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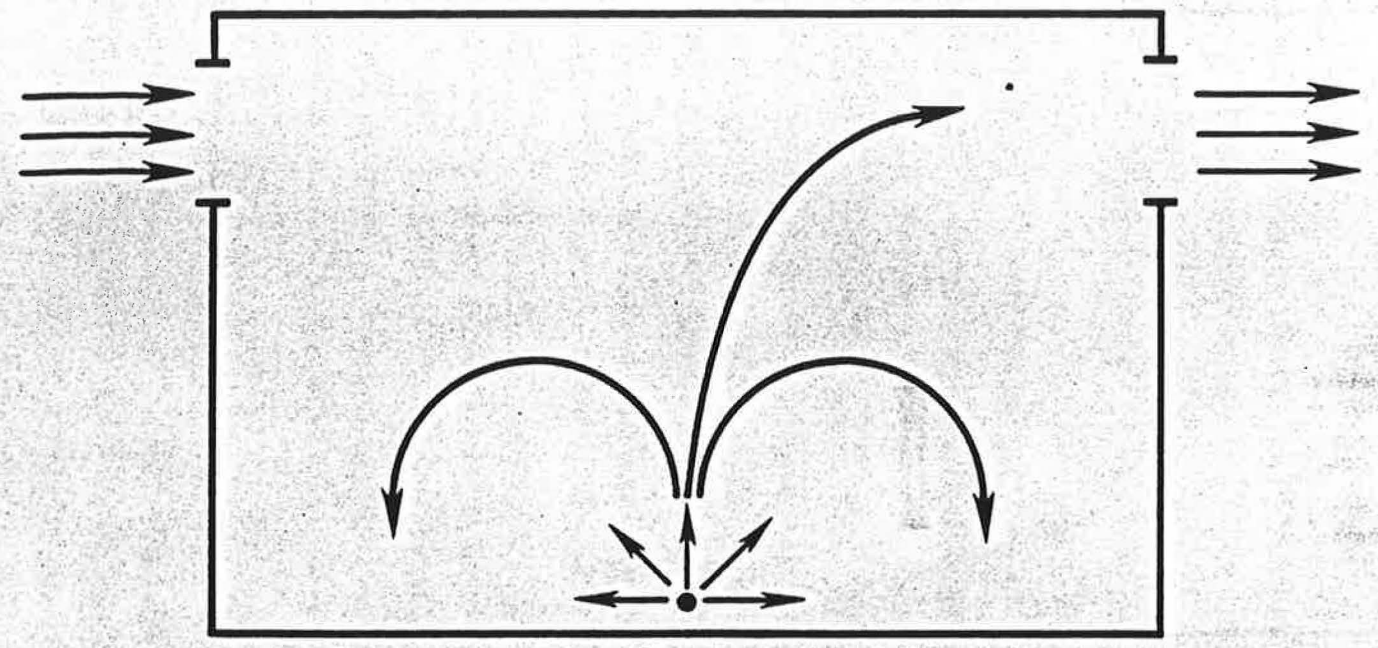

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ABSTRACT

The ability to control indoor contaminants is dependent upon a large number of ventilation system parameters, including system geometry, flow rate, thermal stratification, pollutant source type, and pollutant source distribution. A ventilation model is proposed that makes it possible to determine if undesirable contaminant levels are caused primarily by low contaminant removal rates or by inefficient ventilation air supply and distribution. The results of a series of experiments are presented to demonstrate the capabilities of a unique laboratory-scale facility that has been designed to provide a quick and cost-effective method of evaluating ventilation system performance as a function of system geometry and operating conditions. Image processing and flow visualization techniques are coupled with direct velocity measurements to evaluate three-dimensional pollutant distributions. The information from this scale facility, when combined with the results of full-scale measurements and numerical calculations, will make it possible to determine whether mitigation efforts should focus on source control, increased ventilation rates, or ventilation system modification.

BACKGROUND

Because of the large fraction of time that is spent in the indoor environment, the quality of indoor air is an important component in determining our overall level of health and comfort. Indoor air quality is also closely linked to building energy consumption. Conditioning of ventilation air (heating, cooling, humidifying, dehumidifying, cleaning) accounts for a large fraction of the energy usage in buildings ranging in size from residences to high-rise office buildings. More than 25 quads of energy are used to heat, cool, light, and ventilate buildings in the United States every year. This energy costs the users more than $166 billion and represents 36% of the primary energy use in the United States. Conditioning and transport of ventilation air account for 50 to 60 percent of total

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building energy requirements. Preliminary research results indicate that increased understanding of building airflow can produce substantial increases in ventilation effectiveness and corresponding reductions in building energy use. If one assumes that the results of this research can be used to reduce ventilation energy use by 15 percent and also assumes that this knowledge penetrates only 15 percent of the total market, one still has the potential to save nearly two billion dollars per year.

A broad range of pollutants affect indoor air quality. Some of the major pollutants include:[1]

Combustion byproducts. Unvented heaters and gas stoves can produce particulates, carbon monoxide, carbon dioxide, and nitrogen oxides. The health effects of carbon monoxide are well known. The nitrogen oxides are suspected of causing acute health effects (Traynor 1985).

Tobacco smoke. The health effects of cigarette smoke on smokers are well understood. Recent research suggests that some health effects may also occur in passive smokers (Repace 1984) and the odor of tobacco smoke is objectionable to some people.

Radon. Radon is a decay product produced from naturally occurring radium in soils, groundwater and building materials. The EPA recently estimated that several million homes in the U.S. have high enough levels of radon that the lung cancer risk associated with lifetime exposure would be substantially higher that those for nonsmokers (Nero et al. 1986).

Formaldehyde. Formaldehyde is widely used in many building materials including particle board and some types of foam insulations. Formaldehyde is an irritant and is also suspected of being a human carcinogen (EPA 1984).

Carbon Dioxide. The concentration of carbon dioxide in exhaled breath is 3.8% as opposed to 0.04% in outside air (Woods 1986). Control of carbon dioxide levels forms one of the criteria for ventilation rate recommendations in ASHRAE Standard 62-81 (ASHRAE 1981).

As a result of the emphasis on energy conservation in the past decade, lighting levels have been reduced (from 4 watts/ft^2 to 2 watts/ft^2), buildings have been better insulated, and air infiltration levels have been reduced. Ventilation systems with reduced flow rates, such as variable air volume systems, have also become popular because of the corresponding reduction in first costs and operating costs that can be achieved by ventilating at reduced flow rates. These reduced flow rates have the potential to reduce comfort and increase problems with indoor air pollutants. Energy-conserving homes have in some cases aggravated indoor air quality problems (Andersen et al. 1975).

The simplest models for gauging indoor air quality have assumed that the air in different building zones is well mixed. Measurements indicate that this assumption may not be valid in all cases (Sandberg 1981; Rezvan 1984; Skaret and Mathison 1982). Janssen, Hill, Wood, and Maidonado (1982) found that nearly 50% of the supply air in a school short-circuited directly to the return duct without mixing with room air. A recent study by Offerman (1988) reports similar findings.

Building occupants are sensitive to air quality issues. One survey of office workers by Honeywell indicated that a significant minority of office workers (24%) were dissatisfied with air quality in the workplace. Twenty percent of the respondents felt that poor air quality interfered with their ability to do their job. (IAQ Survey 1985).

One objective of a ventilation system is to remove the pollutants generated inside a building and distribute a sufficient quantity of outside air so that an acceptable level

[1] Additional background information on indoor air pollutants can be found in ASHRAE 1987a.

of indoor air quality can be maintained. These functions are shown in Figure 1. The ventilation rate that is required can vary considerably, depending upon the type, number, and distribution of pollutant sources in the zone that is being ventilated.

The ventilation rate also depends upon the ventilation strategy that is used. Two idealized ventilation strategies are shown in Figures 2a and 2b. In Figure 2a the supply air is immediately and completely mixed with the air that is already in the room. This approach has the advantage of diluting pollutant concentrations but makes it difficult to directly remove pollutants before they mix with room air. Under conditions of perfect displacement (Fig. 2b), the supply air acts like a piston which displaces an equal volume of air as it enters the room. Most ventilation systems depend upon a combination of mixing and displacement for removal of pollutants. Researchers in Europe have suggested that displacement systems that use thermal stratification to produce a piston-like flow, may produce efficient pollutant removal (Skaret 1983).

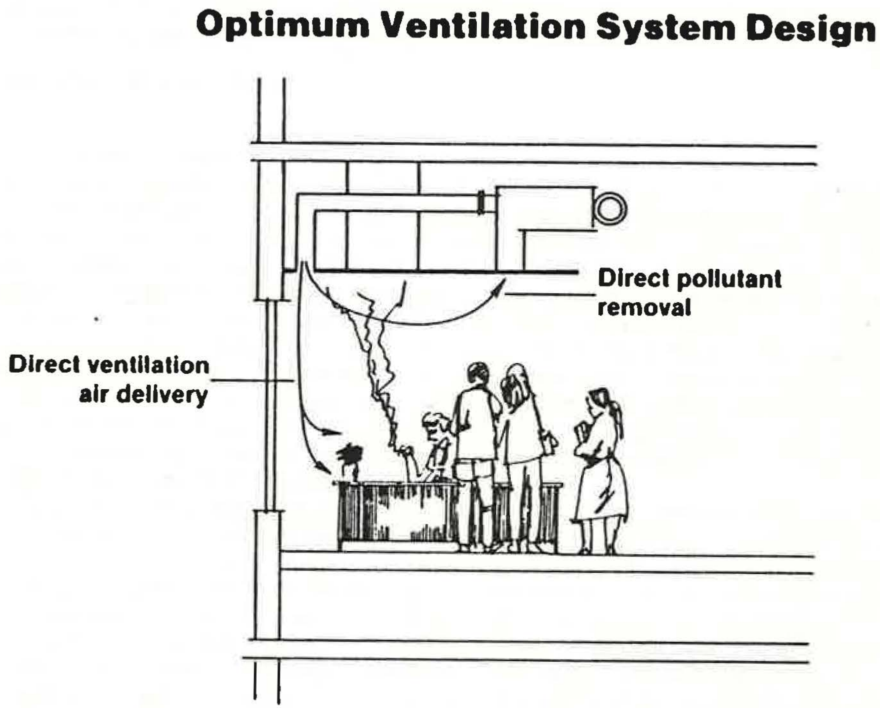
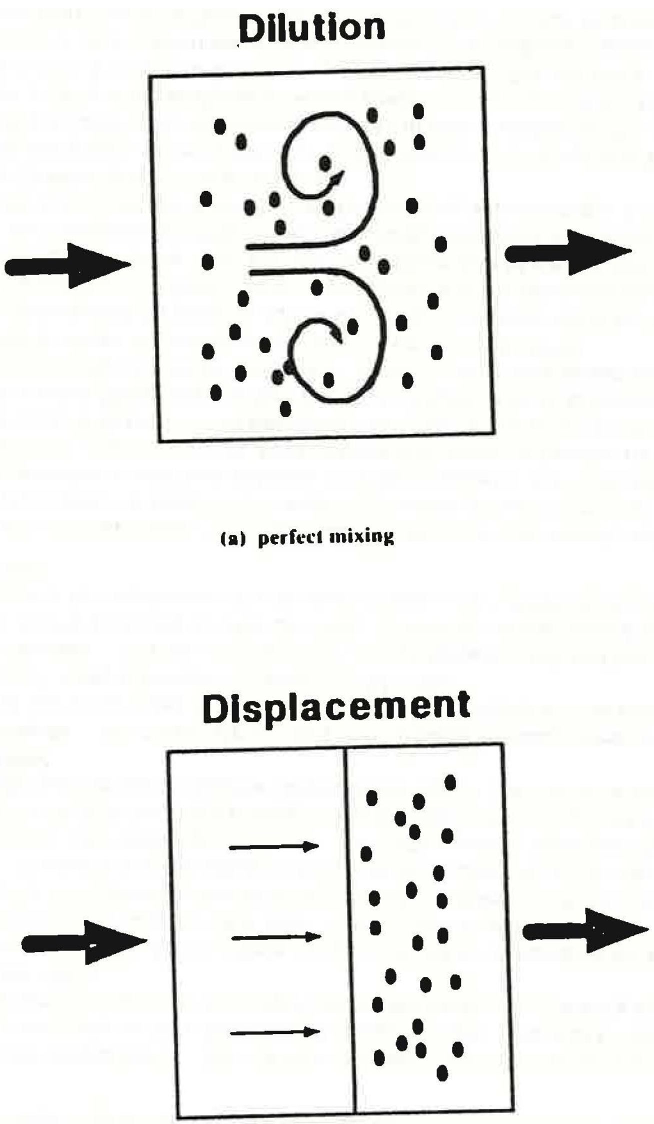


Figure 1 IAQ control functions provided by ventilation systems

Dilution



(a) perfect mixing

Displacement

(b) perfect displacement

Figure 2 Idealized ventilation strategies

In both mixing and displacement ventilation systems, system performance is degraded if the supply air is allowed to short circuit directly to the return duct *without* removing pollutants from the room. Reductions in system performance can also be produced by uneven distribution of the supply air, causing localized recirculation zones that are not exposed to the primary airflow.

Nevins and Miller (1972a, 1972b) have studied the role of ventilation in providing thermal comfort. They varied the configuration of supply and return ducts in a full-scale test room and quantified the comfort level in the room by defining an "effective draught temperature" based upon the local flow velocity and temperature. The performance of a specific ventilation configuration was measured by defining the Air Distribution Performance index (ADPI) to be "the number of measuring positions, uniformly distributed in a test plane of space, at which the comfort criteria are satisfied, expressed as a percent of the total number of positions at which measurements of temperature and velocity are made." The ADPI was found to depend upon the *isothermal* throw length of the supply diffusers and the thermal load in the room. Nevins and Miller demonstrated that a high ADPI could be achieved if the throw length of the diffusers was the same order of magnitude as the characteristic dimensions of the room.

A number of standards have been created to determine the isothermal throw of diffusers for use in ADPI design calculations (ADC 1984; ASHRAE 1987b; ISO 1984). However buoyancy caused by temperature differences can have a significant effect upon diffuser performance. Int-Hout and Weed (1988) have recently written an overview of the ADPI design method, including rules of thumb to correct throw for nonisothermal conditions. There are at present no design standards to rate the pollutant removal performance of diffusers and return ducts for either isothermal or non-isothermal conditions. Ventilation guidelines that provide thermal comfort do not necessarily guarantee a high level of indoor air quality. This is particularly true as improvements in building thermal design reduce thermal loads and infiltration. As the ventilation rates that are required to maintain thermal comfort are reduced, the mixing and dilution produced by the ventilation system is also reduced.

RESEARCH APPROACH

A number of different methods have been suggested to evaluate the performance of ventilation systems (Janssen, et al. 1982; Persily 1985; Skaret 1984). The method used in the present study was developed using the following criteria:

1. The method must be applicable to a broad range of geometries and pollutant types so that quantitative comparisons of performance can be made between ventilation systems in a wide range of applications,

2. The method must allow performance evaluations based upon short term measurements so that the method is cost effective and easy to use,

3. The method must have direct physical significance so that results of measurements can be used as diagnostic tools to determine appropriate mitigation strategies for systems with low performance, and

4. The method must decouple the air distribution characteristics of the ventilation system from the pollutant transport characteristics of the ventilation system so that the flow and transport characteristics can be evaluated separately.

These criteria are closely linked to the objectives of this research project and are based upon both fundamental and practical considerations.

Ventilation Efficiency Model

Previous studies of ventilation performance have established that flow nonuniformities and pollutant source variations can create localized zones within buildings that are not well ventilated even though the overall building ventilation rate meets acceptable standards. These nonuniformities are caused by two effects: (1) only a fraction of the volume of a room is directly ventilated (Figure 3), and (2) only a portion of the pollutant generated by the source is removed by ventilation air before it can mix with room air (Figure 4). These effects are quantified in the present study by the introduction of two efficiency measures: the displacement efficiency, η_d, and the removal efficiency, η_r.

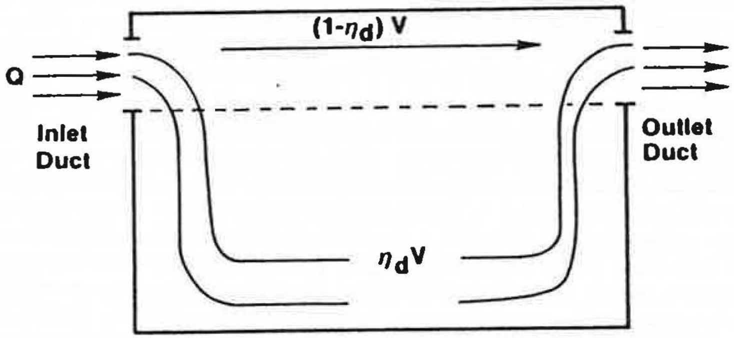


Figure 3 Relationship between directly ventilated region of room and η_d

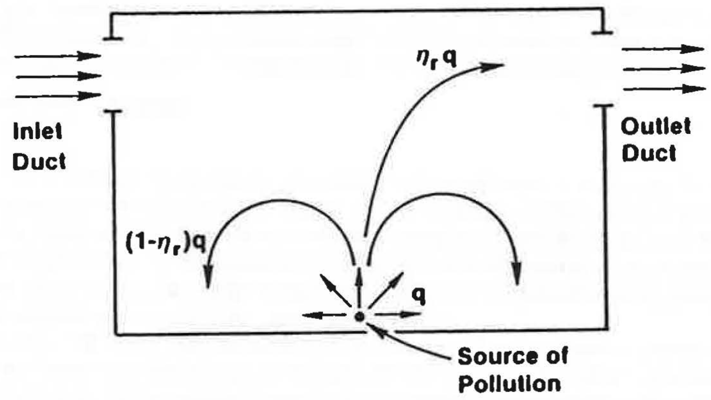


Figure 4 Relationship between pollutant removal produced by ventilation system and η_r

The displacement efficiency is defined to be the fraction of room air that is displaced by the ventilation system during the time that one volume change is supplied to the room. For a steady flow system, the displacement efficiency is

$$\eta_d = \left(\frac{\bar{c} - c_o}{c_{in} - c_o}\right)_{t=\tau} = \frac{1}{\tau}\int_0^\tau \left(\frac{c_{in} - c_{out}}{c_{in} - c_o}\right) dt \qquad (1)$$

with the initial conditions

$$c = c_o,\ t \leq 0 \qquad (2a)$$

$$c_{in} > c_o,\ t > 0 \qquad (2b)$$

In Equation 1, c is the average concentration of pollutant in the room, c_o is the initial concentration in the room, τ is the time required to supply one air change to the room, c_{in} is the concentration in air supplied to the zone, and c_{out} is the concentration in air removed from the zone. The maximum value that the displacement efficiency can have is 1 and the minimum value is 0. For a piston flow system, $\eta_d = 1$; while for a perfectly mixed system, $\eta_d = 0.67$.

The removal efficiency is defined to be the fraction of pollutant introduced into a uniformly mixed zone that is removed after one air change is supplied to the zone. For a steady flow system with a volumetric pollutant source the removal efficiency is

$$\eta_r = \left(1 - \frac{\bar{c} - c_o}{q/Q\, c_s}\right)_{t=\tau} = \frac{1}{\tau}\int_0^\tau \frac{[c_{out}(1 + q/Q) - c_o]\,Q}{q\, c_s}\, dt \qquad (3)$$

with the test conditions

$$c_{in} = c_o,\ c_s = 0 \quad t \leq 0 \qquad (4a)$$

$$c_{in} = c_o,\ c_s > 0 \quad t > 0 \qquad (4b)$$

In Equation 3, c_s is the concentration supplied by the pollutant source, q is the volumetric flow rate of the pollutant source and Q is the volumetric flow rate of the ventilation system. It has been assumed during the derivation of Equation 3 that the concentration in the air supplied by the ventilation system during the period of integration is equal to the initial concentration, c_o, in the room. The removal efficiency, like the displacement efficiency, is constrained by definition to have a value between 0 and 1. The removal efficiency of a perfectly mixed system is 0.37, and the removal efficiency of a piston flow system with a uniformly distributed source is 0.5.

The displacement efficiency and removal efficiency defined by Equations 1-4 are based upon the response of the ventilation system to a step function input. Once this response is determined from experimental measurements, the response of the system to an arbitrary input function can be determined through the use of Duhamel integrals (Wylie 1975).

The displacement efficiency is independent of pollutant source characteristics and measures the size of the subregion in a room that is directly affected by the ventilation system. The removal efficiency depends upon pollutant source characteristics and provides a measure of the ability of the ventilation system to remove pollutants *before* they are mixed with room air. Because of this difference in sensitivity to pollutant characteristics, small values of η_d do not necessarily imply small values of η_r. The removal efficiency, η_r can still be large when η_d is small, provided that the pollutant sources are located in the actively ventilated subregion of the room. Because of the sensitivity of η_r to pollutant source type and location, the ventilation model described by Equations 1-4 can be used to determine ventilation performance of pollutant source type.

The impact of variations in η_d and η_r on ventilation performance is summarized in Figure 5. In general, increases in η_r at constant η_d reduce pollutant concentrations without increasing ventilation rates, and reductions in η_d at constant η_r reduce ventilation rates without increasing pollutant concentrations.

A two-dimensional ventilation efficiency plot such as that shown in Figure 6, can be used as a diagnostic tool to evaluate the performance of a particular ventilation system. The most efficient ventilation systems for the control of indoor air pollutants are those that have large removal efficiencies and the smallest displacement efficiencies that are consistent with supply of ventilation air to the occupied subregion of the zone.

Laboratory-Scale Test Cell

Three levels of research are required to fully understand ventilation performance in buildings. These levels are laboratory-scale research, full-scale research, and numerical research. The advantages and disadvantages of each type of research are listed in Table 1. In many cases, the advantages of one level of research complement the shortcomings of another. Full-scale research is required for problem definition and solution verification. Small-scale and numerical research can be cost-effective methods for obtaining research results leading to the development of solutions to full-scale problems.

This paper describes the results of tests conducted in a unique laboratory-scale test facility. The test facility has been designed to provide direct visualization and measurement of pollutant transport processes, and serve as a test bed for innovative solutions to air management problems faced by the HVAC industry. The results from the small-scale facility will be combined with the results from full scale tests and numerical tests to provide a comprehensive understanding of pollutant transport in buildings.

The small-scale test cell has been designed to accurately model airflows in full-scale buildings, based upon the principles of dynamic similitude first introduced by Osborne Reynolds in 1883 (Schlicting 1979). Because of the difference in kinematic viscosity between water and air, water can be used in a ¼ scale test cell to reproduce the Reynolds numbers that occur in full scale buildings. Water can also be used to study buoyant flows in buildings (Bohn and Anderson 1984), but initial testing has been limited to isothermal conditions. Scale modeling techniques can also be applied to air-filled test cells with some limitations (Kiobut, et al. 1987). A unique advantage of a fluid filled test cell is that it allows room dimensions and velocity to be scaled by equal amounts, preserving the same convective time scales that occur in a full-scale building.

A schematic of the test cell is shown in Figure 7. The test section is 0.44 m (17.5 in) high, 0.76 m (30 in) long, and 0.38 m (15 in) wide. An even distribution of the flow is ensured by using flow straighteners before the flow enters the test section and after the flow leaves the test section. The locations of the flow inlet and outlet can be varied to study the impact of duct location and orientation on ventilation efficiency. The duct configuration can also be varied to simulate the throw and entrainment characteristics of various types of diffusers. A pump is used to circulate the flow through the test cell. A heat exchanger and precision chiller are used to control the temperature of the flow that enters the test cell and maintain isothermal conditions.

Velocity measurements in the test cell are made with a Laser Doppler Anemometer System (LDA). The LDA is mounted on a three-dimensional traverse. One beam of the

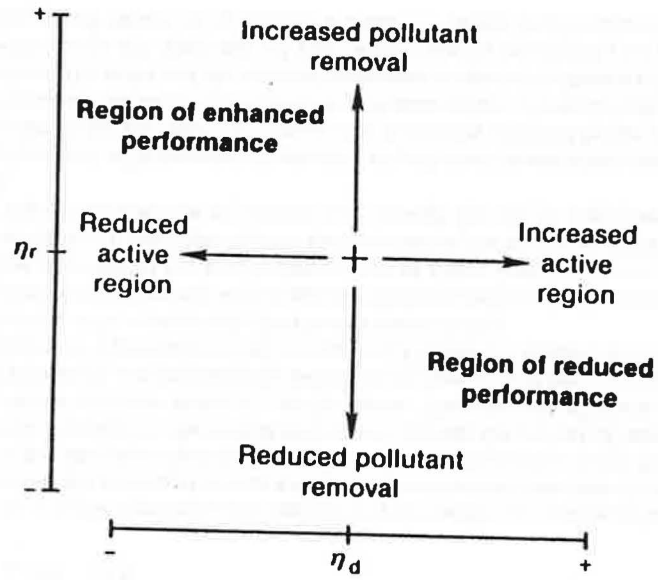


Figure 5 Impact of η_d and η_r on HVAC system performance

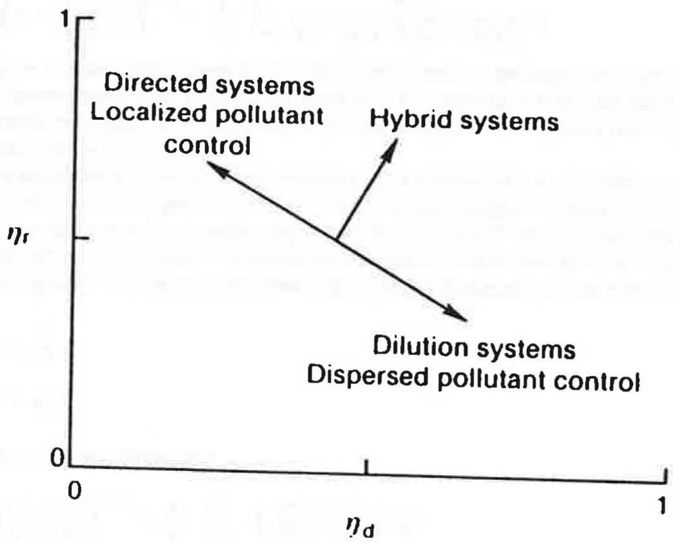


Figure 6 Types of ventilation systems

TABLE 1

Research Approaches

Advantages	Disadvantages
Full-Scale Research	
Phenomena of interest can be studied in real buildings. (Essential for initial problem definition and validation of final results.)	Boundary conditions are difficult to modify and control.
	The presence of a large number of uncontrolled parameters makes it difficult to determine cause and effect relationships.
Laboratory Scale Research	
High level of control over boundary conditions.	Difficult to exactly match all of the complexity of a full-scale building.
Easily modified.	
Relatively easy to collect large amounts of highly accurate data.	
Numerical Research	
Simple two-dimensional, laminar flow problems can be solved quickly and inexpensively.	Three-dimensional calculations can be expensive.
	Existing turbulence models have not been adequately validated for building applications.

LDA is frequency-shifted with respect to the other to allow the detection of flow reversals and to increase accuracy in low-velocity flows. The front and back of the test cell are constructed of glass to allow optical access to the test section. The LDA is operated in forward scatter mode with tracker-type signal processing and can measure velocities to within 0.03 cm/s. A comparison between velocity measurements made of a wall jet in the test cell and previous full scale measurements by Bajura and Szewczyk (1970) is shown in Figure 8. The close agreement between the two sets of measurements confirms the accuracy of the scaling approach used during the present experiments.

The test cell has been designed to study volumetric pollutant sources such as cigarette smoke, respiration byproducts, combustion byproducts and radon gas. These pollutants are modeled by using suspended particles and pH indicator dyes. The pollutant source solution was injected into the test cell during the present tests through a hypodermic needle positioned at the desired source location. Sensors have been installed in the supply and return ducts to make the concentration measurements that are required to calculate η_d and η_r. A microcomputer-based image analysis system is being developed to measure *local* pollutant concentrations as a function of time. An example of a concentration distribution measured with this system is shown in Figure 9.

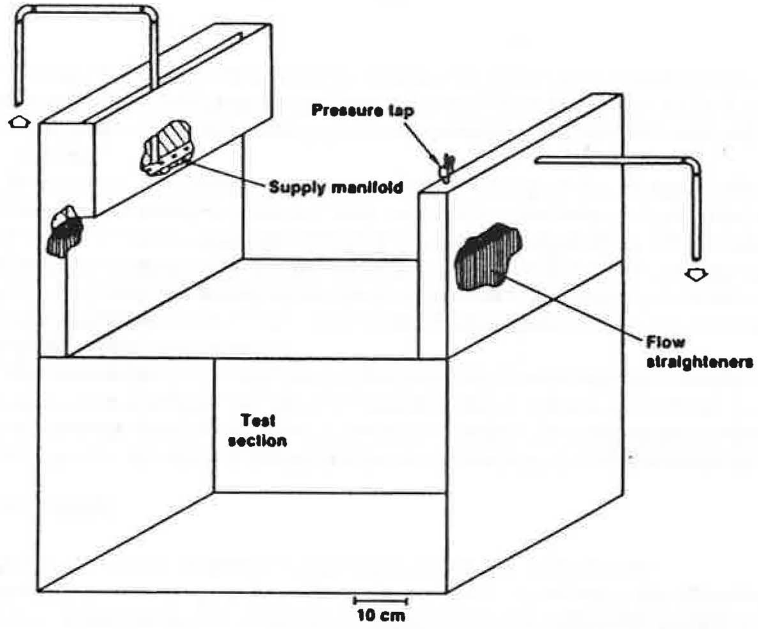


Figure 7 Schematic of ventilation test cell

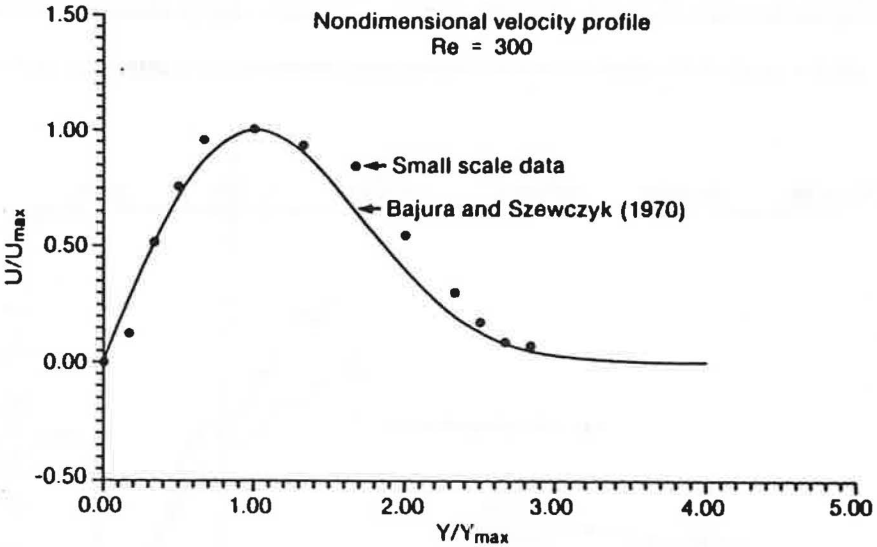


Figure 8 Comparison between full-scale and small-scale velocity distributions

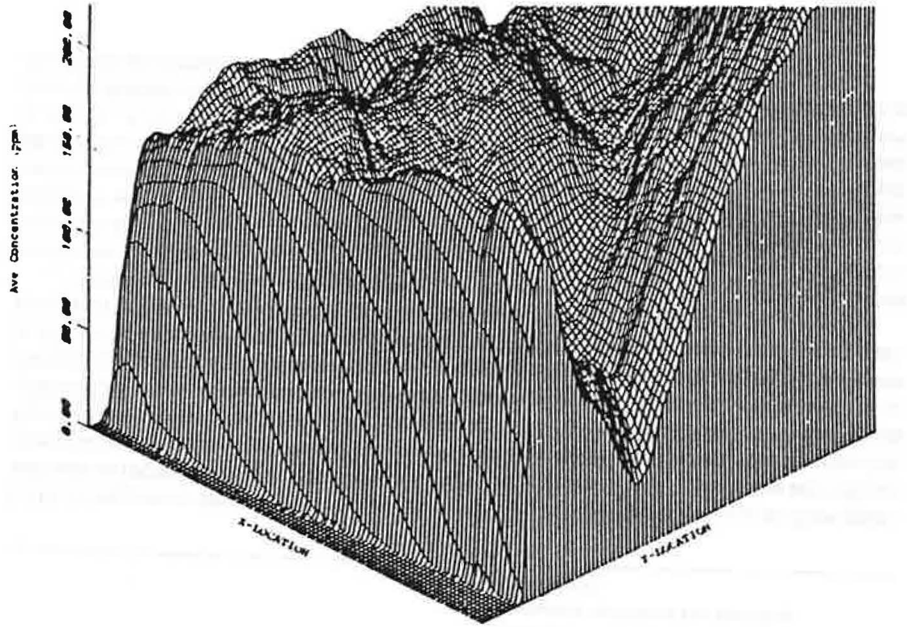


Figure 9 Digitized image of pollutant distribution in test cell

RESULTS OF SCALE VENTILATION EFFICIENCY TESTS

Experiments were conducted to determine the effect of return duct location and pollutant source location upon the displacement efficiency and removal efficiency in a room with a floor supply, such as is common practice in residential applications. To simplify the geometry for testing purposes, the supply and return openings were extended across the entire width of the test cell. The supply vent was located horizontally at the bottom of the left-hand end wall. The return vent was located at either the top or the bottom of the right hand end wall. The flow rate during the tests was fixed at 10.3 volume changes per hour (corresponding to a Reynolds number of 1000). The pollutant source was a buoyant plume that was injected with a low velocity at either the top or the bottom of the test cell. Hydroxal ions produced by varying the pH of the fluid in the test cell were used to simulate pollutant concentrations.

During the displacement efficiency tests, a step change in concentration was applied to the supply flow, and the concentration was monitored as a function of time at the return duct. A plot of the outlet concentration as a function of time during the displacement efficiency tests is shown in Figure 10. The displacement efficiency was found to be relatively independentof return duct location. However, the shape of the concentration decay curves were significantly different. The floor return produced a more rapid decay, indicating more mixing than the ceiling return location.

Results of removal efficiency tests are shown in Figures 11 and 12 as a function of the location of the pollutant source. Figure 11 shows the results for the case of a ceiling

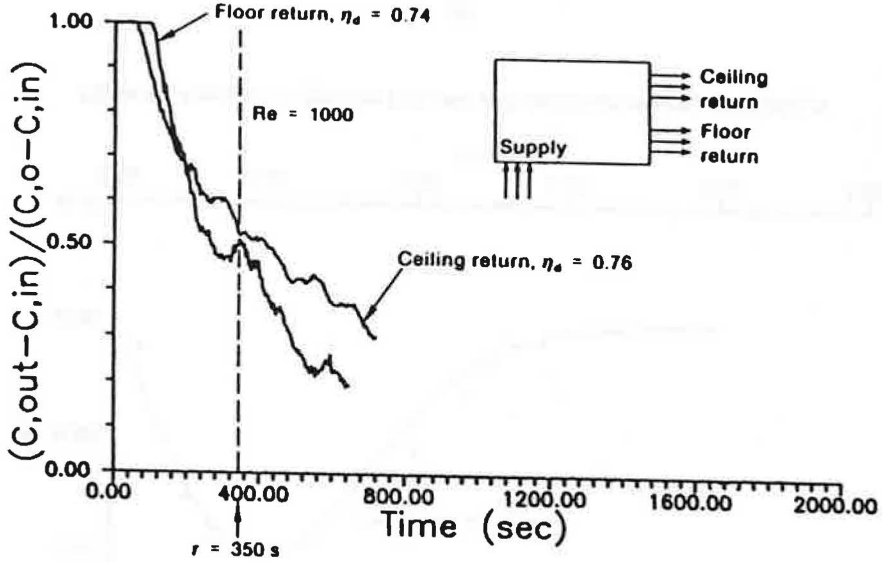


Figure 10 Outlet pollution concentration during replacement efficiency test (d/D = 0.11)

return. The removal efficiency shows very little change in Figure 11 when the source location is changed. Because the pollutant source is bouyant, the ceiling return is effective regardless of source location.

The same is not true when the return is located at the floor (Figure 12). The removal efficiency is still high when the source is near the floor, because of direct removal by the floor return. However, the removal efficiency was reduced dramatically when the source was moved to the ceiling. The ceiling source tended to get trapped in the upper portion of the test cell rather than being directly removed by the floor return.

CONCLUSIONS

A unique laboratory-scale test facility has been developed that reproduces the flow patterns in a full-scale room with the same boundary conditions. The test facility has been equipped for flow visualization, velocity, and concentration measurements. The flow supply and return can be reconfigured to study the effect of ventilation system design on indoor pollutant concentrations.

A ventilation performance model is proposed that allows ventilation systems to be evaluated based upon the results of simple transient tests. The performance model provides separate measures of air displacement efficiency and pollutant removal efficiency. The performance model has been designed to serve as a diagnostic tool to determine the reasons for low ventilation system performance and is expected to be useful in providing guidance for mitigation of existing systems, as well as aiding the design of advanced systems.

The performance model is used to evaluate ventilation performance during simple scale tests in a single zone with a bouyant point source of pollution. During the tests reported in this paper, the supply was located on the floor, and the return was located

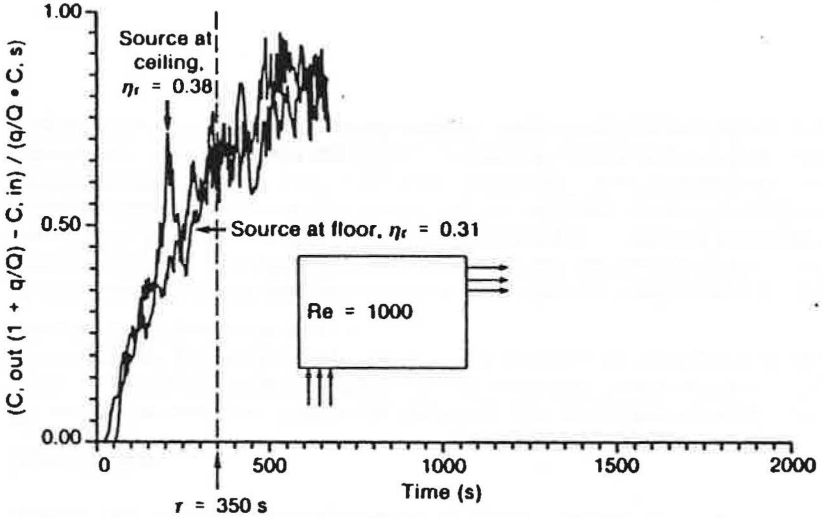


Figure 11 Removal efficiency test with ceiling return (d/D = 0.11)

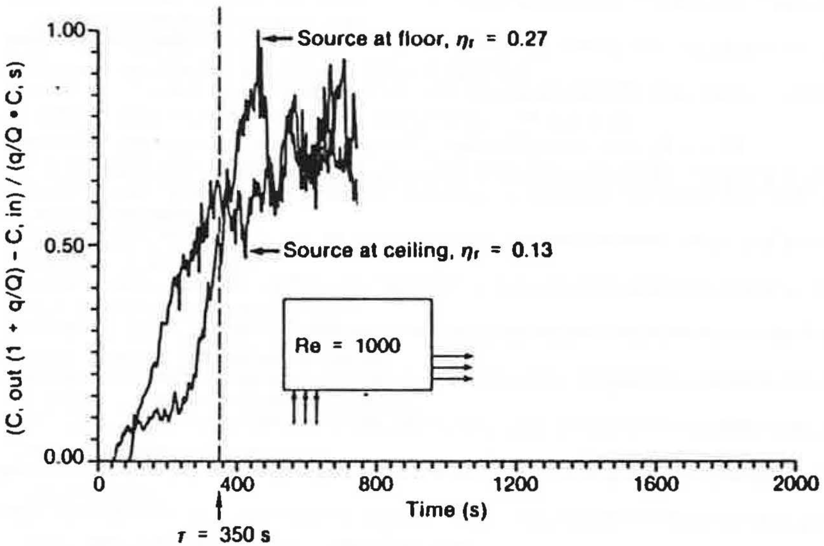


Figure 12 Removal efficiency test with floor return (d/D = 0.11)

either at the floor or at the ceiling. The displacement efficiency was found to be relatively independent of the return location. However, the return location was found to have a large effect on the removal efficiency, demonstrating that both supply and return location must be considered in evaluating the efficiency of ventilation systems.

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NOMENCLATURE

c = concentration, kg/m^3
d = duct width, m
H = room height
q = pollutant generation rate, m^3/s
Q = volumetric flow rate, m^3/s
U = Velocity of supply, m/s
V = Volume, m^3
Re = Reynolds number based upon duct width, Ud/ν

Subscripts

o = initial condition
out = return
in = supply
s = pollutant source

Greek

η_d = Displacement efficiency, Equation 1
η_r = Removal efficiency, Equation 3
ν = Kinematic viscosity, cm^2/s
τ = time to supply one volume change, V/Q

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DISCUSSION

H. Levin, University of California, Berkeley: For your removal studies, was your pollutant source continuous or episodic? Did you consider dynamics of the system related to stripping of pollutant by ventilation at the source or to changes in source partial pressures? Will you in the future?

R. Anderson: The pollutant source we are currently using undergoes a step change in concentration at $t = 0$ and then maintains that concentration during the remainder of the test. To date, we have limited our study to simple volumetric pollutant sources representative of environmental tobacco smoke and respiration. Our scale modeling techniques could also be used to study mass transport of pollutants from surfaces by convection. Once we understand the dynamics of mass transport from volumetric sources, it would make sense to include sources whose strength is coupled to room airflow.

Predicting Velocity and Contamination Distribution in Ventilated Volumes Using Navier-Stokes Equations

R.H. Horstman, P.E.

ABSTRACT

A method has been developed that predicts the velocity distribution, airflow circulation pattern, and airborne contamination distribution within a ventilated volume.

The ventilated volume in this case is an aircraft passenger cabin, but this method can be applied to other volumes such as buildings or enclosures.

The method utilizes finite difference stream functioning equations and modified Taylor series approximations of the vorticity equations to solve the Navier-Stokes equations in two dimensions.

The velocity distribution, streamlines, and flow pattern are derived from this solution and are used for the basis of establishing the contamination propagation and distribution.

The contaminant (in this case, carbon dioxide) is transported between nodes of the finite difference array by two mechanisms: convection and diffusion. The carbon dioxide laden air exits the subject's mouth and passes node by node throughout the volume ultimately attaining a steady-state concentration in each node.

This concentration variation between nodes represents the approximation of how the contaminant is distributed throughout the volume. This can be used as a tool to develop optimum ventilation schemes and to estimate ventilation effectiveness.

The velocity distribution prediction has been verified by test, and the contamination levels in exhaust ports compare well with the mixing equation.

INTRODUCTION

The problem of predicting the effectiveness of ventilation systems has been difficult in the past, primarily because of the complex interaction of the system components within the ventilated space.

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