Title
VENTILATION EFFICIENCIES OF WALL- OR WINDOW-MDUNTED RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS

Author
Offermann, F.J.

Publication Date
1983-05-01

Lawrence Berkeley Laboratory

UNIVERSITY OF CALIFORNIA

ENERGY & ENVIRONMENT DIVISION

Presented at the 1983 ASHRAE Annual Meeting,
Washington, DC, June 26-30, 1983

VENTILATION EFFICIENCIES OF WALL- OR WINDOW-MOUNTED RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS

F.J. Offermann, W.J. Fisk, D.T. Grimsrud,
B. Pedersen, and K.L. Revzan

May 1983

ENERGY
AND ENVIRONMENT
DIVISION

Prepared for the U.S. Department of Energy under Contract DE-AC03-76SF00098

DISCLAIMER

To be presented at the
1983 ASHRAE Annual Meeting,
June 26-30, Washington, D.C.

LBL-15376
EEB-Vent 82-15

VENTILATION EFFICIENCIES OF WALL- OR WINDOW-MOUNTED RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS

Francis J. Offermann, William J. Fisk, David T. Grimsrud,
Brian Pedersen, and Kenneth L. Revzan

Building Ventilation and Indoor Air Quality Program
Lawrence Berkeley Laboratory
University of California
Berkeley, CA 94720

May 1983

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Building Energy Research and Development, Buildings Systems Division of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098, and the Bonneville Power Administration, Portland, Oregon.

VENTILATION EFFICIENCIES OF WALL- OR WINDOW-MOUNTED RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS

Francis J. Offermann, William J. Fisk, David T. Grimsrud,
Brian Pedersen, and Kenneth L. Revzan

ABSTRACT

Mechanical ventilation systems with air-to-air heat exchangers can be installed into residences to provide energy-efficient supplementary ventilation for the purpose of controlling indoor concentrations of contaminants, odors, and moisture. Wall- or window-mounted units have become particularly attractive because they are relatively inexpensive and easy to install. However, because they lack an air-distribution system, concern has arisen over their ventilation performance. To address this concern, a series of experiments was conducted on two different models of wall- or window-mounted heat exchangers in two multi-room research facilities. The nominal ventilation efficiencies of these units have been determined by measurement of tracer-gas decay rates at several indoor locations to be in the range of 0.44 to 0.65. No significant correlations between nominal ventilation efficiency and heat exchanger model or operational strategies were observed. Significantly higher local ventilation efficiencies were noted in the rooms where the heat exchangers were operating. Some preliminary tests indicate that internal leakage between the airstreams contributes significantly to the ventilation inefficiency of these systems.

Keywords: air-to-air heat exchanger, cross-stream leakage, energy conservation, Indoor air quality, mechanical ventilation, residential buildings, ventilation efficiency.

INTRODUCTION

In the United States, most of the ventilation of residential buildings is a result of infiltration -- the natural leakage of air through cracks in the building envelope and natural ventilation through open windows and doors. A significant amount of energy is required to heat or cool this ventilation air. In existing homes, conservation measures such as weatherstripping and caulking can reduce infiltration and thus save energy (Dickinson et al. 1982; Harrje and Mills 1978). In new homes, incorporation of special weatherization components (e.g., installation of continuous polyethylene vapor barriers in walls and ceilings, installation of weatherstripping, and sealing of joints and penetrations through the building envelope) can substantially reduce air leakage (Offermann et al. 1981; Beach 1978).

Francis J. Offermann, William J. Fisk, David T. Grimsrud,
Brian Pedersen, and Kenneth L. Revzan
Building Ventilation and Indoor Air Quality Program,
Lawrence Berkeley Laboratory,
University of California,
Berkeley, CA 94720

One of the problems associated with reduced ventilation is that indoor humidity levels and concentrations of indoor-generated air contaminants are often increased (Traynor et al. 1981; Hollowell and Miksch 1981; Nero 1981). The concentration of any indoor air contaminant is determined by its rate of emission into (source strength) and by its rate of removal from the indoor air. One of the primary removal mechanisms for indoor-generated air contaminants is the dilution and flushing that occurs when outdoor air leaks into the house and replaces indoor air. When the rate of air leakage is decreased the removal rate of indoor-generated contaminants is reduced, leading to higher indoor contaminant concentrations. Elevated concentrations of indoor air contaminants can also occur in residences where infiltration rates are normal but contaminant source strengths are high, and, in fact, the variation in contaminant source strengths among residences is greater than the variation in infiltration rates.

One energy-efficient solution to many indoor air quality problems is to install a mechanical ventilation system with an air-to-air heat exchanger (MVHX system, often referred to as a residential heat exchanger). Such systems provide a controlled supply of ventilation air and recover much of the energy that would be lost if the ventilation had occurred without heat recovery. A residential heat exchanger generally consists of a core, two fans, and two filters installed in an insulated case (figure 1). One fan brings outdoor air (supply air) through the core and into the house, while the second fan causes an equal amount of house air (exhaust air) to pass through the core and out of the house. As the two airstreams pass through the core, heat is transferred from the warmer to the cooler airstream (without mixing); thus, during a heating season, the supply air is warmed before entering the house.

Currently little information is available on the performance of MVHX systems under the actual operating conditions found in residences. Laboratory tests (Fisk, Roseme, and Hollowell 1980; Fisk et al. 1981; Persily 1982) indicate that residential MVHX systems can preheat or precool ventilation air by 45 to 85 percent of the difference between indoor and outdoor temperatures. Various field studies in occupied houses including one in nine occupied Rochester, NY, residences (Offermann 1981) indicate that MVHX systems are effective in reducing elevated indoor contaminant concentrations. In general, however, little information is available as to the efficiency with which these systems ventilate homes.

Most MVHX systems are used with a duct system for air distribution (figure 2a). The supply ductwork carries outdoor air to the exchanger and then distributes it to various locations throughout the residence. (In many houses, the furnace duct system can be used for a portion of the supply ductwork.) The exhaust ductwork carries house air to the heat exchanger and then out of the house. Some MVHX systems, as shown in figure 2b, are designed to be mounted through a wall or window. These units are similar in size to small window-mounted air conditioners, require no external ductwork, are relatively inexpensive, and are easy to install.

The performance of MVHX systems that are used without ductwork in ventilating a structure has received little study. Because the air exits and enters these heat exchangers at locations in close proximity, recirculation is possible (i.e., air exiting from the exchangers at locations interior and exterior to the building envelope may be entrained into the corresponding airstreams entering the exchangers). As with all air-to-air heat-transfer equipment, significant leakage between airstreams within the core is another possibility. Finally, the room in which the heat exchanger is installed may be ventilated more rapidly than other rooms within the structure (i.e., the rate of mixing between rooms may be slow compared to the mechanical ventilation rate). All these factors can reduce the effectiveness at which the heat exchangers remove indoor air contaminants.

To investigate the efficiency with which wall- or window-mounted heat exchangers ventilate indoor spaces and thus reduce elevated indoor contaminant concentrations, the Lawrence Berkeley Laboratory (LBL) has conducted a series of experiments to determine the ventilation efficiencies of these systems.

In this report, ventilation efficiency and other parameters relevant to the ventilation performance of these exchangers are discussed, the measurement techniques used are described, and the results of measurements made on two different models of wall- or window-mounted MVHX systems, for several different operating configurations, and in two different test facilities, are presented.

VENTILATION PERFORMANCE PARAMETERS

Ventilation is the process of supplying or removing air by natural or mechanical means to or from a specific space for the purpose of heating, cooling, and controlling the levels of moisture, odors, and indoor air contaminants. The performance of the ventilation system is constrained by at least two factors: (1) discomfort due to excessive air movement and/or noise and (2) equipment and operating costs. The purpose of a residential MVHX system is to provide sufficient ventilation to control indoor levels of moisture, odors, and contaminants. A separate system (e.g., furnace or air-conditioner) is generally used in residences for heating or cooling.

To assess their ventilation performance, the MVHX systems were installed in two multi-room structures and a series of tests performed. (A more detailed description of the structures and the tests is provided later.) A tracer gas was injected into the structures and the indoor air was mixed to establish a uniform initial concentration. The tracer-gas concentration was then measured as a function of time at six indoor locations and in the supply and exhaust airstreams of the heat exchanger. These test data were used to calculate a number of performance parameters including (1) tracer-gas decay rates, (2) local air exchange rates, and (3) ventilation efficiencies. Each term is defined below.

Tracer-Gas Decay Rates

Tracer-gas decay rates are determined by fitting the measured data for tracer-gas concentration versus time to an equation of the form

$$C(t) = C(0)e^{-\lambda t} \tag{1}$$

where

$C(t)$ = the tracer-gas concentration at time t
$C(0)$ = the tracer-gas concentration at time 0
λ = the tracer-gas decay rate
t = a time variable

Solving for λ yields

$$\lambda = \frac{1}{t} \ln \frac{C(0)}{C(t)}. \tag{2}$$

If the indoor air is perfectly mixed, the tracer gas is nonreactive and not present in outdoor air, then the decay rate, λ, corresponds to the air-exchange rate, i.e., the rate at which indoor air is replaced by outdoor air. However, if the indoor air is not perfectly mixed, the parameter λ, based upon measurements at some indoor location, cannot be considered the local air-exchange rate. Skaaret (1981), Sandberg (1981), Malstrom (1981), and others have shown that with imperfect mixing of indoor air and an initially uniform tracer-gas concentration, the decay rate initially varies from location to location but eventually attains the same value at all locations. However, the concentrations of tracer gas at different indoor locations become unequal and after a uniform decay rate is established, the ratio of any two concentrations is constant. Areas with the lowest concentrations are the zones receiving the greatest amount of ventilation.

Local Air-Exchange Rates

When the indoor air is not perfectly mixed a local air exchange rate can be calculated. The concept of a local air exchange rate (and local ventilation efficiency) was first introduced by Sandberg (1981). In this paper, however, a method recommended by Skaaret is used to introduce the local air exchange rate. For a small, perfectly mixed volume element in an imperfectly mixed indoor space, the mass balance equation for the element is

$$\frac{dC}{dt} = -C\,\frac{dQ}{dV} \tag{3}$$

where

C = the concentration of tracer gas in the element
dV = the volume of the element
dQ = the flow rate of fresh (tracer-free) outdoor air into the element
t = a time variable

The fresh air-flow rate, dQ, is an artificial quantity. Actually, the air entering the element is a mixture of outdoor and indoor air, and the quantity dQ is the flow rate of 100% outdoor air that would cause the observed rate of change in concentration. If the local air exchange rate at point j, n^j, is defined to be dQ/dV and assumed to be constant with respect to time, equation 3 can be integrated to yield an expression for the local air exchange rate

$$n^j = -\int_{C^j(0)}^{C^j(t)} dC \Big/ \int_0^t C^j\,dt \tag{4}$$

where

n^j = the local air exchange rate at point j
$C^j(0)$ = the tracer concentration observed at point j at time t=0
$C^j(t)$ = the tracer concentration observed at point j at time t

In these experiments the local air exchange rates were calculated for each point as the measured change in tracer concentration divided by the area under the concentration curve, C(t), which was numerically integrated over a period of one hour. The local air exchange is an indicator of the amount of ventilation that occurs at each location. Comparing local ventilation rates from location to location indicates how ventilation air is distributed throughout the space that is ventilated.

Ventilation Efficiencies

Ventilation efficiencies relate the observed concentrations, decay rates, or local air exchange rates to predictions for a reference case. Calculations here were compared to the case when the indoor air is perfectly mixed, and no recirculation or leakage occurs between airstreams. It should be noted, however, that perfect mixing is not always the optimal condition but only serves as a convenient reference case. In many applications it is desirable to ventilate only a specific region, e.g., the zone of occupation or the region near a concentrated pollutant source.

Based upon measurements, a nominal ventilation efficiency is defined using the equation

$$E_1 = \frac{(\lambda - x\lambda_*)}{Q/V} \tag{5}$$

where

E_1 = the nominal ventilation efficiency
V = the volume of the structure ventilated
Q = the rate of airflow through the heat exchanger
λ = spatial average of the six indoor tracer-gas decay rates when the heat exchanger is operating
λ_* = spatial average of the six indoor tracer-gas decay rates when the heat exchanger is not operating
x = a correction factor to account for air leakage through

the heat exchanger when installed but not operating (this will be described more fully later)

This nominal ventilation efficiency relates the spatial average increase in tracer-gas decay rate with operation of the MVHX system, to the increase that would occur in the reference case. It indicates how effectively the system provides ventilation to the space as a whole but provides no information on the distribution of ventilation. The spatial average decay rate is a volume-weighted average based on estimates of the volume associated with each individual measurement point.

A local ventilation efficiency can also be calculated for each indoor location based upon the local air-exchange rate using the equation

$$E_2 = \frac{(n^j - xn_*^j)}{Q/V} \tag{6}$$

<u>where</u>

n^j = the local air-exchange rate observed at point j with the heat exchanger operating

n_*^j = the local air-exchange rate observed at point j when the heat exchanger is not operating

This local efficiency relates the increase in local air-exchange rate with operation of the heat exchanger to the increase predicted for the reference case described above. Comparison of local ventilation efficiencies measured at different points indicates how the ventilation air is distributed throughout the test space.

In addition to the nominal and local ventilation efficiencies, a relative ventilation efficiency can be calculated for each location from the equation

$$E_3(t) = C_e(t)/C^j(t) \tag{7}$$

<u>where</u>

$E_3(t)$ = the relative ventilation efficiency

$C_e(t)$ = the average concentration of tracer gas in the airstream exhausted by the heat exchanger

$C^j(t)$ = the concentration of tracer gas at the indoor location

This relative ventilation efficiency compares the exhaust airstream concentration to the concentration at an indoor location. In the reference case, the exhaust concentration would equal the indoor concentration and all relative ventilation efficiencies would have a value of unity. Relative ventilation efficiencies are generally compared from point to point, however, a spatial average value can be calculated by substituting the algebraic average of the six volume-normalized indoor concentrations for $C^j(t)$. It is this spatial average relative ventilation efficiency that is reported in this paper.

Measurements of tracer-gas decay rate performed when the heat exchanger was not operating are affected by air leakage through the heat exchanger. To determine the impact on ventilation of installing and operating the exchanger, i.e., not just operating a previously installed unit, a correction factor, x, is used in equations 5 and 6. This factor corrects for air leakage through the nonoperating heat exchanger and has been calculated from measurements of airflow rate versus pressure difference through the heat exchangers (when not operating) and the test structures. (The technique employed is based upon a method commonly used for modeling residential infiltration and is described more fully by Sherman and Grimsrud [1980].) The data for flow rate versus pressure difference was fit to an equation of the form

$$Q = A_e \sqrt{\frac{2\Delta P}{\rho}} \tag{8}$$

<u>where</u>

Q = the airflow rate

A_e = the effective leakage area

ΔP = the pressure difference

ρ = the air density

Assuming a pressure difference of 4 Pa, which is typical of the pressure differences driving natural infiltration, the effective leakage areas calculated for each heat exchanger were 35 E-4 m^2 for Unit A and 25 E-4 m^2 for Unit B. The effective leakage areas of the two test structures without heat exchangers installed were 63 E-4 m^2 for the Richmond Research House and 290 E-4 m^2 for the Walnut Creek Research House. The correction factors, x, were then calculated for each case by dividing the leakage area of the structure with the sum of the leakage areas of the structure and the heat exchanger. The resulting correction factors were 0.64 and 0.72 for three-room tests at the Richmond Research House with Units A and B, respectively, and 0.89 for tests at the Walnut Creek Research House. For one-room tests at the Richmond Research House (described later), correction factors of 0.65 and 0.75 were determined for Units A and B, respectively, based upon tracer-gas decay measurements with and without the heat exchanger sealed to prevent air leakage.

EXPERIMENTAL PROTOCOL

The following is a brief description of the MVHX systems, structures, instrumentation, and test procedures used in this study. The tests performed for this study are also described.

Description of Heat Exchangers Tested

As depicted in figure 3, Unit A is a crossflow-type heat exchanger which uses a heat-transfer surface, or core, made of a treated paper, that transfers moisture as well as heat between airstreams. Figure 4 illustrates Unit B, a rotary-type heat exchanger, which uses a rotating counterflow "heat wheel" core coated with a desiccant to transfer heat and moisture between airstreams. Both units are similar in size to small window-mounted air conditioners and contain two fans driven by a single motor to provide the flow of supply and exhaust air. Unit A has three fan speeds and Unit B has two fan speeds. Table 1 lists airflow rates and fan-power requirements for each heat exchanger as a function of their different fan speeds. The data for Unit A were determined from tests at LBL (Fisk et al. 1981), and the data for the Unit B were obtained from the manufacturer's published literature.

When installed, the housing of Unit A penetrates the window or wall. Unit B is installed inside the house on the surface of an exterior wall and has two ducts (80 mm in diameter) that penetrate to the outside. Both heat exchangers utilize weather hoods over the outside vents, which were installed for the tests described here.

Description of Structures and Locations of Heat Exchangers

Experiments were performed in two structures. The test space of the Richmond Research House, shown in figure 5, consists of three interconnected rooms with a total floor area of 54.2 m^2, a ceiling height of 2.36 m, and a total volume of 128 m^3. The structure has been renovated to assure low rates of air leakage through the building envelope. The heat exchangers were installed 0.9 m above floor level through a window in a central location (room 2) or through a door at an end location (room 3). To simulate installation of Unit B through a wall but allow it to be installed at the same locations as Unit A, Unit B was mounted on a small wooden box with the same thickness as a typical wall. The box was then installed through the window or door as required for the tests.

The Walnut Creek Research House, shown in figure 6, is a typical single-family, single-story dwelling with a total floor area of 90.0 m^2, a ceiling height of 2.44 m (except in a hall where the ceiling is 2.14 m high), and a total volume of 231 m^3. This building envelope has also been renovated to assure low rates of natural infiltration. The forced-air heating system includes a single return vent in the ceiling of the bedroom hall and floor-mounted supply vents distributed throughout the house. Two MVHX systems were installed in this house. One was mounted in the master bedroom 1.31 m above floor level through a window and another was mounted in the living room, 1.94

m above floor level through the wall.

Instrumentation

Ventilation was measured by the tracer-gas decay technique using sulfur hexafluoride (SF_6). The SF_6 tracer was introduced into the test space and mixed to establish an initially uniform concentration of approximately 50 ppm. The concentration was then measured versus time at eight different locations using two nondispersive infrared analyzers. The analyzers were calibrated at the beginning and end of each decay using three primary standard SF_6 calibration gases (10, 20, and 50 ppm). Two of the eight sampling points were used to sample from the heat exchanger airstreams. In order to obtain samples that would be representative of the average SF_6 concentration in the supply and exhaust airstreams, air was sampled through small tubular integrating manifolds centered in each airstream. The other six sampling points were located within the test space as depicted in figures 5 and 6. Air was sampled at two points simultaneously for a one-minute sampling period at a flow rate of 20 L/min and sequenced so that all eight points were sampled every four minutes.

At the Richmond Research House, tests were carried out with the aid of a microprocessor that controlled the injection of SF_6, operation of the mixing fans, operation of the heat exchanger, and calibration of the SF_6 analyzers. Throughout the tests, the computer also collected data from the SF_6 analyzers and environmental data. The temperatures of the test space, the supply airstream entering the test space, and the outside air were measured along with wind speed and wind direction. Tests were conducted at the Walnut Creek Research House without the aid of a computer control and data acquisition system.

During four tests at the Richmond Research House, portable electric baseboard heaters maintained a 20 ± 5°C temperature differential between the test space and the outside. Four one-kilowatt, 1.24 m-wide freestanding heaters were located under four windows in the test space, as shown in figure 5. Each heater was controlled separately by a thermostat mounted nearby on an inside wall.

Test Procedure

All tests at the Richmond Research House consisted of an alternate series of "natural" decays without operation of the MVHX system and "heat exchanger" decays with the MVHX system operating. At the Walnut Creek Research House, the natural decay rate was measured once with and once without the furnace fan operating and on the same day that the decay rates with the MVHX system operating were measured.

The sequence of operations performed for each test were as follows:

1. Calibrate SF_6 analyzers with primary standard calibration gases.
2. Start SF_6 injection and mixing fans.
3. Stop SF_6 injection when concentration reaches 50 ppm. Continue operation of mixing fans to distribute SF_6 uniformly throughout the test space.
4. Measure pre-decay SF_6 concentration at six locations in the well-mixed test space.
5. Stop mixing fans and begin operation of the MVHX system(s)
6. Begin monitoring the SF_6 concentration in air at eight different locations.
7. At completion of decay, stop MVHX system, start mixing fans, and measure post-decay SF_6 concentration at six locations in the remixed test space. (This step was performed for tests at the Richmond Research House only.)

Tests Performed

The first two tests at the Richmond Research House were one-room tests conducted in Room 2 with both doors to the room closed. Unit A was operated in the central location for the first one-room test and Unit B was operated in the same location for the second one-room test. Because of the small volume of the test space, the heat exchangers were operated at the low fan-speed setting to assure a sufficiently long measurement period.

The remaining eight tests at the Richmond Research house were conducted in the entire three-room test space. In these tests, all possible combinations of three variables -- heat exchanger model, heat exchanger location, and baseboard heater operation -- were used. The heat exchangers were operated with fan speeds typical of those selected by homeowners in a LBL field study, i.e., medium fan speed for Unit A (110 m^3/h) and high fan speed for Unit B (90 m^3/h).

For the four tests at the Walnut Creek Research House, combinations of two variables -- heat exchanger configuration and furnace fan operation -- were used. Two tests were conducted with a single Unit A heat exchanger located in the living room and operating at the medium fan-speed setting. For the other two tests, two Unit A heat exchangers, one in the living room location and the other in the master bedroom, were operated simultaneously at the low fan-speed setting. For one of the tests conducted with each heat exchanger configuration, the forced-air furnace system's fan was not operating and all vents were sealed. For the other tests, the vents were not sealed and the furnace system's fan was cycled on for ten minutes, then off for ten minutes throughout the decay. To observe the ventilation of a room separated from the rest of the test space, the door to bedroom No. 1 was closed during all of the tests at the Walnut Creek Research House.

RESULTS AND DISCUSSION

Tracer Gas Concentration Versus Time

Figure 7 is a semilog plot of SF_6 concentration versus time during test 11 for the six indoor sampling locations and two heat exchanger (supply and exhaust airstream) sampling locations. For this test, one Unit A heat exchanger was operating at medium fan speed and installed in the living room of the Walnut Creek Research House. The furnace's forced-air system was not operating and the ducts were all taped shut. As with all the tests in the Walnut Creek Research House, bedroom No. 1 was closed at the start of the tracer decay period. As can be readily perceived from this plot, the tracer gas decay rate in the closed bedroom is negligible. In fact, there is an apparent slight rise in tracer concentration in this room (an effect seen only for this test), which may be an indication that initial mixing of the tracer gas was not good. The remaining sampling points have similar tracer decay rates as indicated by the parallel lines. As expected, the sampling points at the end of the house farthest from the heat exchanger have tracer gas concentrations consistently higher (15-20%) than those points closer to the heat exchanger. The concentration of tracer gas in the heat exchanger's exhaust airstream is consistently lower than the concentrations measured at the six indoor locations. The ratio of the exhaust concentration to the average indoor concentration is approximately 0.78. The relatively low concentration of tracer gas in the supply airstream entering the house indicates that the internal cross-stream leakage from the exhaust to the supply airstream and the external recirculation from the exhaust to supply airstream outside the house is small. However, as will be discussed later, it is uncertain how well the measured airstream concentrations represent the average concentration in the airstreams.

Ventilation Efficiencies

The results of the ventilation efficiency measurements made in the Richmond Research House are compiled in table 2 and the results for tests performed at the Walnut Creek Research House are compiled in table 3. A number of measured and calculated indicators of ventilation performance are presented in these two tables. They include the nominal ventilation efficiency, the local ventilation efficiencies at six indoor locations, and the effective ventilation rate. The estimated uncertainty in nominal ventilation efficiency is ± 0.06 and ± 0.07 for measurements performed at the Richmond and Walnut Creek research houses, respectively. This estimate was made by considering the uncertainties associated with measurements of SF_6 concentration, structure volume, airstream flow rate, and the correction factor x in equations 5 and 6. A more detailed discussion of the uncertainty analysis is available in Offermann et al. (1982).

Nominal Ventilation Efficiency

Richmond Research House. The nominal ventilation efficiency for all ten tests performed at the Richmond Research House averaged 0.54 ± 0.05 (± one standard deviation). Nominal ventilation efficiency averaged 0.56 ± 0.05 for Unit B, which, based on our estimated uncertainty of ± 0.06 for a single test is not a significantly higher efficiency than the 0.52 ± 0.05 average observed for Unit A. The nominal ventilation efficiency for one-room tests averaged 0.63 ± 0.02, which is significantly greater than the average efficiency of 0.52 ± 0.03 for all three-room tests. One would expect better mixing and higher efficiencies in the one-room tests; however, this difference cannot be assumed to be caused entirely by the different geometries and sizes of the structures ventilated, because the heat exchangers operated at low fan speed for the one-room tests and at a medium fan speed for the three-room tests. The amount of recirculation and cross-stream leakage of air may depend on the fan speed.

The average ventilation efficiency for three-room tests with electric baseboard heaters operating equaled 0.51 ± 0.01, which is not significantly different from the 0.53 ± 0.04 average efficiency when no heaters were operated. Heat exchanger location also had no significant impact on the average nominal ventilation efficiency for three-room tests. Ventilation efficiency averaged 0.52 ± 0.04 for tests with centrally located heat exchangers and 0.52 ± 0.03 for tests with the heat exchangers in an end location.

An average ventilation efficiency can also be calculated from the tracer concentration measurements made during the pre- and post-decay mixup periods. The decay rate necessary to reduce the average pre-decay tracer concentration to the observed post-decay concentration was calculated using equation 2. The ventilation efficiency computed for all ten tests from the mix-to-mix effective decay rates averaged 0.54 ± 0.05, which compares very well with the 0.54 ± 0.05 average nominal efficiency calculated from the individual decay rates observed at six indoor locations. This close agreement indicates that the six measured indoor concentrations represent the spatial average indoor concentration fairly well.

Walnut Creek Research House. The nominal ventilation efficiency for the four tests performed at the Walnut Creek Research House averaged 0.59 ± 0.08. A comparison of the data for tests run without operation of the furnace fan yields a nominal ventilation efficiency of 0.63 for the test with one heat exchanger operating, which is not significantly different from the value of 0.66 measured with two heat exchangers operating. Operation of the furnace fan is associated with a small decrease in nominal ventilation efficiency from 0.66 to 0.61 for the tests with two heat exchangers operating, and a large decrease in nominal ventilation efficiency from 0.63 to 0.47 when only one heat exchanger was operating.

One explanation for these reductions in ventilation efficiency is that during the tests with the furnace fan operating a significant amount of the supply air from the heat exchanger installed in the east wall was entrained into the furnace return (not an unlikely scenario since the supply airstream of this heat exchanger was directed down the hallway where the furnace return is located). A significant amount of infiltration is associated with operation of the furnace fan (e.g. 0.17 ach), which is likely a result of leakage from the pressurized side of the distribution system to the outside air under the house. For this reason, any coupling of the heat exchanger supply airstream with the furnace return will result in reduced ventilation rates.

Local Ventilation Efficiencies

As described earlier, comparison of the local ventilation efficiencies at different indoor locations indicates how the ventilation air is distributed throughout the test space.

Richmond Research House. As expected, the highest local ventilation efficiencies were observed at points nearest to the heat exchanger. For the three-room tests with the heat exchanger operating in Room 3 (end location), the local ventilation efficiencies were highest at points 2 and 5, which were located in Room 3. The local efficiencies at these points averaged 0.61 for the four tests, which is 45% higher than the 0.42 average for points 1 and 3 in Room 2 and 27% higher than the 0.48 average of points 4 and 6 in Room 1. For the three-room tests with the heat exchanger operating in Room 2 (central location), the variance in the local ventilation efficiencies was much less pronounced than that observed in tests with the heat exchanger operating in the end location. For the four tests with the heat exchanger operating in Room 2, the local ventilation efficiencies of points 1 and 3 in Room 2 averaged 0.56, which is just 10% higher than the average of 0.51 observed in Room 3 and 8% higher than the average of 0.52 observed in Room 1.

For the one-room tests, sampling points were established at the center of four quadrants of equal area and on a plane 1.37 m above the floor. In addition, two sampling points were located on a vertical axis at the center of one quadrant, at point 4, 0.15 m above the floor, and at point 3, 0.15 m below the ceiling. Thus, the one-room tests constituted the only experiments with sample points at heights other than 1.37 m above the floor. For the one-room tests conducted with Unit A, the local ventilation efficiency was 0.67 for the point near the floor and 0.46 for the point near the ceiling, which are slightly higher and lower than the 0.60 ± 0.04 average efficiency calculated for the four points located in the middle of the airspace. For the one-room tests conducted with Unit B, the local ventilation efficiency was 0.61 at both the floor and ceiling sample points which is slightly lower than the 0.68 ± 0.04 average efficiency calculated for the four points located in the middle of the airspace.

Walnut Creek Research House. As was found in the Richmond Research House tests, the highest local ventilation efficiencies were observed at points nearest to the heat exchanger(s). For test 11, where one heat exchanger was operated in the living room, the highest local ventilation efficiency was observed at point 1 in the living room. For test 12, where two heat exchangers were operated, one in the living room and one in the master bedroom, the highest local ventilation efficiencies were observed in these two rooms. In both of these tests, the lowest local ventilation efficiencies observed were at point 6 in bedroom No. 1 which was isolated from the rest of the test space by a closed door. For tests 13 and 14, which were replicates of tests 11 and 12, but with the furnace fan operating, the variance in the local ventilation efficiencies was reduced, which indicates that distribution of the ventilation air was improved. With the furnace fan operating, the lowest local ventilation efficiencies were still observed in the closed bedroom No. 1, however, the local ventilation efficiency of this room improved from essentially zero to an average of 0.32 with the furnace fan on, i.e., approximately one-half the average local ventilation efficiency observed at points in the open rooms.

Relative Ventilation Efficiencies

In addition to monitoring the tracer concentration at six indoor locations, the tracer concentration was measured in the exhaust and supply airstreams at points where the airstreams exited the heat exchanger case. Small, two-axis, multipoint sampling manifolds were used to sample the tracer concentrations in the heat exchanger airstreams. The degree to which these measurements represented the true average tracer concentration of the airstreams is not known. The tracer mass balance ratios, calculated from the airstream flow rates and airstream tracer concentration measurements made on all four sides of the heat exchanger (in each airstream and on each side of the heat exchanger core), indicate significant measurement error. Despite this uncertainty the relative ventilation efficiencies calculated using these data are discussed briefly.

The average relative ventilation efficiency as calculated by the ratio of the exhaust concentration to average indoor concentration was 0.78 ± 0.02 for Unit A and 0.84 ± 0.05 for Unit B. However, since significant contamination of the supply airstreams of both heat exchangers was observed, it was decided to calculate the relative ventilation efficiency as the ratio of the net difference between exhaust and supply airstream concentrations to the average indoor concentration. This average relative ventilation efficiency corrected for supply stream contamination equaled 0.67 ± 0.04 for Unit A, which is 29% higher than the average nominal ventilation efficiency of 0.52 ± 0.05 observed for this heat exchanger. For Unit B, the average relative ventilation efficiency corrected for supply stream contamination is 0.49 ± 0.03, which is 13% lower than the observed average nominal ventilation efficiency of 0.56 ± 0.05. Much of this disagreement between relative and nominal ventilation efficiencies may be a result of errors in the measurements of the average tracer concentration in the heat exchanger airstreams, in the estimates of the heat exchanger airstream flow rates, and in the estimates of the true average indoor tracer concentration.

Effective Ventilation Rate

An additional parameter presented in tables 2 and 3 is the effective ventilation rate, which equals the rate of airflow through the heat exchanger (actually the higher of the two airstream flow rates) multiplied by the nominal ventilation efficiency. This number is useful for estimating the increased air-exchange rate, caused by operation of the heat exchanger in similar structures with a different volume than that of the Richmond Research House or Walnut Creek Research House.

As an example of the utility of this parameter we may compare our predictions of increased tracer gas decay rates with actual decay measurements made by Persily et al. (1982) with a Unit A heat exchanger in a 16.6 m^3 test structure using ethane as a tracer. The average increase in tracer decay rate observed by Persily with the heat exchanger operating at low speed was 2.36, which compares well with the increase of 2.41 we calculate from his reported chamber volume and our measured effective ventilation rate of 40 m^3/hr for this heat exchanger at low speed.

Supply and Exhaust Plume Visualization

In order to visualize the supply and exhaust plumes of these ventilation systems, white smoke was introduced into each airstream and a series of photographs taken against a black background. Photographs of the supply-air and exhaust-air plumes of Unit A and Unit B are presented in figure 8. The supply air plume photographs depict the airstream at 0.8 and 4.5 seconds following injection of the smoke, and the exhaust air plume photographs depict the airstream at 0.3 and 3.2 seconds following injection of the smoke.

As can be seen from the photographs the shape, of the two supply-air plumes are substantially different; the supply air plume of Unit A is discharged horizontally across the room with little divergence, while the supply air plume of Unit B is discharged in a broad fan shaped pattern 45 $^{\circ}$ above and below the horizontal. This difference might have been expected to result in significantly different ventilation efficiencies; however, as reported earlier, significant differences were not observed in the ventilation performance of these two heat exchangers.

The exhaust plumes of both units are discharged vertically downward; however, the exhaust plume of Unit B is discharged close to the exterior wall and during tests in the central location is partially diverted by the window ledge, which causes it to curl upward about the supply-air intake. The exhaust plume of Unit A is also discharged vertically downwards but at a point away from the exterior wall, thus clearing the window ledge and dispersing on the ground.

As indicated earlier, Unit B is designed for easy installation through an exterior wall and would not likely be installed above a window ledge. Therefore, the results from tests of Unit B in the central location may not be as representative of typical performance as results from tests in the end location where no window ledge obstructed the exhaust plume. Measurements of airstream tracer concentrations, however, did not indicate a significant increase in the rate of recirculation due to partial obstruction of the exhaust plume by the window ledge.

In summary, the supply and exhaust plumes of the two exchangers appeared quite different, but the appearance of the plumes could not be correlated to the measured performance.

SUMMARY

The nominal ventilation efficiencies of the two commercially available, residential wall- or window-mounted heat exchangers have been determined by multipoint tracer gas decays to be in the range of 0.47 to 0.66 for a variety of operating configurations. Because the effective ventilation rate of these type systems is substantially less than the manufacturers' specified ventilation rate, additional equipment and/or fan power will be required to obtain the desired ventilation. No significant correlations between ventilation efficiency and heat exchanger model or operational strategies were observed.

The highest local ventilation efficiencies were observed at points near the heat exchanger, i.e., in the same room as the heat exchanger. Better distribution of ventilation air was observed when these unducted MVHX systems were installed in a central location rather than in an end location. The ventilation observed in rooms isolated from the rest of the house by closed doors is negligible unless a central furnace fan is operating.

Some preliminary tests indicate that, in the heat exchangers tested in this study, the ventilation inefficiency resulting from internal cross-stream leakage is significant. Additional testing of these MVHX systems in a laboratory setting is necessary to accurately determine the magnitude of different sources of ventilation inefficiency.

REFERENCES

Beach, R.K. 1979. "Relative tightness of new housing in the Ottawa area." Division of Building Research, National Research Council of Canada, Ottawa.

Dickinson, J.B.; Grimsrud, D.T.; Krinkel, D.L.; and Lipschutz, R.D. 1982. "Results of the Bonneville Power Administration weatherization and tightening projects at the Midway Substation residential community." University of California, Lawrence Berkeley Laboratory Report LBL-12742.

Fisk, W.J.; Roseme, G.D.; and Hollowell, C.D. 1980. "Performance of residential air-to-air heat exchangers--Test methods and results." University of California, Lawrence Berkeley Laboratory Report LBL-11793.

Fisk, W.J.; Archer, K.M.; Boonchanta, P.; and Hollowell, C.D. 1981. "Performance measurements for residential air-to-air heat exchangers." Presented at the International Symposium on Indoor Air Quality, Amherst, MA, October 13-16. University of California, Lawrence Berkeley Laboratory Report LBL-12559.

Harrje, D.T., and Mills, T.A., Jr. 1978. "Air infiltration reduction through retrofitting." In ASTM, STP 719, Building Air Change Rate and Infiltration Measurements, pp. 60-72. Eds., C.M. Hunt, J.C. King, and H.R. Trechsel. American Society for Testing and Materials; LBL-6849.

Hollowell, C.D., and Miksch, R.R. 1981. "Sources and concentrations of organic compounds in indoor environments." In Proceedings of the New York Academy of Medicine Symposium on Health Aspects of Indoor Air Pollution, New York, NY, May 28-29.

Malström, T.G., and Ahlgren, A. 1981. "Aspects of efficient ventilation in office rooms," Royal Institute of Technology, Division of Heating and Ventilating, Stockholm, Sweden.

Nero, A.V. 1981. "Airborne radionuclides and radiation in buildings: A review." University of California, Lawrence Berkeley Laboratory Report LBL-12948, submitted to Health Physics.

Offermann, F.J., et al. 1981. "Residential air leakage and indoor air quality in Rochester, New York." University of California, Lawrence Berkeley Laboratory Report LBL-13100.

Offermann, F.J.; Fisk, W.J.; Pedersen, B.; and Revzan, K.L. 1982. "Residential air-to-air heat exchangers: A study of the ventilation efficiencies of wall- or window-mounted units." University of California, Lawrence Berkeley Laboratory Report LBL-14358.

Persily, A. 1982. "Understanding air infiltration in homes." Center for Energy and Environmental Studies, Princeton University, Princeton, NJ.

Sandberg, M. 1981. "What is ventilation efficiency?" Building and Environment f1 16:123-135.

Sherman, M.H., and Grimsrud, D.T. 1980. "Measurement of infiltration using fan pressurization and weather data." Presented at the First Symposium of the Air Infiltration Centre on Instrumentation and Measurement Techniques, Windsor, England, October 6-8; University of California, Lawrence Berkeley Laboratory Report LBL-10852.

Skaaret, E., and Mathisen, H.M. 1981. "Ventilation efficiency." The Norwegian Institute of Technology, Division of Heating and Ventilating, Trondheim.

Traynor, G.W.; Apte, M.G.; Girman, J.R.; and Hollowell, C.D. 1981. "Indoor air pollution from domestic combustion appliances." University of California, Lawrence Berkeley Laboratory Report LBL-12886, presented at the 1981 International Gas Research Conference, Los Angeles, CA, Sept. 28-Oct. 1.

ACKNOWLEDGEMENTS

We gratefully acknowledge the substantial assistance we received during the course of this project--in particular, from our LBL colleagues Lloyd Davis, James Koonce, William Nazaroff, and Alan Robb for their assistance in designing and fabricating the equipment used to make the numerous multipoint tracer decay measurements in this study; Keith Archer and Paris Greenwood, who assisted in reducing the large amounts of test data; Suzanne Doyle for analysis of the radon samples; and Eimund Skaaret, Max Sherman, and Gail Schiller, who kindly reviewed this report. The efforts of Nancy Morrison in preparing the final transcript of this report are also greatly appreciated. The authors also wish to acknowledge the support of Phil Thor, the BPA project manager, and Howard Ross, the DOE project manager

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Buildings and Community Systems, Buildings Division of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098, and the Bonneville Power Administration, Portland, OR.

TABLE 1
Heat Exchanger Specifications

Model	Fan Speed	Flow rate m^3/h Supply	Flow rate m^3/h Exhaust	Power Requirement (watts)
Unit A[a]	Low	56	65	24
	Medium	90	110	42
	High	117	144	57
Unit B[b]	Low	53	53	25
	High	95	95	41

[a]Data from Fisk, W.J.; Archer, K.M.; Boonchanta, P.; and Hollowell, C.D. 1981. "Performance measurements for residential air-to-air heat exchangers." Lawrence Berkeley Laboratory Report LBL-12559.

[b]Data from manufacturer for 100V/60Hz electricity.
Heat exchangers operated with 100V/60Hz electricity during tests.

TABLE 2

Results of Ventilation Efficiency Tests at the LBL Richmond Research House

Test No., Description[a]	T_s-T_{in} (^oC)[b]	Natural Decay Rate Observed[c]	Mech. Vent Decay Rate Observed[d]	Predicted Increase in Decay Rate[e]	Nominal Ventilation Efficiency[f]	Effective Ventilation Rate (m^3/hr)[g]	Local Ventilation Efficiencies[h] 1	2	3	4	5	6
1-A1C	2.5	0.23	1.45	2.01	0.61	40	0.54	0.61	0.67	0.46	0.62	0.61
2-B1C	1.1	0.17	1.22	1.64	0.64	34	0.71	0.71	0.61	0.61	0.68	0.63
3-B3C	1.5	0.07	0.49	0.74	0.57	54	0.63	0.51	0.67	0.57	0.64	0.52
4-B3CH	5.8	0.08	0.46	0.74	0.51	48	0.55	0.51	0.62	0.47	0.48	0.48
5-A3C	2.9	0.05	0.49	0.86	0.50	55	0.48	0.44	0.52	0.51	0.49	0.51
6-A3CH	8.2	0.09	0.51	0.86	0.49	54	0.46	0.52	0.49	0.53	0.47	0.51
7-B3E	1.4	0.06	0.47	0.74	0.55	52	0.42	0.62	0.40	0.50	0.75	0.54
8-B3EH	5.6	0.08	0.46	0.74	0.51	48	0.37	0.69	0.39	0.41	0.62	0.43
9-A3E	1.8	0.09	0.51	0.86	0.49	54	0.37	0.53	0.39	0.51	0.58	0.47
10-A3EH	4.0	0.07	0.52	0.86	0.51	56	0.49	0.51	0.50	0.48	0.53	0.49

NOTE: See table 3 for footnotes.

TABLE 3

Results of Ventilation Efficiency Tests at the LBL Walnut Creek Research House

Test No., Description[a]	Natural Decay Rate Observed[c]	Mech. Vent. Decay Rate Observed[d]	Predicted Increase in Decay Rate[e]	Nominal Ventilation Efficiency[f]	Effective Ventilation Rate (m^3/hr)[g]	Local Ventilation Efficiencies[h] 1	2	3	4	5	6
11-1A	0.12	0.42	0.48	0.63	69	0.80	0.68	0.76	0.58	0.79	-0.26
12-2A	0.12	0.49	0.56	0.66	86	0.80	0.72	0.87	0.60	0.72	-0.04
13-1AF	0.29	0.52	0.48	0.47	52	0.46	0.62	0.38	0.58	0.43	0.21
14-2AF	0.29	0.63	0.56	0.61	79	0.75	0.87	0.69	0.71	0.68	0.42

[a] A = Unit A Heat Exchanger (H.E.); B = Unit B H.E.; 1 = 1-room test; 3 = 3-room test; C = central H.E. location; E = end H.E. location; H = with electric baseboard heat; 1A and 2A refer to one and two heat exchangers operating at the Walnut Creek House, respectively; F = with furnace fan operating.

[b] Average indoor temperature, T_{in}, minus temperature of supply air, T_s, entering house from heat exchanger.

[c] Spatial average of tracer-gas decay rate measured at six indoor locations without operation of H.E. and corrected for leakage through installed but not operating H.E.

[d] Spatial average tracer-gas decay rate measured at six indoor locations with operation of H.E.

[e] Predicted increase in tracer-gas decay rate assuming outdoor concentration is zero and perfect mixing of indoor air (airflow rate through H.E./ volume of structure ventilated).

[f] Ratio of measured to predicted increase in tracer-gas decay rate (column D minus column C divided by column E).

[g] Equivalent airflow rate through H.E. if no cross-stream leakage or recirculation occurred and if indoor air was perfectly mixed (actual flow rate through H.E. multiplied by nominal ventilation efficiency).

[h] The observed increase in local air-exchange rate at six indoor locations divided by the predicted increase in local air-exchange rate.

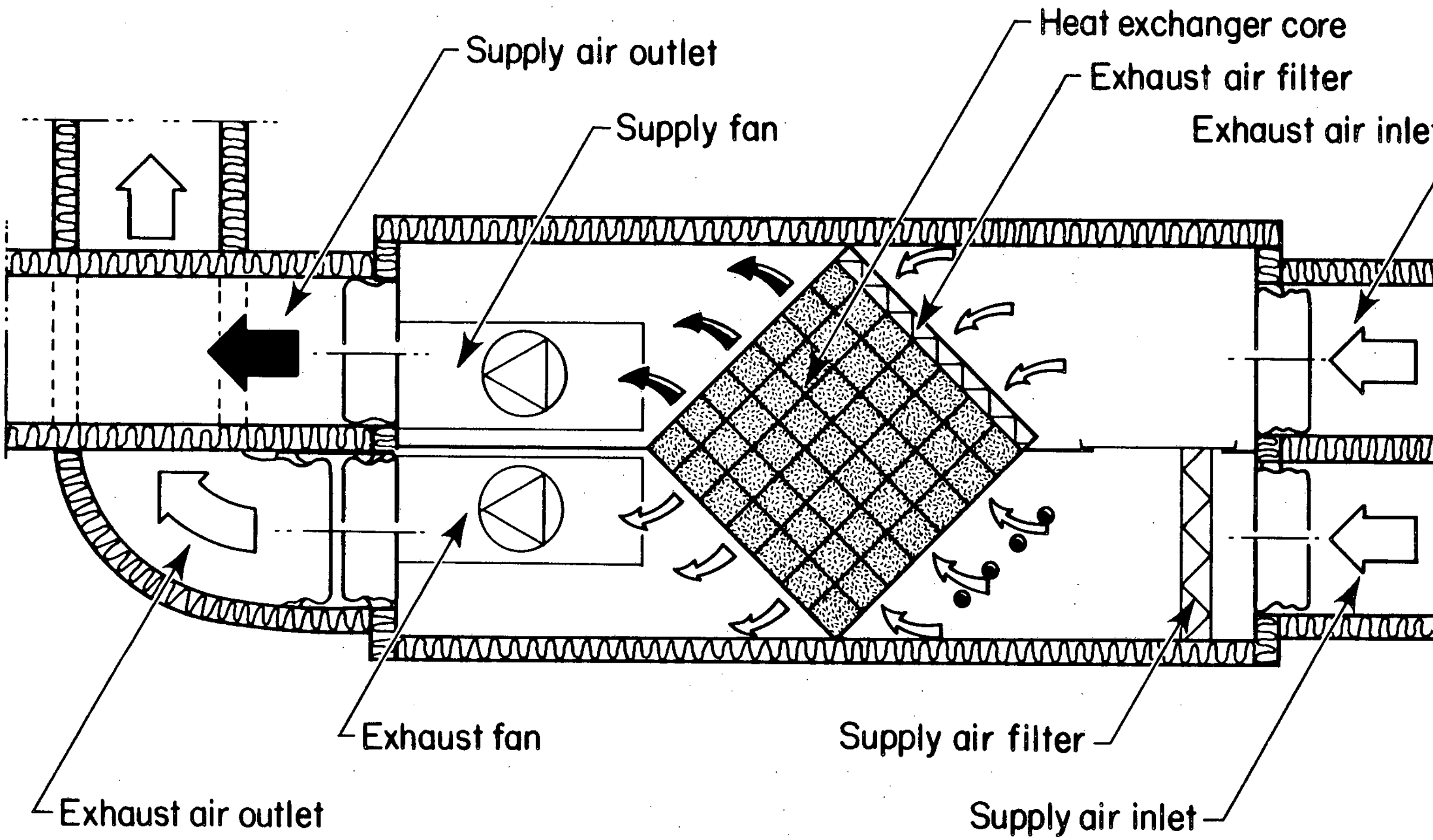


Figure 1. Schematic diagram of a residential air-to-air heat exchanger.

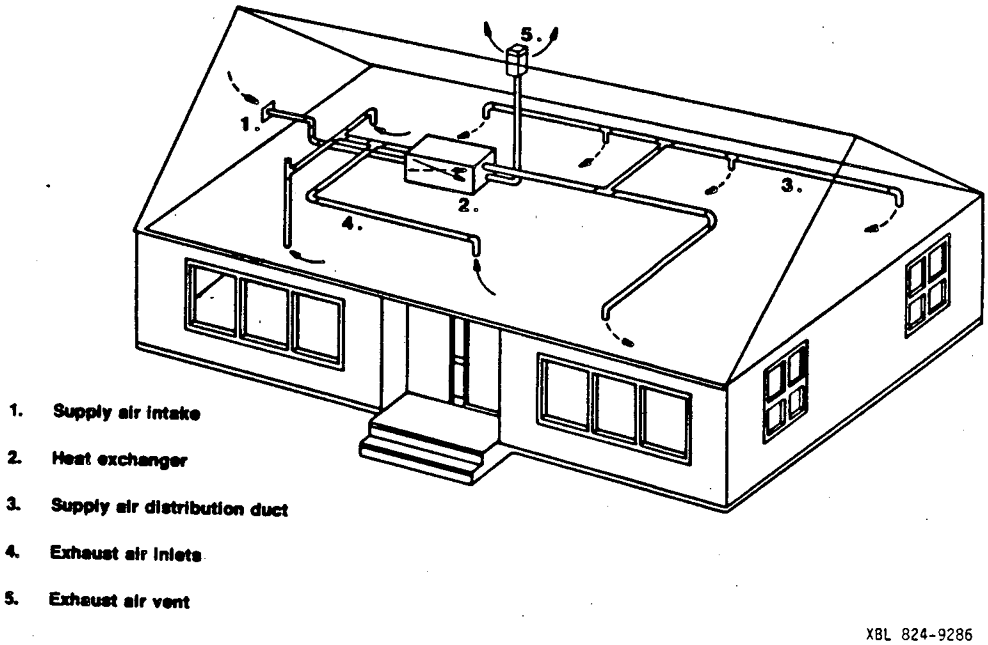


Figure 2a. Illustration of a fully ducted installation of a residential air-to-air heat exchanger.

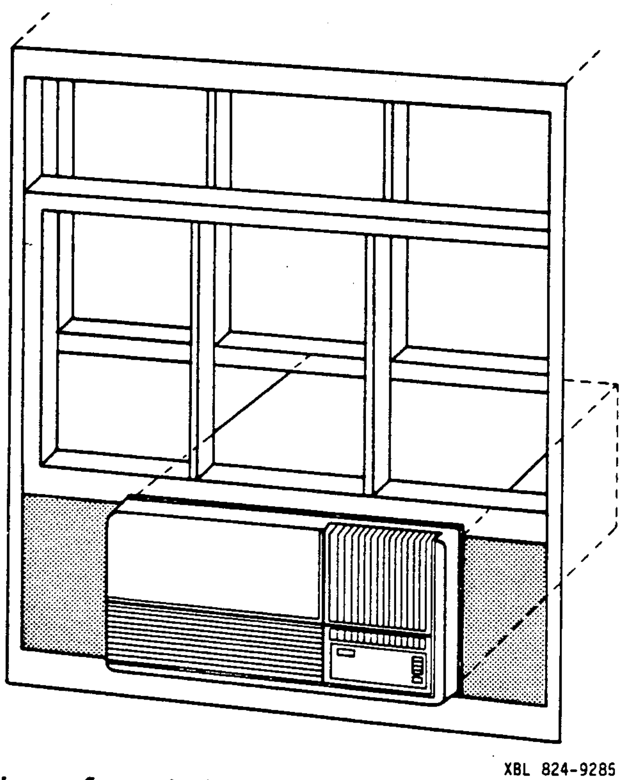
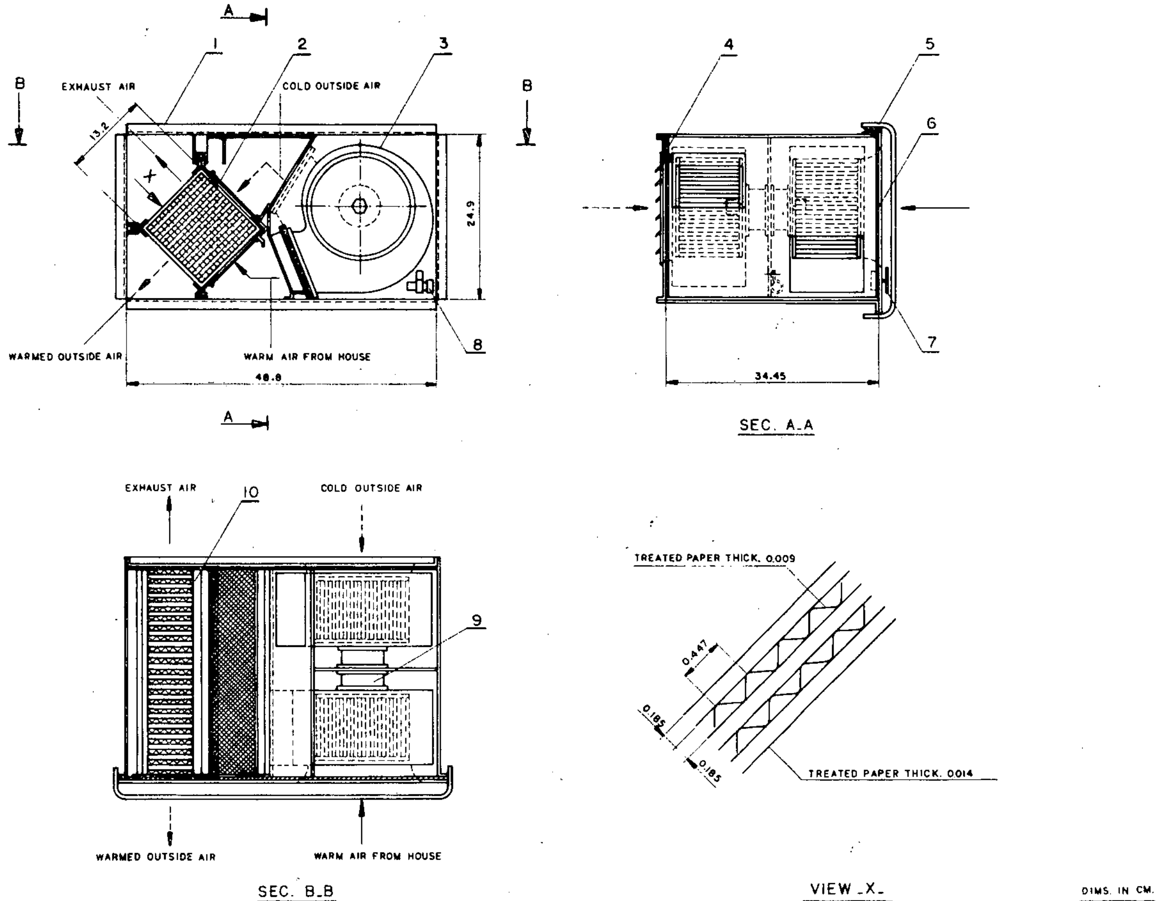
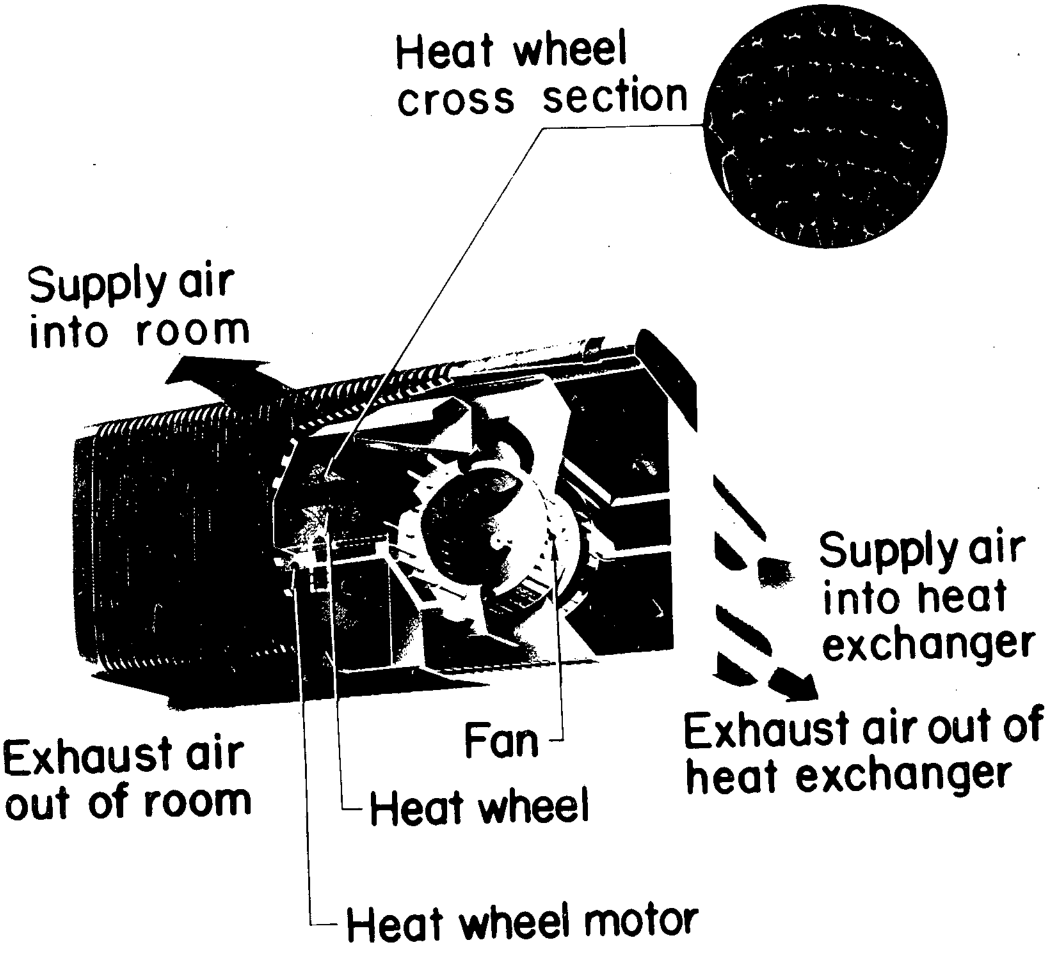


Figure 2b. Illustration of a window installation of a residential air-to-air heat exchanger.



XBL 8012-12838

Figure 3. Drawing of Unit A, a cross-flow type heat exchanger.



NOTE: patented product

CBB 820-10070

Figure 4. Drawing of Unit B, a rotary type heat exchanger.

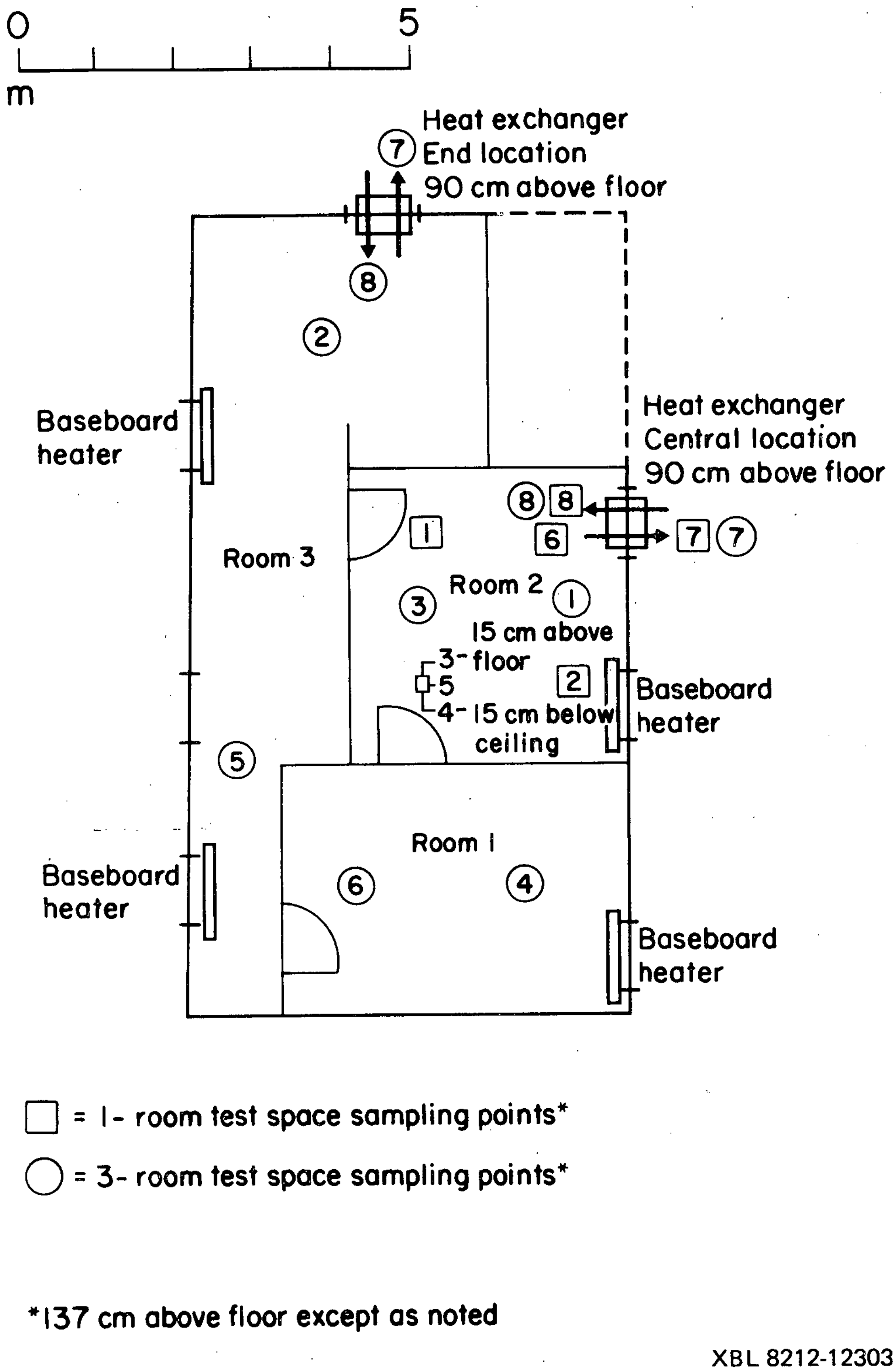


Figure 5. Floor plan of the Richmond Research House one- and three-room sampling patterns.

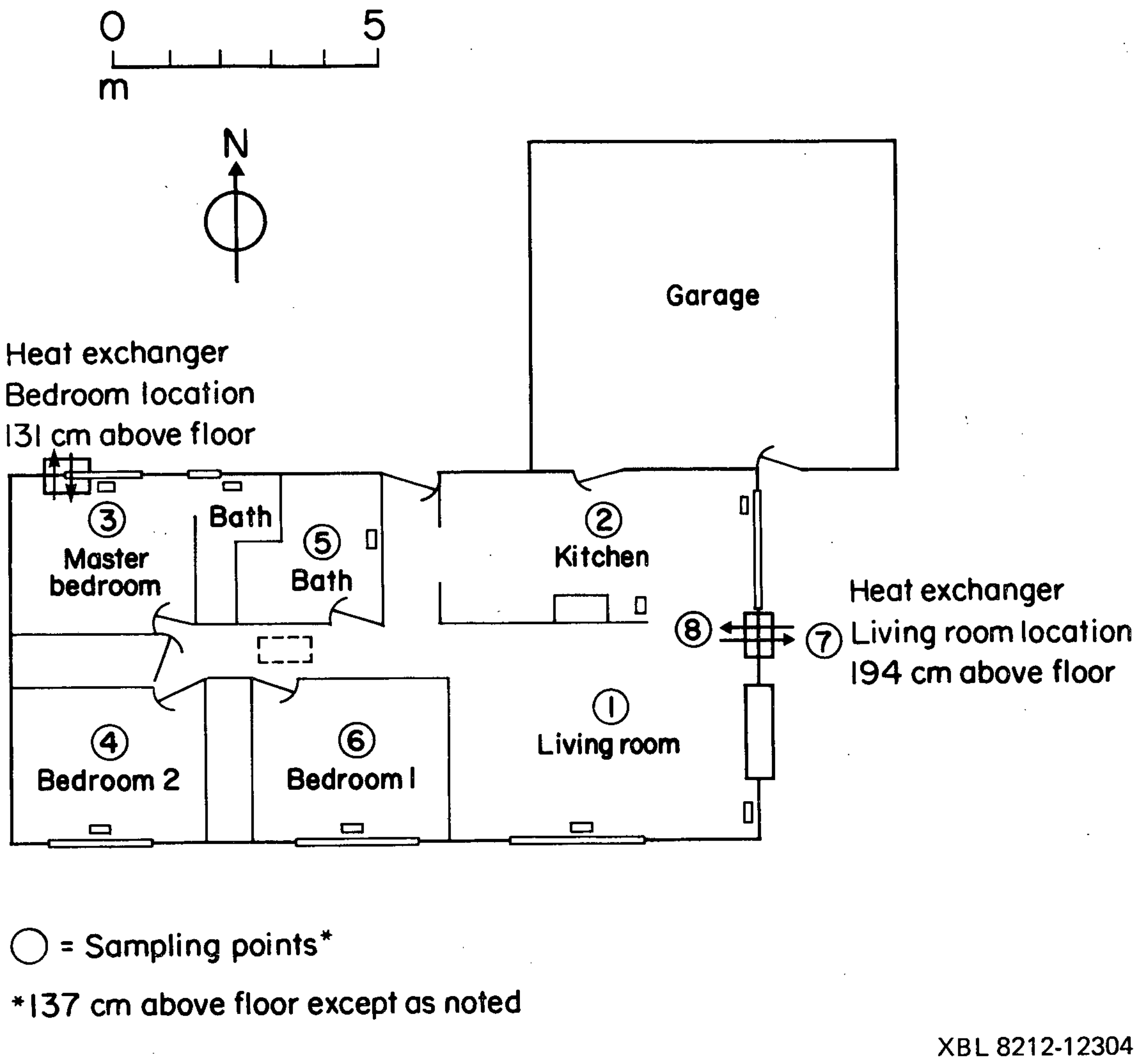


Figure 6. Floor plan of the Walnut Creek Research House sampling pattern.

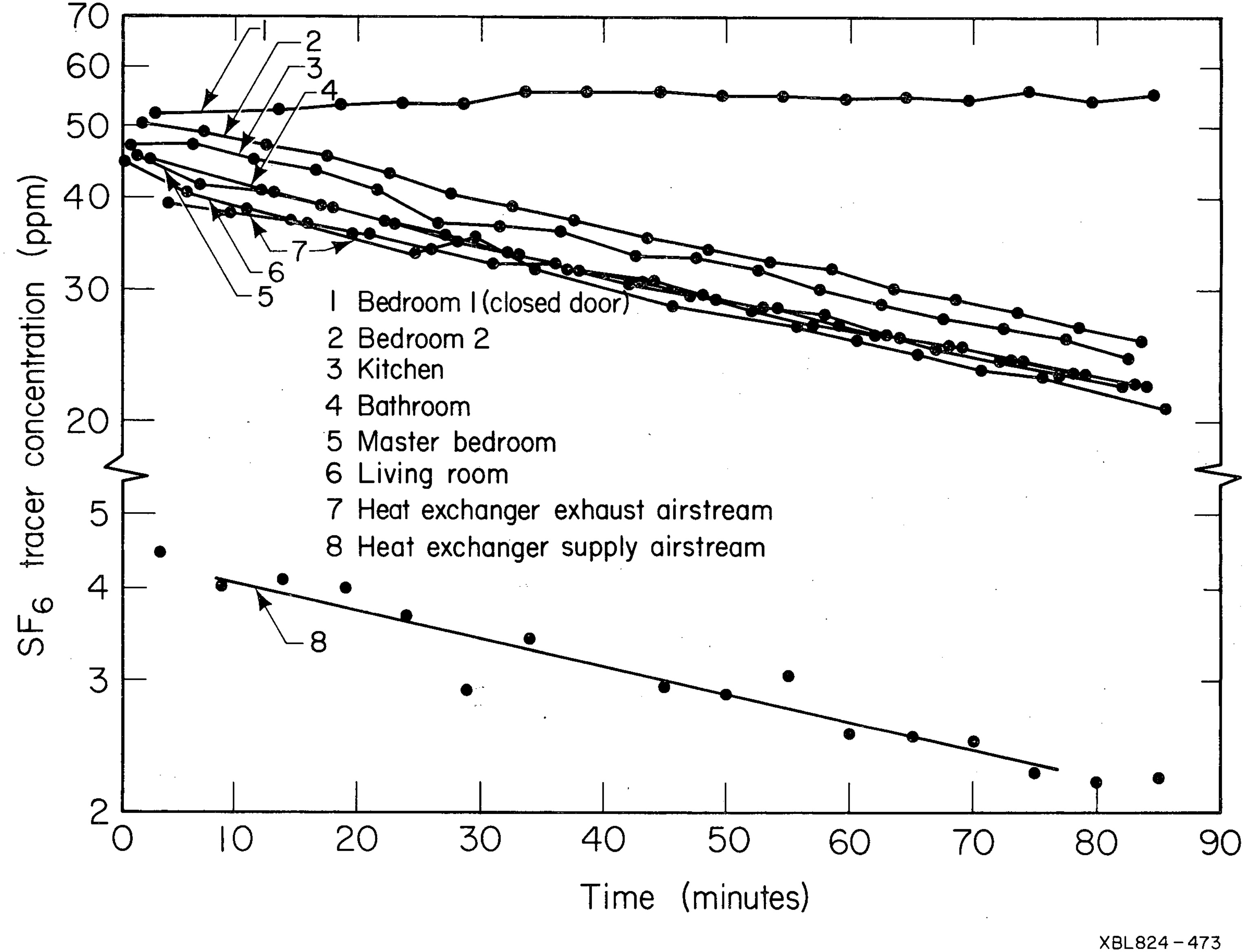
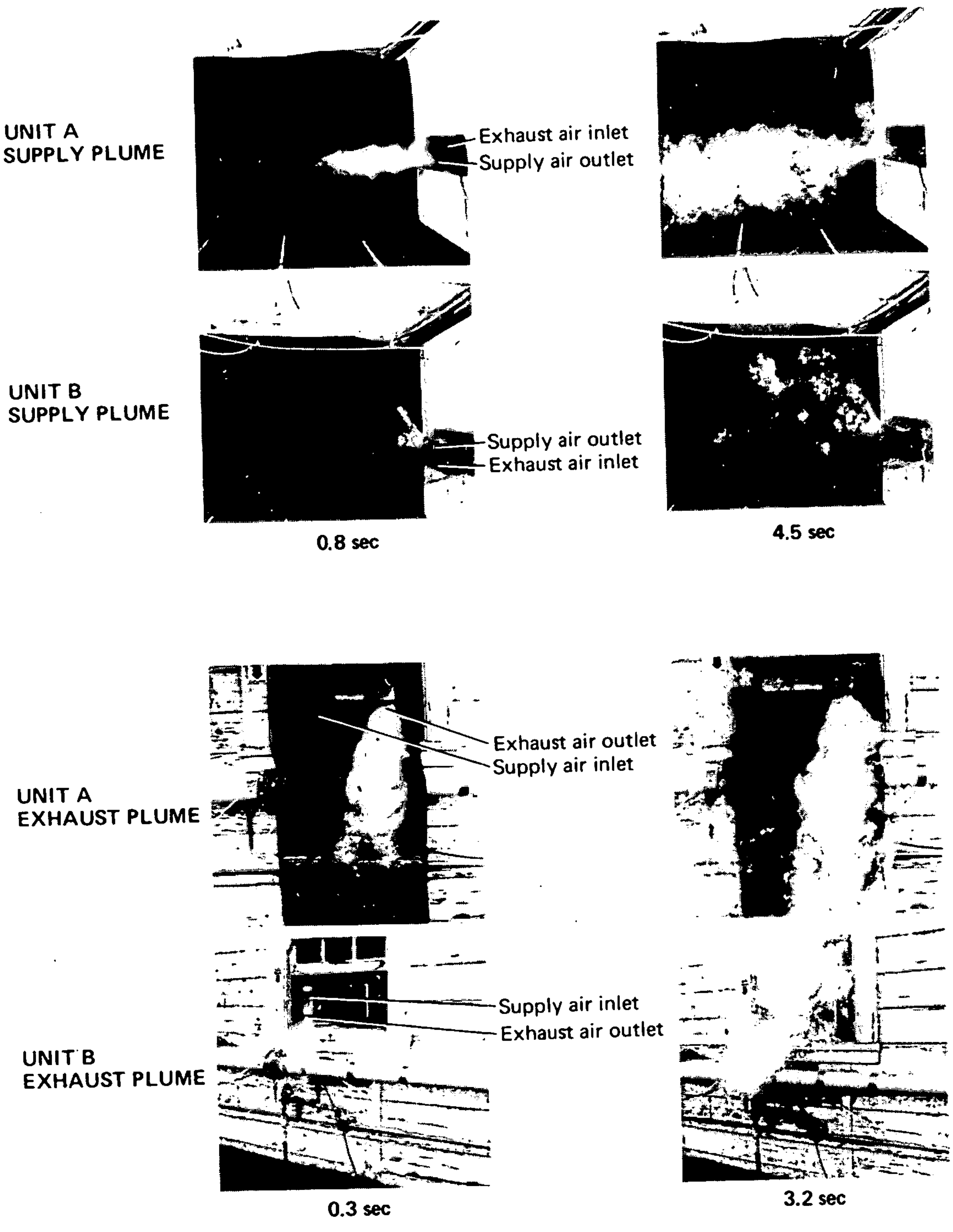


Figure 7. A semi-logarithmic plot of SF_6 tracer concentration versus time for eight locations during test #1 in the Walnut Creek Research House (Unit A operating at medium speed in a living room location and bedroom 1 closed).



XBB 8212-10931

Figure 8. Photographs of the supply and exhaust air plumes of Unit A and Unit B heat exchangers. Time = 0 seconds at the start of smoke injection.

TECHNICAL INFORMATION DEPARTMENT
LAWRENCE BERKELEY LABORATORY
UNIVERSITY OF CALIFORNIA
BERKELEY, CALIFORNIA 94720