones were measured by hot-wire traverses inside the fan box, submains, and floor branches. The combined standard uncertainties associated with the fan box, submain, and floor branch traverse techniques are approximately 4%, 5% and 2% of the measured values, respectively. Fan box traverses yield the supply airflow rate to the air handler zone directly. Air handler zone supply airflow rates based on submain and floor branch traverses require the summation of the individual submain or floor branch airflow rates. Airflow rates measured in the fan box were consistently higher than those obtained by the summation of the submain or floor branch airflow rates. For each set of measurements the percent difference of each value from the average was less than twenty percent.

Date	Supply Airflow Rate [L/s]			
	System	Fan Box	Σ Submains	Σ Floor Branches
11/18/92	SFC5&6	28,800	19,600	-
11/19/92	SFC5&6	27,100	21,500	25,100
12/08/92	SFC5&6	25,000	20,600	-
	SFC3&4	32,700	23,800	-
12/09/92	SFC5&6	26,400	21,600	23,800
12/10/92	SFC5&6	30,800	23,100	23,400

Table 3: Supply Airflow Rates to Air Handler Zones

The discrepancies between the fan box traverses and the submain measurements may be due to air leakage between the two measurement locations or to systematic measurement errors. If the difference was due to leakage between the fan box and submain measurement locations, then the leakage rates would be between twenty and thirty percent of the airflow rate measured inside the fan box. One possible source of systematic error is the sparseness of the traverses performed inside of the fan boxes. Because the cross sectional area inside the fan boxes is so large, 7.9 m by 3.35 m (25.9 ft by 11.0 ft), it was impractical to perform a traverse using equal areas whose centers are a maximum of 15 cm (6 inches) apart as recommended by ASHRAE [7]. Another potential source of systematic error in the fan box measurements is uncertainty in the cross sectional area of the fan box. The section in which the fan box traverses were performed is a region of significant transition in the airflow. Fan coils and bypass dampers are upstream of the measurement site, and a set of flow straightening devices, with a significantly smaller cross sectional area than the coils, is located downstream. While this region of the fan box is the only accessible practical location to measure the total supply airflow rate, it is far from ideal. The obstructions to the airflow pattern both upstream and downstream of the measurement location and the short distance between the change in cross sectional area make it difficult to define the effective cross sectional area.

Submain Zones

Supply airflow rates to submain zones were measured by direct traverses of the submain ducts and by the summation of the airflow rates through the floor branch ducts which are served by the submains. Results of these measurements are presented in Table 4. These measurements were performed on the submain zones of the West air handler (SFC-5&6), denoted as A, B, and C. The combined uncertainties associated with the submain traverses and floor branch airflow summations are about 10% and 5% of the measured flow rate, respectively. In most cases the flow rate obtained by the summation of the floor branches is about 10% to 20% higher than was obtained by a direct traverse of the submains. This suggests a systematic measurement error as opposed to the leakage of air from the system between the two measurement locations. Leakage would cause the summation of the floor branch measurements to be less than the submain values. These measurements compare more favorably to each other than do the air handler zone measurements. This may be due to the fact that the traverse pattern conformed more closely to the recommended guidelines for performing airflow measurements and the cross sectional areas of the ducts are more accurately known than those of the fan box.

Date	Supply Airflow Rate [L/s]		
	Submains		Σ Floor Branches
11/19/92	A	2,970	3,180
	B	7,440	9,080
	C	11,100	12,300
12/09/92	A	2,890	3,170
	B	7,120	8,710
	C	11,500	11,900
12/10/92	A	3,460	3,250
	B	8,950	9,080
	C	10,700	12,100

Table 4: Supply Airflow Rates to Submain Zones

Floor Branch Zones

Supply airflow rates to the two floor branch zones on the west side of the sixth floor were measured by direct traverses and by the summation of the airflow rates through the diffusers of each floor branch. As indicated previously, these two floor branch ducts are called 6-North and 6-South. Results of these measurements are presented in Table 5. The combined uncertainties for the floor branch traverses and the summation of the diffuser airflow rates are 8% and 6% of the measured values, respectively. For the five cases in which both measurements were made, the sum of the diffuser airflow rates are 10% to 15% lower than the floor branch measurements. The direction of the difference is consistent with duct leakage, but a systematic measurement error can not be ruled out.

Date	Supply Airflow Rate [L/s]		
	Duct	Floor Branch	Σ Diffusers
12/9/92	6-North	1,110	--
	6-South	2,560	--
12/10/92	6-North	1,160	--
	6-South	2,550	--
1/13/93	6-North	1,550	--
1/14/93	6-North	890	--
	6-South	2,990	2,710
2/24/93	6-North	1,690	1,500
	6-South	3,730	3,140
2/25/93	6-North	1,450	1,320
	6-South	3,910	3,280

Table 5: Supply Airflow Rates to Floor Branch Zones

Terminal Unit Zones

Terminal unit zone supply airflow rates were measured using three different methods for selected terminal units located within the 6-North and 6-South floor branch zones. For each set of measurements a hot-wire traverse was performed in the duct entering the terminal unit along with a summation of diffuser flow rates. In some cases the velocity pressure was measured at the pressure taps of the terminal units and converted to an airflow rate using the conversion chart on the side of each terminal unit. The results of these measurements are presented in Table 6. The combined uncertainties associated with the three measurement techniques: duct traverse, velocity pressure measurement and summation of diffuser airflow rates, are 11% to 14%, 4% to 8% and 3% to 70% of the measured flow rate, respectively. Uncertainties based on the summation of diffuser airflow rates were typically between 3 and 15% of the measured values; however, larger uncertainties are associated with airflow rates that are at or below the measurement uncertainty associated with the hooded velometer, i.e. 3.5 L/s (7.5 cfm).

The airflow rates measured by duct traverses were generally higher than the sum of the diffuser flows; in three cases the difference was more than 25% of the duct traverse value. In one case the diffuser result was about 70% greater than the duct traverse, but the airflow rate was very low. The fact that duct traverse results are generally greater than the sum of the diffuser flow rates is consistent with duct leakage, but there is not enough information to confirm this explanation. Changes in airflow rate during the measurements due to terminal unit damper modulation could also affect the similarity of the results. The results based on the velocity pressure measurements at the terminal unit pressure taps produced consistently lower results than with the other two methods. This could be due to the development of conversion charts based on a laboratory configuration and not as installed in this building.

Date	Terminal Unit	Supply Airflow Rate [L/s]		
		Duct Traverse	Velocity Pressure	Σ Diffusers
2/24/93	TU436	426	313	348
		428	327	369
	TU433	598	-	325
	TU427	29	-	49
2/25/93	TU436	527	311	382
	TU419	265	182	243
		237	192	245

Table 6: Supply Airflow Rates to Terminal Unit Zones

On three separate occasions, the airflow rates through all of the diffusers serving the west side of the sixth floor were measured. The airflow rates of the diffusers serving each terminal unit (with the exception of terminal unit 437 in the first data set) were added together to obtain the supply airflow rate through the terminal units. The results of these diffuser flow summations are shown in Figure 3 with the corresponding design supply airflow rates. In most cases, the measured supply airflow rate at the terminal unit was greater than 70% of the design values, presumably a function of the thermal loads in the space. The actual supply airflow rate through a terminal unit is a function of the thermal loads in the space served by the terminal unit. Depending on the thermal loads, the supply airflow rate will generally be below the design value. There were several instances when the terminal unit airflow rates were less than 20% of their design capacities, and sometimes there was no airflow detected through them at all. There were also several instances when the measured airflow rate was greater than the design capacity. Measured supply airflow rates were less than 20% of the design capacity on at least one occasion at terminal units 420, 426, 427, 428, 434, 435, and 437. Even though these supply airflow rates may be sufficient to satisfy the space conditioning requirements, they will reduce the delivery of outdoor air to the individual occupants of the space. These measurements are only indicative of airflow rates at the time that the measurements were performed and only under the conditions which existed during the performance of the measurements.

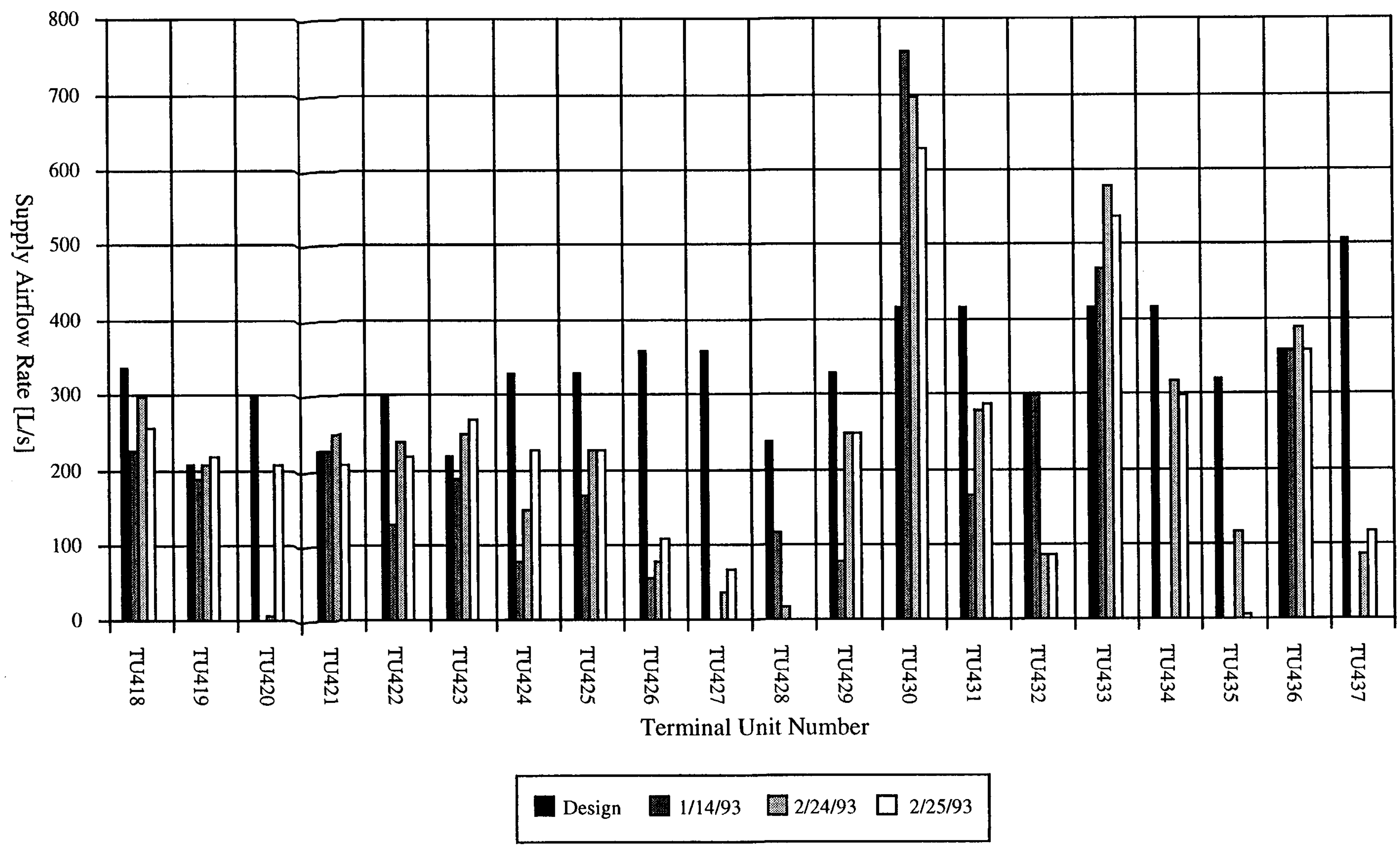

Figure 3: Supply Airflow Rates through Terminal Units on the West Side of the Sixth Floor

Workstation Zones

Supply airflow rates to individual workstation zones were estimated based on airflow rate measurements at individual diffusers. These measurements were made for all ninety-five workstations on the west side of the sixth floor, i.e., within floor branch zones 6-North and 6-South. These measurements were made on three days, and the results for one day are presented in Table 7. Table 7 shows the supply airflow rates to each workstation zone indicated by the terminal unit number and diffuser number nearest to the workstation, e.g., 418-2. Four airflow rates are given for each workstation. The first two columns, labeled Closest Diffuser, are the measured and design supply airflow rates of the diffuser closest to the workstation. These values are based on the measured and design airflow rates multiplied by the ratio of the number of diffusers serving workstations on the west side of the sixth floor to the number of workstations, i.e., 1.37. The use of this multiplier converts the diffuser airflow rates into an estimate of the supply airflow rate per workstation. The last two columns in Table 7, labeled Average of Four Diffusers, are the average of the measured and design airflow rates of the four diffusers closest to the workstation. These values are also adjusted based on the ratio of diffusers to workstations. The average value is calculated and presented because this is an open office space and the supply airflow rate to an individual workstation is impacted by more than just the airflow rate out of the closest diffuser. The average value is used to obtain a better estimate of the actual supply airflow rate to the workstation. The combined uncertainties associated with individual diffuser measurements are based solely on the measurement uncertainty associated with the hooded velometer. When the diffuser airflow rates are greater than 3.5 L/s (7.5 cfm), the uncertainty associated with the hooded velometer, the supply airflow uncertainties are about 5 to 25% of their measured values.

There are several cases in Table 7 for which the supply airflow rate of the closet diffuser is close to or equal to zero. The number of such cases is much less for the average of the four closest diffusers. Table 8 summarizes the results of the workstation supply airflow rate measurements for the three days of testing. The table shows the number of workstations with measured supply airflow rates equal to zero, greater than 0 and less than 10% of the design supply airflow rate, between 10% and 25% of design, between 25% and 50% of design, between 50% and 100% of design, and greater than the design supply airflow rate. These results are given for the workstation supply airflow rate based on the closest diffuser and the average of the four closest diffusers for all three days of testing. The results in Table 8 show several workstations have zero or very low supply airflow rates during the measurements. For the three days, 20 to 30% of the 95 workstations had supply airflow rates from 0 to 25% of their design values based on the closest diffuser. Based on the average of the four closest diffusers, between 10 and 20% of the workstation supply airflow rates were from 0 to 25% of the corresponding design values. The design values in the tables are based on the maximum supply airflow rates through the diffusers. Because this is a variable air volume system and the actual supply airflow rates depend on the thermal loads in the space, it is not unexpected that the measured airflow rates will be below the design values. However, the existence of very low supply airflow rates indicates that little or no supply air, or outdoor air, is being delivered by these diffusers despite the fact that these workstations were occupied during these measurements. It is important to note that these measurements only provide information on the supply airflow rate to the workstation at the time of the measurement.

Workstation	Supply Airflow Rate [L/s]			
	Closest Diffuser		Average of Four Diffusers	
	Measured	Design	Measured	Design
418-2	66	94	59	80
418-3	67	94	57	82
418-4	61	94	55	81
419-1	39	39	39	40
419-2a	40	39	49	48
419-2b	40	39	49	48
419-3a	37	39	41	41
419-3b	37	39	37	40
419-7a	39	45	33	65
419-7b	39	45	51	70
420-3	0	84	0	84
420-4	0	84	11	72
420-5	0	78	19	65
421-1a	6	39	34	37
421-1b	6	39	36	39
421-2	48	39	36	39
421-5	42	39	40	40
421-6	39	39	40	39
421-7	45	42	21	63
421-8	39	42	27	48
422-1	32	58	25	63
422-2	32	58	25	58
422-6	29	58	29	60
422-7	36	65	21	82
423-1	42	32	42	30
423-3	29	32	24	32
423-6	23	32	27	35
423-8	19	32	25	30
424-1	19	149	31	99
424-2	19	149	33	120
424-3a	27	149	32	100
424-3b	27	149	28	119
425-1	36	74	29	84
425-3	45	74	21	139
425-5	32	74	38	74
425-6	36	74	36	74
426-2a	45	246	18	153
426-2b	45	246	40	99
427-1	0	165	12	100
427-2	0	165	8	100
427-3	0	162	18	95
428-1	10	36	8	99
428-2	13	36	31	57
428-3	10	36	41	43
428-5	23	36	21	36
428-6	23	36	14	68
428-7a	26	36	12	100
428-7b	26	36	33	90

Workstation	Supply Airflow Rate [L/s]			
	Closest Diffuser		Average of Four Diffusers	
	Measured	Design	Measured	Design
428-8	23	36	24	35
428-9	16	39	36	45
429-3	32	91	20	108
429-4a	19	91	19	74
429-4b	19	91	40	108
429-5	19	91	19	74
430-4	58	32	60	41
430-7	65	32	70	32
430-11	48	32	55	32
430-12	65	32	61	32
430-15	71	32	50	30
430-16a	58	32	46	31
430-16b	58	32	60	32
430-17	45	29	38	90
431-1	16	58	40	63
431-2	6	58	20	58
431-3	23	58	30	61
431-4	19	58	19	66
431-5	16	58	22	66
431-6	32	58	23	58
431-7	26	58	25	58
431-8	36	58	22	58
431-9	29	58	42	61
432-3	68	68	67	59
432-5	71	68	57	61
432-6	74	74	48	81
433-2	97	74	57	92
433-3	91	74	68	69
433-5	96	74	66	69
433-6	85	74	67	67
433-7	87	68	69	65
434-1	0	52	34	63
434-3	0	52	24	57
434-4	0	52	21	57
434-8	0	52	0	52
434-10	0	52	0	52
434-11	0	52	0	52
435-1	0	146	23	105
435-2	0	146	10	128
435-3	0	146	34	110
436-1	45	42	31	44
436-6	39	42	38	42
436-7	40	42	41	42
436-8	45	42	42	42
436-9	39	42	42	42
436-10	45	42	42	42
436-11	37	42	42	42

Table 7: Supply Airflow Rates to Workstation Zones (1/14/93)

Percentage of Design	Number of Diffusers					
	1/14/93		2/24/93		2/25/93	
	Closest Diffuser	Average of Four	Closest Diffuser	Average of Four	Closest Diffuser	Average of Four
0%	15	4	0	0	12	1
0% < Measured ≤10%	0	3	14	6	4	0
10% < Measured ≤ 25%	12	13	6	4	3	11
25% < Measured ≤ 50%	15	24	13	17	7	9
50% < Measured ≤100%	30	38	32	41	45	49
100% < Measured	23	13	30	27	24	25

Table 8: Summary of Supply Airflow Rates to Workstation Zones

Percent Outdoor Air Intake Rate

Percent outdoor air intake rates are required to determine outdoor airflow rates to the air distribution zones of the building and were measured at the main air handlers using the automated CO_2 system. During these measurements, the economizer cycle controls were overridden to fix the positions of the mixed-air dampers. While maintaining these dampers in a fixed position will not result in a constant percent outdoor air intake in a variable air volume system, it will reduce the variation in this percentage.

Percent outdoor air intake rates measured by the automated tracer gas systems are presented in Table 9. The values for the automated system are the averages of the measurements taken every ten minutes for the entire period during which airflow measurements were being performed. The combined uncertainties of these values are between 3 and 11% of the measured percent outdoor air intake rate.

Date	System	Percent Outdoor Air
12/08/92	SFC-1&2	55
	SFC-3&4	32
	SFC-5&6	51
12/09/92	SFC-5&6	54
12/10/92	SFC-5&6	51
1/13/93	SFC-5&6	14
1/14/93	SFC-5&6	30
2/24/93 am	SFC-5&6	32
pm	SFC-5&6	41
2/25/93	SFC-5&6	38

Table 9: Percent Outdoor Air Intake Rates (Automated CO_2)

Percent Outdoor Air Uniformity

Percent outdoor air uniformity measurements were performed to demonstrate the procedure for determining the extent of mixing of outdoor and return air within the air distribution system. The percent outdoor air within the supply air system was measured using CO_2 as a tracer gas, and the results are presented schematically in Figure 4. The percent outdoor air is presented for the fan box, submains and floor branches of the West air handling system. A reference percent outdoor air intake rate, $\%OA_{ref}$, was measured when the difference in the return and outdoor concentrations measured by the automated system was at its maximum value for the day. This reference value of 50% is presented at the top of Figure 4. The average values and standard deviations for the fan box, submains, and floor branches are $45 \pm 6\%$, $42 \pm 1\%$, and $48 \pm 2\%$ respectively. Based on the measured concentrations and Equation 3, the uncertainty in these measurements is about 10%, in units of percent outdoor air. Although there does appear to be some systematic difference between the measurements performed in the fan box, submains and floor branches, the differences in the values are within the magnitude of the measurement uncertainty. It is evident that the percent outdoor air in the supply air is more uniform between and within the submains and floor branches than in the fan box. While there appears to be some variation in the percent outdoor air in the fan box, the results indicate that the submains do not receive different percentages of outdoor air. This test was performed with outdoor air brought in through both the minimum outdoor air intake fan and the economizer duct. Measurements were not performed under minimum outdoor air intake conditions, and these could have yielded different results.

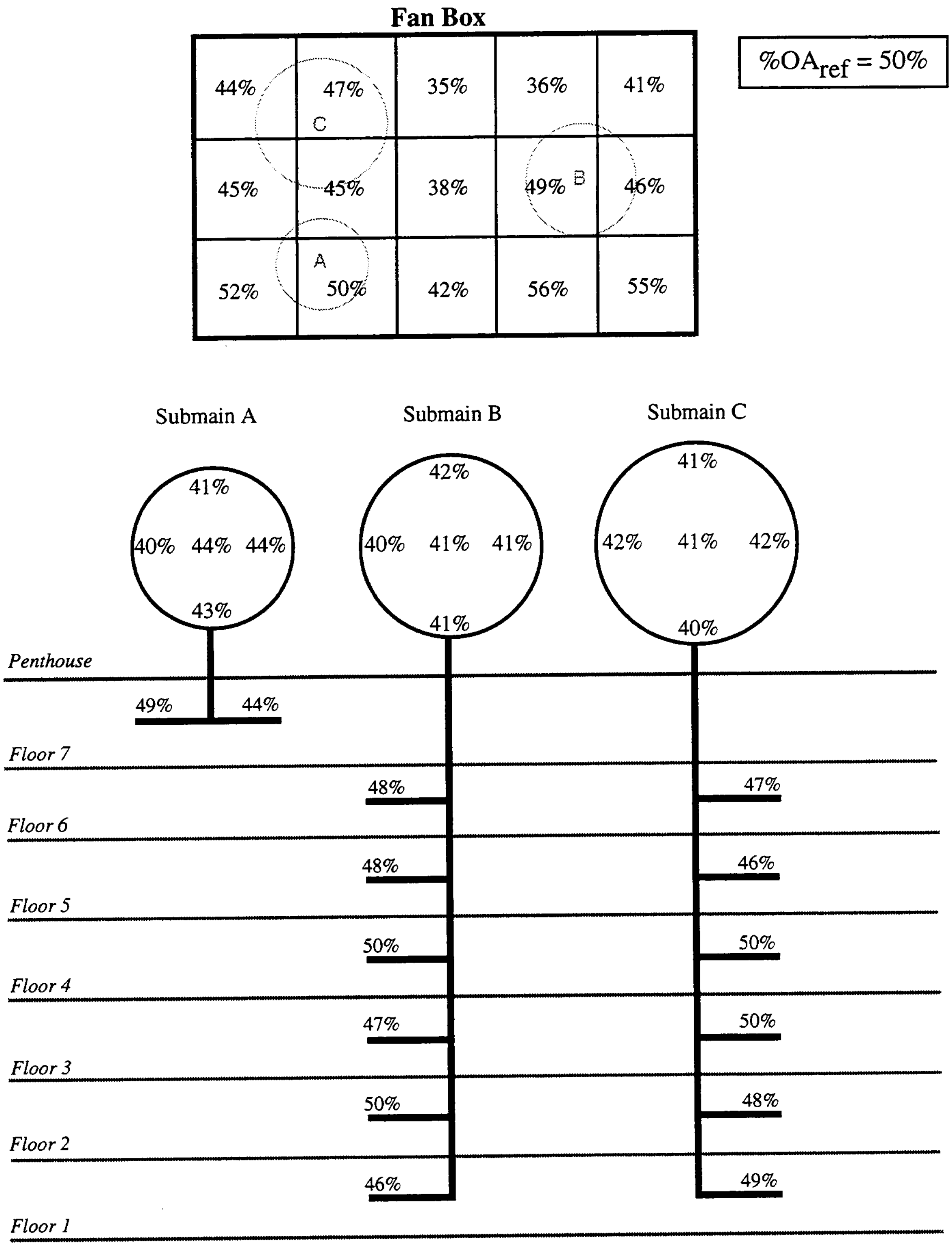

Figure 4: Percent Outdoor Air Uniformity

Outdoor Air Delivery Rate

The main goal of this project was to measure outdoor air delivery rates to sections of the building and to individual occupied spaces. Outdoor airflow rates were determined using the multiplicative method, which combines the results of supply airflow measurements and percent outdoor air intake measurements. Outdoor airflow measurements are presented in order of decreasing size of the air distribution zones, i.e., air handler, submain, floor branch, terminal unit and workstation. The percent outdoor air intake rates, used to determine the outdoor airflow rate, are based on the use of CO_2 as a tracer gas and were presented in Table 9. Total outdoor airflow rates to each zone are presented along with the estimated outdoor airflow rate per person. As stated earlier, these measurements were made during normal occupied hours with typical activities in the space.

Air Handler Zones

Outdoor airflow measurement performed for the air handler zones are presented in Table 10 as the total outdoor airflow rate to the zone in L/s and as an estimate of the outdoor airflow rate per person. This estimate is based on a building occupancy of 2000 people, divided equally between the three air handler zones. The outdoor airflow rates through the air handlers provide information on the total outdoor air delivery to the building, but not on the local outdoor air delivery rates. The per person outdoor air ventilation rates measured at the air handlers are above the minimum recommendation for office space in ASHRAE Standard 62-1989, 10 L/s per person, and above the recommendation in the 1981 version of the standard, 2.5 L/s per person for non-smoking spaces. The uncertainties for the fan box, submain summation and floor branch summation measurements are approximately 5% of the measured value. The supply airflow measurements performed inside the fan box are suspect due to poor airflow conditions at the traverse cross section as discussed earlier.

Date	Fan System	Measurement Location	Outdoor Airflow Rate	
			[L/s]*	[L/s•person]
12/8/92	SFC-3&4	Fan Box	10,500	16
		ΣSubmains	7,620	11
	SFC-5&6	Fan Box	12,800	19
		ΣSubmains	10,500	16
12/9/92	SFC-5&6	Fan Box	14,300	21
		ΣSubmains	11,700	18
		ΣFloor Branches	12,900	19
12/10/92	SFC-5&6	Fan Box	15,700	24
		ΣSubmains	11,800	18
		ΣFloor Branches	12,400	19

* 1 L/s = 2.12 cfm

Table 10: Outdoor Airflow Rates to Air Handler Zones

Submain Zones

The results of the outdoor airflow rate measurement of the submain zones are presented in Table 11. The number of people per zone is based on the floor area of the zone and the average occupant density of the air handler zones, i.e., 6 persons/100 m^2. The combined uncertainties associated with the airflow rates measured by direct traverse of the submains and the summation of the floor branch measurements are about 11% and 6% of the measured values, respectively. Per person outdoor airflow rates measured at the submains and floor branches are within 15% of the average of the two measurements, and are well above the level of 10 L/s per person recommended in ASHRAE Standard 62-1989. These outdoor airflow rates per person are close to those obtained for the air handler zone.

Date	Fan System	Submain Zone	Measurement Location	Outdoor Airflow Rate [L/s]*	Outdoor Airflow Rate [L/s•person]**
12/9/92	SFC-5&6	A	Submain	1,550	20
			ΣFloor Branches	1,700	22
		B	Submain	3,810	15
			ΣFloor Branches	4,670	18
		C	Submain	6,160	20
			ΣFloor Branches	6,380	21
12/10/92	SFC-5&6	A	Submain	1,760	23
			ΣFloor Branches	1,660	21
		B	Submain	4,560	17
			ΣFloor Branches	4,630	18
		C	Submain	5,450	18
			ΣFloor Branches	6,160	20

* 1 L/s = 2.12 cfm

** Based on an occupant density of 6 people/100 m^2

Table 11: Outdoor Airflow Rates to Submain Zones

Floor Branch Zones

Outdoor airflow rates to floor branch zones are presented in Table 12. The combined uncertainties associated with the traverse of the floor branch ducts and the summation of diffuser flow rates are about 10% of the measured results. As in the case of the supply airflow measurement results, the summation of diffuser flow rates yield lower results than traverses of the floor branch ducts which serve the diffusers. Each floor branch duct serves about one sixth of a single floor. The outdoor airflow rate per person is determined in two ways. In the first case, it is based on the occupancy determined from the floor area of the zone and the building average occupant density of 6 persons per 100 m^2. In the second case, it is based on the number of workstations in the zone. As seen in the table, the outdoor airflow rates per person are different for the two cases. The values based on floor area are lower than those based on the number of workstations for the 6-North zone and higher for the 6-South zone. The measured outdoor airflow rates to these floor branch zones are at or above the ASHRAE recommendation of 10 L/s per person for office space, in all but two cases, and are larger than the per person outdoor airflow rates for the air handler zones. The difference between the results for the two methods of determining occupancy and the difference from the air handler and submain zones are due to variations in the occupant density within the building.

Date	Fan System	Air Distribution Zone	Measurement Location	Outdoor Airflow Rate		
				[L/s]*	[L/s•person]**	[L/s•person]***
12/9/92	SFC-5&6	6-North	Floor Branch	600	21	27
		6-South	Floor Branch	1,380	28	19
12/10/92	SFC-5&6	6-North	Floor Branch	590	21	27
		6-South	Floor Branch	1,300	26	18
1/13/93	SFC-5&6	6-North	Floor Branch	220	9	10
1/14/93	SFC-5&6	6-North	Floor Branch	270	11	12
		6-South	Floor Branch	900	18	12
			Σ Diffusers	810	16	11
2/24/93	SFC-5&6	6-North	Floor Branch	690	24	31
			Σ Diffusers	620	22	28
		6-South	Floor Branch	1,530	31	21
			Σ Diffusers	1,290	26	18
2/25/93	SFC-5&6	6-North	Floor Branch	550	20	25
			Σ Diffusers	500	18	23
		6-South	Floor Branch	1,490	30	20
			Σ Diffusers	1,250	25	17

* 1 L/s = 2.12 cfm
** Based on 6 people/100 m^2
*** Based on the number of workstations in each zone

Table 12: Outdoor Airflow Rates to Floor Branch Zones

Terminal Unit Zones

Outdoor airflow rates through the terminal units of the sixth floor west zone were measured on three different days and are presented in Table 13 and Figure 5. These airflow rates are based on summations of measurements at individual diffusers. Uncertainties associated with these summations were typically between 5 and 15% of the measured values. Larger uncertainties are associated with supply airflow rates that were at or below 3.5 L/s (7.5 cfm), the uncertainty in airflow rates measured with the hooded velometer. The number of occupants of the terminal unit zones was based on the floor area associated with each terminal unit and the occupant density of the west side of the sixth floor, about 7.3 persons/100 m^2. This occupant density is based on 95 workstations and 1,300 m^2 (14,000 ft^2) of floor area. The number of occupants in a terminal unit zone is equal to the floor area associated with the terminal unit multiplied by 0.073. As seen in Table 13, the measured outdoor airflow rates through some of the terminal units are below the recommendation in ASHRAE Standard 62-1989 of 10 L/s (20 cfm) per person. Some of the measurements are also below the non-smoking recommendation in ASHRAE Standard 62-1981 of 2.5 L/s (5 cfm) per person, the standard upon which the building design was based. Even though these measurements indicate low outdoor airflow rates to some of the terminal unit zones, they do not account for outdoor airflow from adjoining zones or infiltration of outdoor air through the building envelope.

Terminal Unit Number	Outdoor Airflow Rate					
	[L/s]			[L/s•person]		
	1/14/93	2/24/93	2/25/93	1/14/93	2/24/93	2/25/93
TU418	69	123	99	20	36	29
TU419	57	86	84	12	18	17
TU420	0	4	80	0	2	29
TU421	69	103	80	13	19	15
TU422	39	98	84	8	21	17
TU423	57	103	103	8	15	15
TU424	24	62	87	9	23	32
TU425	51	94	87	15	28	26
TU426	18	33	42	13	24	31
TU427	0	16	27	0	8	13
TU428	36	8	0	6	1	0
TU429	24	103	95	7	30	28
TU430	228	287	239	21	26	22
TU431	51	115	110	7	17	16
TU432	90	37	34	22	9	8
TU433	141	238	205	26	44	38
TU434	0	131	114	0	17	15
TU435	0	49	4	0	24	2
TU436	108	160	137	13	20	17
TU437	--	37	46	--	14	17

Table 13: Outdoor Airflow Rates to Terminal Unit Zones

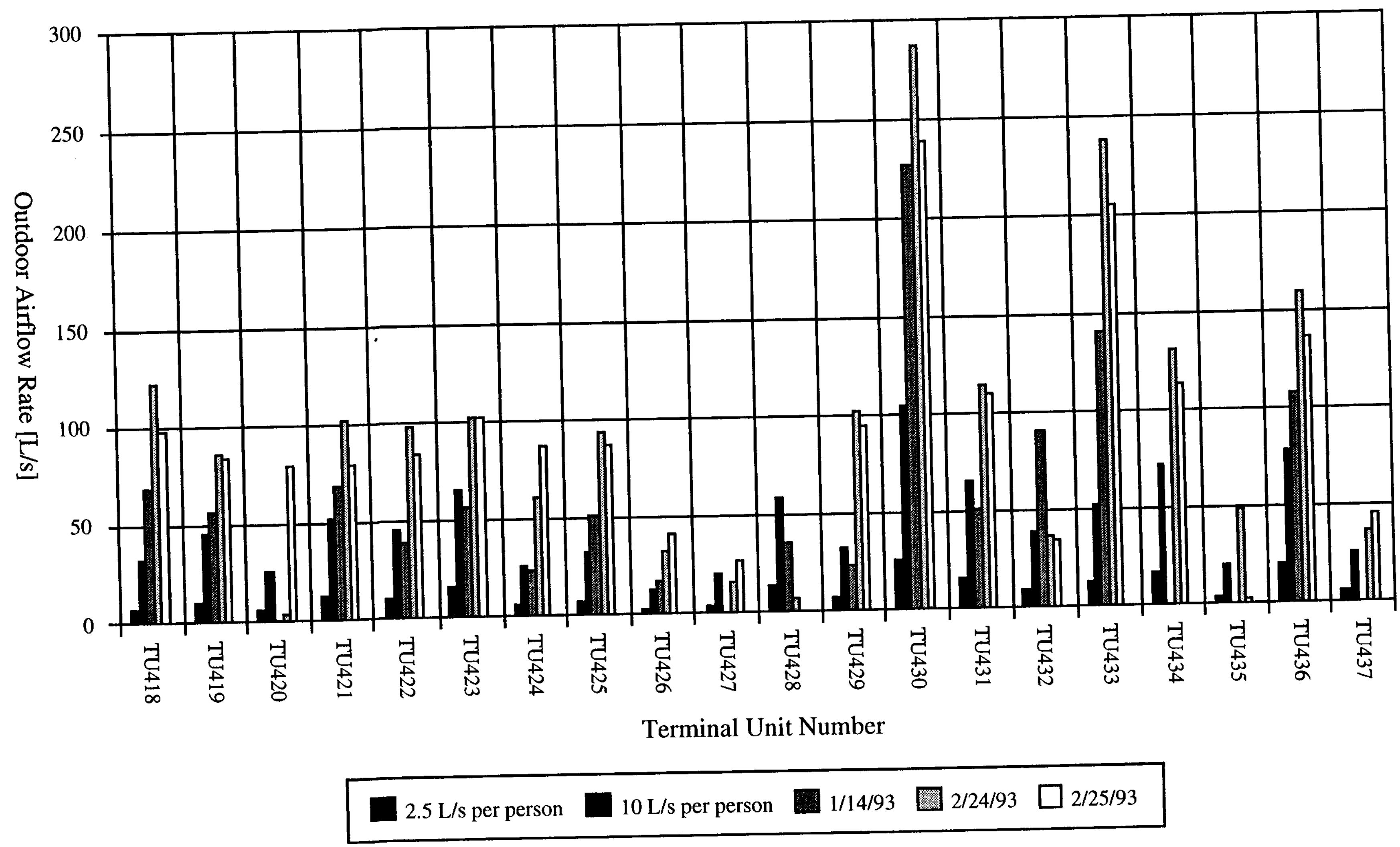

Figure 5: Outdoor Airflow Rates through Terminal Units on the West Side of the Sixth Floor

Workstation Zones

Outdoor airflow rates to individual workstation zones were estimated based on measurements at individual diffusers. These estimates were made for all ninety-five workstations on the west side of the sixth floor on three days, and the results for one day are presented in Table 14. Table 14 shows the outdoor airflow rates to each workstation zone indicated by the terminal unit number and diffuser number nearest to the workstation. Two outdoor airflow rates are given for each workstation. The first column, labeled Closest Diffuser, is based on the outdoor airflow rate measured at the diffuser closest to the workstation. The values in the table are based on the actual measured outdoor airflow rates multiplied by the ratio of the number of diffusers on the west side of the sixth floor that serve workstations to the number of workstations, i.e., 1.37. The use of this multiplier converts the diffuser airflow rates into an estimate of the outdoor airflow rate per workstation. The last column in Table 14, labeled Average of Four Diffusers, is the average of the measured airflow rates of the four diffusers closest to the workstation. These values are also adjusted based on the ratio of diffusers to workstations. This average value is presented because this is an open office space and the outdoor airflow to an individual workstation depends on more than just the airflow rate from the nearest diffuser. The average value is used to obtain a better estimate of the actual outdoor airflow rate to the workstation. The combined uncertainties associated with these measurements are typically between 5 and 30% of the measured outdoor airflow rates when the diffuser supply airflow rates are greater than 3.5 L/s (7.5 cfm), i.e., the measurement uncertainty of the hooded velometer.

There are several cases in Table 14 for which the outdoor airflow rates for the closet diffuser is below the recommendations in ASHRAE Standard 62-1981 and 62-1989 of 2.5 (5 cfm) and 10 L/s (20 cfm) per person, respectively. Table 15 summarizes the results of the workstation outdoor airflow rate measurements for the three days of testing. The table shows the number of workstations with measured outdoor airflow rates equal to zero, between 0 and 2.5 L/s per person, between 2.5 and 5 L/s, between 5 and 10 L/s, and greater than or equal to 10 L/s. These results are given for the workstation outdoor airflow rates based on the closest diffuser and the average of the four closest diffusers for all three days of testing. The results in Table 15 show several workstations have zero or very low outdoor airflow rates during the measurements. For each day of measurements, the outdoor airflow rates of approximately 20% of the 95 workstations were below 2.5 L/s (5 cfm) per person based on the closest diffuser. The number of low-airflow cases is much less for the average of the four diffusers. As mentioned earlier, each workstation does not have a dedicated set of supply air diffusers, and air from any one diffuser will impact more than one workstation. Even though a measurement indicates that the outdoor air supply to an individual workstation is below the recommendations in the ASHRAE standard, the actual rate of outdoor air delivery is also impacted by other nearby diffusers, air mixing in the space and infiltration. It is important to note that these measurements only provide information on the outdoor airflow rate to the workstation at the time of the measurement.

Outdoor Airflow Rate [L/s]		
Workstation	Closest Diffuser	Average of Four Diffusers
418-2	20	18
418-3	20	17
418-4	18	16
419-1	12	12
419-2a	12	15
419-2b	12	15
419-3a	11	12
419-3b	11	11
419-7a	12	10
419-7b	12	15
420-3	0	0
420-4	0	3
420-5	0	6
421-1a	2	10
421-1b	2	11
421-2	15	11
421-5	13	12
421-6	12	12
421-7	14	6
421-8	12	8
422-1	10	8
422-2	10	8
422-6	9	9
422-7	11	6
423-1	13	13
423-3	9	7
423-6	7	8
423-8	6	8
424-1	6	9
424-2	6	10
424-3a	8	10
424-3b	8	8
425-1	11	9
425-3	14	6
425-5	10	11
425-6	11	11
426-2a	14	5
426-2b	14	12
427-1	0	4
427-2	0	2
427-3	0	5
428-1	3	2
428-2	4	9
428-3	3	12
428-5	7	6
428-6	7	4
428-7a	8	4
428-7b	8	10

Outdoor Airflow Rate [L/s]		
Workstation	Closest Diffuser	Average of Four Diffusers
428-8	7	7
428-9	5	11
429-3	10	6
429-4a	6	6
429-4b	6	12
429-5	6	6
430-4	17	18
430-7	19	21
430-11	15	16
430-12	19	18
430-15	21	15
430-16a	17	14
430-16b	17	18
430-17	14	11
431-1	5	12
431-2	2	6
431-3	7	9
431-4	6	6
431-5	5	7
431-6	10	7
431-7	8	8
431-8	11	7
431-9	9	13
432-3	20	20
432-5	21	17
432-6	22	14
433-2	29	17
433-3	27	20
433-5	29	20
433-6	26	20
433-7	26	21
434-1	0	10
434-3	0	7
434-4	0	6
434-8	0	0
434-10	0	0
434-11	0	0
435-1	0	7
435-2	0	3
435-3	0	10
436-1	14	9
436-6	12	11
436-7	12	12
436-8	14	13
436-9	12	13
436-10	14	12
436-11	11	12

Table 14: Outdoor Airflow Rates to Workstation Zones (1/14/93)

Outdoor Airflow Rate [L/s per person]	Number of Diffusers					
	1/14/93		2/24/93		2/25/93	
	Closest Diffuser	Average of Four	Closest Diffuser	Average of Four	Closest Diffuser	Average of Four
0	15	4	0	0	12	1
0 < Measured < 2.5	3	2	16	6	5	1
2.5 ≤ Measured < 5	6	5	3	5	2	7
5 ≤ Measured < 10	25	37	8	15	6	14
10 ≤ Measured	46	47	68	69	70	72

Table 15: Summary of Outdoor Airflow Measurements to Workstation Zones

Local Age of Air

Local age of air measurements were performed using the tracer gas decay technique with SF_6 as a tracer gas. Measurements were performed on 13 and 14 January and 24 and 25 February 1993. These tests are referred to as A, B, C and D respectively, and the results are presented in Tables 16, 17, 18 and 19. During these tests the mixed-air dampers of the three main air handlers were locked in a fixed position to provide a roughly constant percent outdoor air intake rate of approximately 30%, except on 24 February when the percent outdoor air intake rate was about 40%.

The automated sample results shown in the tables are presented according to the return fan which serves the sample locations. Return fans 6, 7 and 8 serve the center, east and west sections of the building respectively. The labels for the sample locations indicate the floor of the building and the structural columns at which the samples were taken. These column numbers refer to floor plans of the building contained in reference [4]. All locations at which air bag samples were collected are on the west side of the sixth floor, which is served by air handler SFC-5&6 and return fan 8. Samples collected using average bags are so indicated in the tables. For each sample location the table gives the initial tracer gas concentration $C_{i,0}$, the local age of air τ_r (for return locations) or τ_i (for space locations), and the air change effectiveness. $C_{i,0}$ is the concentration measured at location i just prior to stopping the tracer gas injection. τ_r is the age of air measured in the main return duct at the air handlers, and $\tau_{r,vent}$ was measured at the ceiling return air vent closest to the occupied space sample location. The air change effectiveness for the return fan and return vent sample locations is the nominal time constant of the building τ_n divided by the age of air in the return. The air change effectiveness for sample locations in the occupied space is the age of air in the return fan or return vent divided by the age of air in the occupied space. The equilibrium tracer gas concentration used to calculated the nominal time constant and the value of τ_n are shown at the bottom of the tables. For the measurements performed in this building, the uncertainty in the local age of air using the numerical integration and the average bag sampling method are respectively about 6% and 8% of the measured value.

As mentioned earlier, these measurements require initial conditions of a uniform tracer gas concentration throughout the building. The fact that this building has three central air handlers that serve all seven floors of the building, and that the center, east and west zones communicate freely on the floors, made it relatively easy to achieve a uniform tracer gas concentration. For the four tests, the standard deviations of the equilibrium tracer gas concentrations $C_{i,0}$ are 12%, 10%, 11% and 12% of their mean values. While there have been very few field measurements of local age of air in mechanically ventilated office buildings, variations of 10% in the equilibrium concentration are probably as small as can reasonably be expected. In the last three tests, the equilibrium concentration at location 6-M17, in the elevator lobby, was well below the average for the rest of the locations. The elevator shafts extend from the underground garage levels up to the seventh floor, and the flow of tracer-free air from the garage to this location may account for the lower tracer gas concentrations at 6-M17.

In the ideal case of uniform distribution of outdoor air to all of the building zones and perfect mixing of the ventilation air within these zones, the age of air equals τ_n and the air change effectiveness equals 1.0 at all locations in the building. The values of air change effectiveness for the return fan locations, τ_n/τ_r are all within 20% of 1.0, with the largest deviations from 1.0 occurring for Test A. Neglecting the results for Test A, the average value of τ_n/τ_r is 0.99 and the standard deviation is 0.09. This result indicates that for these test conditions, the outdoor air ventilation rate is uniform for the three portions of the building served by the different air handling systems.

For the occupied space locations, the values of the air change effectiveness are almost all within 15% of 1.0. A value of τ_r/τ_i that is close to 1.0 indicates the existence of good mixing of the ventilation air within the occupied space, while a value significantly less than 1.0 will occur at locations that are bypassed by the ventilation air. For those locations at which a value of $\tau_{r,vent}$ was available, the values of air change effectiveness based on $\tau_{r,vent}$ are generally closer to 1.0 than the values based on the age of air measured at the return fan. This is presumably because the age of air in a local return is a more appropriate reference age than that measured in a return duct that serves one-third of the entire building.

The age of air measurements made with average bags were from 2 to 30% greater than the values obtained based on concentrations from periodic air samples. All but one of the average-bag values were within 15%, and the average percentage difference was 11%. The reason for this bias is not known at this time.

Location	$C_{i,0}$ [ppb]	Age of Air (hours) Return, τ_r	Space, τ_i	Air Change Effectiveness τ_n / τ_r	τ_r / τ_i	$\tau_{r,vent} / \tau_i$
AUTOMATED SAMPLES						
Return fan 6	69	1.39	--	0.84	--	
6-Q14	66	--	1.41	--	0.99	
Return fan 7	64	1.43	--	0.82	--	
6-H22	83	--	1.11	--	1.29	
6-M22	74	--	1.31	--	1.10	
Return fan 8	62	1.39	--	0.84	--	
7-V4	76	--	1.23	--	1.13	
6-V4	64	--	1.41	--	0.98	0.81
6-S9	72	--	1.25	--	1.11	
5-V4	64	--	1.49	--	0.93	
4-V4	64	--	1.61	--	0.86	
3-V4	62	--	1.60	--	0.87	
2-V4	61	--	1.52	--	0.92	
1-V4	56	--	1.64	--	0.85	
AIR BAG SAMPLES		$\tau_{r,vent}$				
6-V4 Return Vent	67	1.14	--	1.03	--	
6-V4 (average bag)	64	--	1.20	--	1.16	
TU436 Return Vent	76	1.04	--	1.13	--	
TU436 Workstation	82	--	0.89	--	1.56	1.17

Equilibrium concentration = 65 ppb, τ_n = 1.17 hours

Table 16: Results of Local Age of Air Test A

Location	$C_{i,0}$ [ppb]	Age of Air (hours) Return, τ_r	Space, τ_i	Air Change Effectiveness τ_n/τ_r	τ_r/τ_i	$\tau_{r,vent}/\tau_i$
AUTOMATED SAMPLES						
Return fan 6	126	1.45	--	0.94	--	
6-M17	97	--	1.67	--	0.87	
6-Q14	121	--	1.30	--	1.12	
Return fan 7	128	1.52	--	0.90	--	
6-H22	147	--	1.27	--	1.19	
6-M22	147	--	1.21	--	1.26	
Return fan 8	120	1.40	--	1.02	--	
7-V4	142	--	1.24	--	1.13	
6-V4	122	--	1.15	--	1.22	1.08
6-S9	126	--	1.22	--	1.14	
5-V4	120	--	1.35	--	1.03	
4-V4	123	--	1.39	--	1.00	
3-V4	138	--	1.46	--	0.96	
2-V4	119	--	1.35	--	1.04	
1-V4	112	--	1.48	--	0.95	
AIR BAG SAMPLES		$\tau_{r,vent}$				
6-V4 Return Vent	120	1.24	--	1.10	--	
(average bag)	120	1.38	--	0.99	--	
6-V4	118	--	1.24	--	1.13	1.00
(average bag)	118	--	1.35	--	1.04	1.02
TU436 Return Vent	132	1.20	--	1.13	--	
(average bag)	132	1.35	--	1.01	--	
TU436 Workstation	137	--	1.19	--	1.18	1.01
(average bag)	137	--	1.21	--	1.16	1.12
6-Q6 Workstation	151	--	1.09	--	1.28	
(average bag)	151	--	1.12	--	1.25	

Equilibrium concentration = 125 ppb, τ_n = 1.36 hours

Table 17: Results of Local Age of Air Test B

Location	$C_{i,0}$ [ppb]	Age of Air (hours) Return, τ_r	Space, τ_i	Air Change Effectiveness τ_n/τ_r	τ_r/τ_i
AUTOMATED SAMPLES					
Return fan 6	73	1.00	--	1.04	--
6-M17	53	--	.91	--	1.10
6-Q14	66	--	0.97	--	1.03
Return fan 7	74	0.99	--	1.05	--
6-H22	84	--	0.76	--	1.29
6-M22	83	--	0.77	--	1.28
Return fan 8	75	0.93	--	1.12	--
7-V4	83	--	0.83	--	1.13
6-V4	76	--	0.83	--	1.12
6-S9	69	--	0.81	--	1.15
5-V4	64	--	1.02	--	0.92
4-V4	76	--	0.94	--	0.99
3-V4	77	--	0.98	--	0.95
2-V4	75	--	1.01	--	0.92
1-V4	77	--	1.12	--	0.83

Equilibrium concentration = 74 ppb, τ_n = 1.04 hours

Table 18: Results of Local Age of Air Test C

Location	$C_{i,0}$ [ppb]	Age of Air (hours) Return, τ_r	Space, τ_i	Air Change Effectiveness τ_n / τ_r	τ_r / τ_i	$\tau_{r,vent} / \tau_i$
AUTOMATED SAMPLES						
Return fan 6	113	1.07	--	0.90	--	
6-M17	81	--	1.18	--	0.90	
6-Q14	100	--	0.89	--	1.20	
Return fan 7	118	1.09	--	0.88	--	
6-H22	125	--	0.94	--	1.16	
6-M22	117	--	0.90	--	1.22	
Return fan 8	116	0.87	--	1.10	--	
7-V4	116	--	0.84	--	1.02	
6-V4	116	--	0.74	--	1.17	1.05
6-S9	102	--	0.85	--	1.02	
5-V4	98	--	0.91	--	0.95	
4-V4	119	--	0.75	--	1.16	
3-V4	123	--	0.92	--	0.94	
2-V4	118	--	0.73	--	1.19	
1-V4	119	--	0.92	--	0.94	
AIR BAG SAMPLES		$\tau_{r,vent}$				
6-V4 Return Vent	114	0.78	--	1.23	--	
(average bag)	114	0.86	--	1.12	--	
6-V4 (average bag)	116	--	0.81	--	1.07	1.06
TU436 Return Vent	130	0.77	--	1.25	--	
(average bag)	130	0.82	--	1.17	--	
TU436 Workstation	127	--	0.80	--	1.09	0.96
(average bag)	127	--	0.91	--	0.96	0.90
6-Q6 Return Vent	132	0.74	--	1.30	--	
(average bag)	132	0.96	--	1.00	--	
6-Q6 Workstation	144	--	0.70	--	1.24	1.06
(average bag)	144	--	0.77	--	1.13	1.25

Equilibrium concentration = 117 ppb, τ_n = 0.96 hours

Table 19: Results of Local Age of Air Test D

The values of the so-called local nominal time constant $\tau_{n,local}$ were also compared to the local ages of air measured in the occupied space. As defined in Equation 8, $\tau_{n,local}$ is the inverse of the local air change rate based on the building volume associated with a diffuser or a group of diffusers and the outdoor airflow rate from those diffusers determined with the multiplicative method. Table 20 shows the results of this comparison for those locations at which the diffuser airflow rates and the local age of air were measured. The local nominal time constant was calculated for the diffuser closest to the location of the age of air measurement and for the four closest diffusers. The age of air measured in the closest return vent is also included in the table. The uncertainty in $\tau_{n,local}$ calculated using Equation 9 is approximately 13% of the reported values.

Date	Time	Location	Airflow Rate [L/s]		$\tau_{n,local}$		τ_i	$\tau_{r,vent}$
			Single Diffuser	Avg. of Four Diffusers	Single Diffuser	Four Diffusers	Tracer Decay	Tracer Decay
1/14/93	9:45	6-Q6	0	32	-	0.99	1.09	-
	10:13	6-Q8	28	30	1.12	1.07	1.19	1.20
	13:02	6-V4	26	28	1.22	1.14	1.24	1.24
2/25/93	13:22	6-Q6	28	45	0.88	0.56	0.70	0.74
	13:34	6-Q8	26	28	0.96	0.88	0.80	0.77
	15:12	6-V4	38	28	0.66	0.88	0.74	0.78

Table 20: Local Nominal Time Constants Based on Diffuser Airflow Measurements

The local time constants obtained using the four diffuser values were within 10% of the age of air in the occupied space and the return air vents for the measurements on 1/14/93, and within 20% on 2/25/93. If $\tau_{n,local}$ was a good measure of the outdoor air delivery to the space and if the ventilation air was well-mixed within the ventilated space, then $\tau_{n,local}$ and τ_i would be equal. In making these comparisons, it is important to note that the age of air measurements are performed over a relatively long period of time compared to the time required to perform the airflow measurements used to calculate $\tau_{n,local}$. In addition, the local nominal time constant only accounts for air delivered by the air distribution system and does not take into account outdoor airflow into the space from other spaces. The value of the local age of air is influenced by these other outdoor air sources, as well as by envelope infiltration.

MEASUREMENT ISSUES

This section addresses practical issues involved in performing measurements of local outdoor air delivery rates. The level of effort required for preparation, measurement, and analysis for the various air distribution zones of this building are presented. The impacts of the building and HVAC system configuration on performing these measurements are also discussed.

Level of Effort

Level of effort refers to the amount of time, equipment, and technical expertise required to measure a ventilation performance parameter. While the levels of effort required to perform the measurements for this building do not necessarily apply to other buildings, they do provide an indication of the relative amounts of effort.

All the approaches to ventilation assessment have advantages and disadvantages. The amount of effort associated with each technique, and the completeness and quality of the information obtained, is a function of the building being studied including the building layout and HVAC system configuration. Prior to performing any measurements, a thorough understanding of the mechanical ventilation system is required. This involves gathering information from design documentation and drawings, talking to building operators, and performing on-site inspections of the system. The level of effort required for the design evaluation depends on the size and complexity of the building and HVAC system.

Additional resource requirements include the initial cost of the measurement equipment, the cost associated with equipment installation, calibration and maintenance, the number of measurements, and the time for data analysis. The number of measurements is an important consideration when deciding between an automated monitoring system or a manual approach. This decision involves a trade-off between the amount of time required to make the manual measurements and the installation time of an automated system.

Table 21 lists the ventilation performance parameters measured in this study, the air distribution zone for the performance parameter, the measurement technique used, the equipment required to perform the measurement, the initial set-up required prior to performing the tests, the time required to perform the initial set-up, and the time required to apply the measurement technique for each zone. The times listed in Table 21 are based on the average amount of time required to perform the measurements in this building, including the time required to move from one measurement location to another.

Several key issues to consider in performing a ventilation system evaluation are not accounted for in Table 21. Among these issues are the amount of time required to evaluate the mechanical system design and the time required to perform the data analysis once the measurements have been performed. Both can be significant considerations depending on the complexity of the system and the performance parameter being measured. Table 21 also does not account for the skill required to troubleshoot problems with the measurement systems and delays due to problems with controlling the building mechanical systems during the tests.

Parameter Measured	Air Distribution Zone	Measurement Technique	Measurement Equipment	Initial Set-up	Set-up Time Set-up	Measurement Time
Supply Airflow Rate	Air Handler	Velocity traverse of fan box	Hot-wire anemometer	Layout traverse	10 min	16 min
	Submain	Velocity traverse of submain Σ Floor branch traverses	Hot-wire anemometer Hot-wire anemometer	Layout traverse and drill holes	10 min 25 min	20 min 2.5 to 3 hours
	Floor Branch	Velocity traverse of floor branch Σ Diffuser airflows 6-North = 37 diffusers 6-South = 102 diffusers	Hot-wire anemometer Hooded velometer	Layout traverse and drill holes Map out diffusers	25 min 1 hour	20 min 1 hour 2 hours
	Terminal Unit	Velocity traverse of terminal unit duct Σ Diffuser airflows of 2 to 16 diffusers per terminal unit Velocity pressure measurement	Hot-wire anemometer Hooded velometer Digital manometer	Layout traverse and drill holes Map out diffusers Connect tubes to velocity pressure taps of terminal unit	20 min 5 min 15 min	20 min 4 to 32 min 2 min
	Workstation	Σ Diffuser airflows of 4 diffusers	Hooded velometer	Assemble hooded velometer	10 min	8 min
	Diffuser	Diffuser airflow	Hooded velometer	Assemble hooded velometer	10 min	2 min
Percent Outdoor Air	Air handler	Tracer gas	Automated CO_2 system Automated SF_6 system	Install tubing and automated system to sample supply, return, and outdoor air of a main air handler Calibrate SF_6 detector Calibrate CO_2 detector	8 hours 1 hour 30 min	1 hour
Local Age of Air	Workstation	SF_6 tracer gas decay	Air sample bags Portable sample pumps	Calibrate SF_6 detector Inject SF_6 tracer gas to obtain an equilibrium concentration Clean air sample bags (60 bags)	1 hour 6 hours 45 min	4 hours

Table 21: Levels of Effort Required to Obtain Ventilation Performance Parameters

Practical Constraints

Several airflow measurement techniques were demonstrated in this particular building. Due to practical constraints and physical limitations imposed by the configuration of the mechanical system, not all techniques could be applied. In some instances, the direct measurement of the airflow rate in a section of the air distribution system was impractical due to inaccessibility of the duct or severely non-uniform airflow patterns. In other cases, the amount of time involved in performing the measurements was impractical due to the modulation of supply airflow rates during the course of the measurement. Every building should be considered on an individual basis in terms of which measurement techniques will provide the desired information in the most practical and accurate manner.

Fortunately, in most cases different techniques can be used to measure the flow rate to a given air distribution zone with varying degrees of accuracy, set-up time, and measurement time. For example, the airflow rate to an entire floor could be obtained by performing hot-wire traverses of each floor branch duct serving the floor or by using a hooded velometer to measure the flow rate out of each supply air diffuser on the floor. In the case of the sixth floor there are six cold supply fan floor branch ducts which could be traversed. Set-up time for the traverse of each duct would require layout and drilling of traverse holes, whereas the diffuser measurements would require no initial set up. However, the diffuser measurements would not necessarily take less time since there may be as many as 300 diffusers on a single floor. Also, diffuser airflow measurements are associated with a larger degree of measurement error [7].

SUMMARY AND CONCLUSIONS

Outdoor air delivery rates were determined for various air distribution zones ranging in size from an individual workstation to the space served by an air handler. These outdoor airflow rates were determined using the multiplicative method which entails the measurement of supply airflow and percent outdoor air intake rates to these various zones of the building. Values of the local age of air were also measured, as an indicator of the distribution of ventilation air to individual workstations.

The measurements of supply airflow rates revealed several key issues. When performing supply airflow measurements, the measurement location and the use of recommended guidelines is important. In some cases, good agreement was obtained between measurements of the same supply airflow rate at different locations in the air distribution system. In other cases, differences existed between supply airflow rates measured in a single duct and the sum of supply airflow rates measured in multiple downstream ducts. These differences are consistent with, but not necessarily attributable to, duct leakage. Measured supply airflow rates to several terminal units were less than 25% of design capacity, and in some instances, the supply airflow rates through some terminal units were not detectable. Supply airflow rates to about one-fifth of the measured workstations were below 25% of the design capacity.

The outdoor airflow rate measurements revealed that while an appropriate amount of outdoor air may be brought in by the main air handling system, it is not necessarily delivered to all spaces served by the air handler. The measured outdoor airflow rates per person on the scale of air handlers, submains and floor branches were consistent with the recommendations of 10 L/s per person given in ASHRAE Standard 62-1989. However, the outdoor airflow rate per person to some of the terminal unit zones and workstation zones were below this recommended level. Several instances were observed when airflow to some terminal units was completely shut off, eliminating the flow of outdoor air to as many as fifteen diffusers at a time. When these flows drop to zero, the only sources of outdoor air for the workstations served by these diffusers are infiltration and mixing of air from other nearby zones. These results are only indicative of supply and outdoor airflow rates at the time of the measurements and under the conditions during the measurements. In order to more fully characterize the performance of the air distribution system, real-time airflow measurements are required under a variety of heating and cooling conditions.

Both automated and manual sampling techniques were demonstrated for measuring local age of air to provide information on the distribution and mixing of ventilation air. Values of air change effectiveness based on tracer gas decay measurements of the local age of air were consistent with good mixing of the ventilation air in the occupied space. However, the complexity of this measurement technique makes it inappropriate for wide use in field evaluations of building ventilation.

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