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RESIDENTIAL AIR-TO-AIR HEAT EXCHANGERS: A STUDY OF THE VENTILATION EFFICIENCIES OF WALL- OR WINDOW-MOUNTED UNITS

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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, the Lawrence Berkeley Laboratory has conducted a series of experiments 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 or 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, 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 this ventilation air -- estimates range from 20-40% (1, 2) of the total residential heating load or, on a national scale, 2 to 4 quads of energy per year. In existing homes, conservation measures such as weatherstripping and caulking can reduce infiltration and thus save energy (3, 4). 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 (5, 6).

One of the problems associated with reduced ventilation rates is that indoor humidity levels and concentrations of indoor-generated air contaminants are often increased (7, 8, 9). 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 outside air leaks into the house and replaces indoor air. When the rate of air leakage is reduced 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 between 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). 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 and the exhaust air is cooled before leaving the house.

Currently little information is available on the performance of MVHX systems under the actual operating conditions found in residences. Laboratory tests (10, 11) 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 by Offermann et al. (5) in nine occupied Rochester, New York residences 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 2). 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 units, as shown in Figure 3, 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. However, the performance of these units in ventilating a structure has received little study. Because air exits and enters these heat exchangers from locations in close proximity, considerable recirculation, or short circuiting between airstreams, 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 mixing between rooms is not perfect). 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 on the ventilation efficiency of these systems.

In this report we discuss ventilation efficiency and other parameters relevant to the ventilation performance of these exchangers, describe the measurement techniques used in this study, and present 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.

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 in which a series of tests were performed. (A 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 concentration. With and without the heat exchanger(s) operating, the tracer gas concentration was then measured versus time at six indoor locations and in the air-streams supplied and exhausted by the heat exchanger. The test data was used to calculate a number of performance parameters including: (1) tracer gas decay rates; (2) local air exchange rates; and (3) ventilation efficiencies that are discussed below.

A. 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,
and 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 non-reactive 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. Skaret (12), Sandberg (13), Malstrom (14), 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. The concentrations of tracer gas at different indoor locations become unequal, however, the ratio of any two concentrations is constant after a uniform decay rate is established. Areas with the lowest concentrations are the zones which receive the greatest amount of ventilation.

B. 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.[13] In this paper, however, we use a method recommended by Skaret to introduce the local air exchange rate. If we consider a small perfectly mixed volume element in an imperfectly mixed indoor space, the mass balance equation for the element is

$$-CdQ = dV\frac{dC}{dt} \tag{3}$$

where: C = the concentration of tracer gas in the element,
dV = the volume of the element,
t = a time variable
and dQ = the flow rate of fresh (i.e., outdoor) air into the element.

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, equation 3 can be rewritten in the form

$$-n^j C dt = dC. \tag{4}$$

Integrating both sides of the equation yields the expression for local air exchange rate

$$n^j = -\int_{C(0)}^{C(t)} dC \Big/ \int_0^t Cdt \tag{5}$$

where: n^j = the local air exchange rate at point j,
C(0) = the tracer concentration observed at point j at time t=0,
and C(t) = the tracer concentration observed at point j at time t.

The local air exchange is an indicator of the amount of ventilation that occurs at each location and comparing local ventilation rates from location to location indicates how ventilation air is distributed throughout the space that is ventilated.

C. Ventilation Efficiencies

Ventilation efficiencies (excluding relative ventilation efficiencies) relate the observed concentrations, decay rates, or local air exchange rates to predictions for an ideal case. Various indicators of ventilation efficiency can be calculated depending upon whether the data is from experiments with steady state or transient concentrations, whether tracer gas concentrations or actual contaminant concentrations are measured, and depending on the comparison employed. For our calculations we compare to the ideal case when the indoor air is perfectly mixed, and no short-circuiting or leakage occurs between airstreams. It should be noted, however, that perfect mixing is not always the optimal condition but 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 our measurements, a nominal ventilation efficiency is defined using the equation

$$E_1 = (\lambda - x\lambda_*)\ V/Q \qquad (6)$$

where: E_1 = the nominal ventilation efficiency,

V = the volume of the structure ventilated,

Q = the rate of air flow 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,

and x = a correction factor to account for air leakage through the heat exchanger when installed but not operating (described more fully later).

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

$$E_2 = (n^j - xn_*^j)\ V/Q \tag{7}$$

where: 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 of perfect mixing and no short-circuiting or leakage between airstreams.

In addition to the ventilation efficiencies described above, a relative ventilation efficiency can be calculated for each location from the equation

$$E_3(t) = C_e(t)/C(t) \tag{8}$$

where: $E_3(t)$ = the relative ventilation efficiency,

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

$C(t)$ = 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. Relative ventilation efficiencies are generally compared from point to point, however, a spatial average value can be calculated by substituting the average of the six measured indoor concentrations for C(t). It is this spatial average relative ventilation efficiency that is reported in this paper. Since its value can vary somewhat throughout the decay, the average value for the decay is reported.

Measurements performed when the heat exchanger was not operating are affected by air leakage through the heat exchanger. Since we wish 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 6 and 7. This factor corrects for air leakage through the non-operating heat exchanger and has been estimated from tests in which the natural decay rate was monitored with and without leakage through the heat exchanger. From these tests we estimated correction factors to be 0.65 for one-room tests at the Radon Research House, 0.75 for three-room tests at the Radon Research House, 0.86 for tests with the furnace fan off at the Walnut Creek Research House, and 0.94 for tests with the furnace fan on at the Walnut Creek Research House.

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

The ventilation performance of two commercially available MVHX systems designed for installation through walls or windows was studied. One unit is manufactured by Mitsubishi Electric of Japan (Model VL-1500) and is distributed by Mitsubishi Electric Sales America, Inc., Compton, California. The second unit is manufactured by Sharp of Japan (Model GV-120) and is distributed by Berner International Corporation, Woburn, Massachusetts. Both units are similar in size to small window-mounted air conditioners.

As shown in Figure 4, the Mitsubishi exchanger uses a crossflow heat transfer surface, or core, made of a treated paper which transfers moisture as well as heat between airstreams. Figure 5 illustrates the Sharp heat exchanger which uses a rotating counterflow "heat wheel" core coated with a dessicant to transfer heat and moisture between airstreams. Both heat exchangers contain two fans driven by a single motor to provide the flow of supply and exhaust air. The Mitsubishi unit has three fan speeds and the Sharp unit has two fan speeds. Table 1 lists air-flow rates and fan-power requirements for each heat exchanger as a function of their different fan speeds. The data for the Mitsubishi unit was determined from tests at LBL (11), and the data for the Sharp unit was obtained from the manufacturer's published literature.

When installed, the housing of the Mitsubishi unit penetrates the window or wall. The Sharp unit is installed inside the house on the surface of an exterior wall and has two ducts (8 cm in diameter) that penetrate to the outside. Both heat exchangers utilize weather hoods over the outside vents, and the weather hoods were installed for the tests described here.

Description of Structures and Locations of Heat Exchangers

Experiments were performed in two structures. The Radon Research House, shown in Figures 6 and 7, is located at the Richmond Field Station in Richmond, California, a research laboratory operated by the University of California at Berkeley. The test space, three interconnected rooms with a total floor area of 56.0 m^2, a ceiling height of 2.36 m, and a total volume of 132 m^3, has been renovated to assure low rates of air leakage through the building envelope. The existing forced-air furnace ducts were sealed off during the tests. The heat exchangers were installed 0.9 m above floor level in a central location (room 2) or an end location (room 3).

The Walnut Creek Research House, shown in Figure 8, is a residence located in Walnut Creek, California. The house 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 house's 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 and another was mounted in the living room, 1.94 m above floor level.

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. The concentration was then measured versus time at eight different locations using two non-dispersive infrared analyzers (Foxboro-Wilkes Model 101). The analyzers were calibrated at the beginning and end of each decay using three primary standard SF_6 calibration gases (e.g. 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 air stream. The other six sampling points were located within the test space as depicted in Figures 6, 7, and 8. Air was sampled at two points simultaneously for a one-minute sampling period at a flow rate of 20 liters per minute and sequenced so that all eight points were sampled every four minutes.

At the Radon Research House, tests were carried out with the aid of a microprocessor. A BASIC language program 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 Radon 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 free-standing heaters were located under four windows in the test space, as shown in Figures 6 and 7. Each heater was controlled separately by a thermostat mounted nearby on an inside wall.

One test at the Radon Research House involved simultaneous use of SF_6 and radon (Rn) tracer gases to determine if the decay rates of each gas would be the same. For this test, Rn was injected into the test space from a controlled radium source and mixed to an indoor average concentration of approximately 100 pCi/L. Radon concentrations were measured during the decay by taking grab samples of air at three points. Descriptions of the measurement technique for radon are available elsewhere (15).

Test Procedure

All tests at the Radon 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. (Essentially the same test procedure was followed for the one test when radon was used as a tracer in addition to SF_6.)

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 6 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 6 locations in the re-mixed test space. (This step was performed for tests at the Radon Research House only.)

Tests Performed

Table 2 summarizes details of the fourteen tests performed. The first two tests at the Radon Research House were one-room tests conducted in Room 2 with both doors to the room closed. The Mitsubishi heat exchanger was operated in the central heat exchanger location for the first one-room test and the Sharp heat exchanger 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 decay for our measurement system. The remaining eight tests at the Radon 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 the Mitsubishi exchanger (110 m^3/h) and high fan speed for the Sharp exchanger (95 m^3/h). In one of these eight tests radon and SF_6 decay rates were measured simultaneously.

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 Mitsubishi heat exchanger located in the living room and operating at the medium fan-speed setting. For the other two tests, two Mitsubishi 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 #1 was closed during all of the tests at the Walnut Creek Research House.

RESULTS AND DISCUSSION

The results of the ventilation efficiency measurements made in the Radon Research House are compiled in Table 3 and the results for tests performed at the Walnut Creek Research House are compiled in Table 4.

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. An estimate of the uncertainty in the reported values for nominal ventilation efficiency and local ventilation efficiency is presented in Appendix A. Prior to discussing the results presented in these tables, we discuss a plot of SF_6 concentration versus time for one test.

Tracer Gas Concentration Versus Time

Figure 9 is a semi-log plot of SF_6 concentration versus time for the six indoor sampling locations and two heat exchanger (supply and exhaust airstream) sampling locations made during test #11. This test was made with one Mitsubishi heat exchanger operating at medium fan speed and installed in the living room of the Walnut Creek Research House. The furnace 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 #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 (but only slightly higher) 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 short circuiting from the exhaust to supply airstream outside the house is small. However, as will be discussed later, we are uncertain as to how well the measurements represent the average concentration in the airstreams.

Nominal Ventilation Efficiency

Radon Research House -- The nominal ventilation efficiency for all ten tests performed at the Radon Research House averaged 0.53 ± 0.06 (± one standard deviation). Nominal ventilation efficiency averaged 0.56 ± 0.06 for the Sharp heat exchanger which based on our estimated uncertainty of ± 0.06 for a single test is close to being a significantly higher efficiency than the 0.51 ± 0.06 average observed for the Mitsubishi heat exchanger. In each of the five pairs of tests, we consistently calculated higher efficiencies for the Sharp heat exchanger. The probability that these consistent differences in efficiency are a result of random measurement error is only 3.5% based on a two tailed t-test. However, a 10% error in our estimate of the actual airstream flow rates for the Sharp heat exchanger would also explain the apparent difference.

The nominal ventilation efficiency for one-room tests averaged 0.63 ± 0.03 which is significantly greater than the average efficiency of 0.51 ± 0.04 for all three-room tests. We would expect better mixing and higher efficiencies in the one-room tests, however, this difference can not 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 short-circuiting 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 0.50 ± 0.02 which is not significantly different from the 0.52 ± 0.05 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 0.51 ± 0.04 for tests with centrally located heat exchangers and 0.51 ± 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 mix-up periods. The decay rate necessary to reduce the average pre-decay tracer concentration to the observed post-decay concentration was calculated assuming perfect mixing from equation 2. The ventilation efficiency computed for all ten tests from the mix-to-mix effective decay rates averaged 0.54 ± 0.06 which compares very well with the 0.53 ± 0.06 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 and suggests that the air in each room was fairly well mixed.

Walnut Creek Research House -- The nominal ventilation efficiency for the four tests performed at the Walnut Creek Research House averaged 0.58 ± 0.10. A comparison of the data for tests run without operation of the furnace fan yields a nominal ventilation efficiency of 0.64 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 0.58 for the tests with two heat exchangers operating, and a large decrease in nominal ventilation efficiency from 0.64 to 0.44 when only one heat exchanger was operating. We have no explanation for these reductions in ventilation-efficiency. The low ventilation efficiency of 0.44 observed for test #13 is atypical and should not be assumed to be a result of the operation of the furnace fan. A careful examination of the data has not revealed any clues to the cause of this low efficiency.

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.

Radon 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 49% higher than the 0.41 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 0.55 which is just 8% higher than the average of 0.51 observed in both Room 1 and Room 3.

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, our one-room tests constituted the only experiments with sample points at heights other than 1.37 m above the floor. For the Mitsubishi one room tests, 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 0.61 $\pm$ 0.01 average efficiency calculated for the four points located in the middle of the air space. For the Sharp one-room tests the local ventilation efficiency was 0.63 at both the floor and ceiling sample points which is slightly lower than the 0.70 $\pm$ 0.03 average efficiency calculated for the four points located in the middle of the air space.

Walnut Creek Research House -- As we found in the Radon 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 #1. Since the door to this room was closed during testing, these results are consistent with out expectations. For tests #13 and #14, which were replicates of tests #11 and #12, only 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 #1, however, it is noted that the local ventilation efficiency of this room improved from essentially zero to an average of 0.26 with the furnace fan on (i.e., 46% of the average local ventilation efficiency observed in the open rooms).

Measurements of Concentration in the Heat Exchanger Airstreams

In addition to monitoring the tracer concentration at six indoor locations we measured the tracer concentration in the exhaust and supply airstreams at points were 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 published airstream flow rates and actual airstream tracer concentration measurements made on all four sides of the heat exchanger (i.e., in each airstream and on each side of the heat exchanger core) indicate significant measurement error. Despite this uncertainty we discuss briefly the relative ventilation efficiencies calculated using this data.

The average relative ventilation efficiency as calculated by the ratio of the exhaust concentration to average indoor concentration was 0.78 ± 0.02 for the Mitsubishi heat exchanger and 0.84 ± 0.05 for the Sharp heat exchanger. However, since significant contamination of the supply airstreams of both heat exchangers was observed, we 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 the Mitsubishi which is 31% higher than the average nominal ventilation efficiency of 0.51 ± 0.06 observed for this heat exchanger. For the Sharp heat exchanger the average relative ventilation efficiency corrected for supply stream contamination is 0.49 ± 0.03 which is 13% lower than the observed average nominal efficiency of 0.56 ± 0.06. We suspect that much of this disagreement between relative and nominal ventilation efficiencies is a result of error in our measurements of the average tracer concentration in the heat exchanger airstreams, our estimates of the heat exchanger air stream flow rates, and our estimates of the true average indoor tracer concentration.

Effective Ventilation Rate

An additional parameter presented in Tables 3 and 4 is the effective ventilation rate which equals the rate of air flow through the 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 structures with a different volume than that of the Radon Research House or Walnut Creek Research House. Because the effective ventilation rate can be significantly less than the specified nominal ventilation rate, additional equipment and/or fan power may be required to obtain the desired ventilation.

Radon Decay Rate

In test #6 radon gas was injected along with the SF_6 as a second tracer and the decay rate of each was monitored simultaneously. Radon was sampled at one location in each of the three rooms. The radon decay rates in these three rooms were 0.48, 0.47, and 0.36 respectively. These decay rates agree fairly well with the observed average SF_6 decay rate of 0.51 ± 0.01 which indicates that radon gas was removed at approximately the same rate as our SF_6 tracer gas. This is consistent with our expectations since both radon and SF_6 are chemically inert gases which are not normally present in outdoor air in significant concentrations.

SUMMARY

The nominal ventilation efficiencies of the Sharp CV-120 and Mitsubishi VL-1500 have been determined by multipoint tracer gas decays to be in the range of 0.44 to 0.65 for a variety of operating configurations. 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.

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Appendix A: Uncertainty Analysis

The uncertainty in the measured values of nominal ventilation efficiency was estimated by considering the uncertainty in SF_6 concentration, structure volume, airstream flow rate, and the correction factor x in equations 6 and 7.

The largest cause of error in measurement of SF_6 concentration was drift in the response of the SF_6 analyzer which was calibrated automatically at the start and end of each decay. Typically, the drift in response to a calibration gas was 1 to 3% and if the drift was greater than 6% the data was discarded and another test performed. If we assume an error in measurement of SF_6 concentration that varies linearly with time from 0 to 6%, the associated uncertainty in nominal ventilation efficiency is approximately ± 0.03 (i.e., due to analyzer drift a measured efficiency of 0.55 could be as low as 0.52 or as high as 0.58).

The volumes of the structures were determined from detailed measurements. We estimated an uncertainty of ±2% and ±5% in the volume of the Radon Research House and Walnut Creek Research House, respectively, which causes corresponding uncertainties of ± 0.01 and ± 0.03 in the nominal ventilation efficiency.

The airstream flow rates for the Mitsubishi exchanger were determined from measurements performed during a previous study. The estimated uncertainty in flow rate of ±7%, corresponds to an uncertainty of ±0.04 in the nominal ventilation efficiency. For the Sharp exchanger, we used the manufacturer's published values for airstream flow rates -- no other source of data was available. If the actual flow rates for the Sharp exchanger differ substantially from the published values, the result would be a corresponding error in the reported ventilation efficiencies for this exchanger.

The values for correction factor x, which relates infiltration rate with and without installation of the heat exchanger(s), are described earlier. By considering the data used to estimate values for x, the uncertainty in x was estimated for each case (i.e., one-room tests, three-room tests, etc) and the corresponding uncertainty in the nominal ventilation efficiency was determined to be approximately 0.03 for all cases.

Since it is highly unlikely that all independent errors add, a common practice is to take the square root of the sum of the squares of each individual uncertainty. This procedure yields an estimated uncertainty of ± 0.06 and ±0.07 for measurements of nominal ventilation efficiency at the Radon Research House and Walnut Creek Research House, respectively.

The factors described above cause essentially the same uncertainty in the local ventilation efficiencies. However, the local ventilation efficiencies are very sensitive to the initial and final tracer gas concentrations because the numerator of the expression for local air exchange rate equals the difference between initial and final concentrations. Examination of the test data indicates that the measured tracer gas concentrations occasionally deviate by ±2 ppm from the line that represents the best fit to the local data for concentration versus time. (We believe that these deviations are caused primarily by imperfect mixing -- not imprecise measurement.) If the initial or final tracer gas concentration deviates as described above, the result would be a variation in local ventilation efficiency of approximately ±0.07. Thus, the local ventilation efficiency for two slightly different time periods could differ by approximately ± 0.07. Examination of the data indicates that variations of this magnitude due to deviation in the initial or final concentrations occurred in 15% or less of the reported values of local ventilation efficiency.

As an indicator of the random variation associated with our measurements of nominal ventilation efficiency, test #6 was repeated three times while holding experimental conditions as steady as possible. The resulting efficiencies calculated were 0.47, 0.48, and 0.50. While this is a small set of repeated tests, it does indicate that the variation in nominal ventilation efficiency caused by uncontrolled factors (e.g., small changes in indoor and outdoor temperature, windspeed and direction) is relatively small.

In summary, the estimated uncertainty in nominal ventilation efficiency is ±0.06 and ±0.07 for measurements performed at the Radon Research House and Walnut Creek Research House, respectively. A similar uncertainty exists in the local ventilation efficiencies with an additional variance of ± 0.07 estimated to be associated with 15 percent of the reported local ventilation efficiencies.

Table 1. Heat Exchanger Specifications

Manufacturer, Model	Fan Speed	Flow rate m^3/h (ft^3/min)		Power Requirement (watts)
		Supply	Exhaust	
Sharp GV-120[a]	Low	53 (31)	53 (31)	25
	High	95 (56)	95 (56)	41
Mitsubishi VL-1500[b]	Low	56 (33)	65 (38)	24
	Medium	90 (53)	110 (65)	42
	High	117 (69)	144 (85)	57

[a]Data from manufacturer. All data is for 100V/60Hz electricity. Heat exchangers operated with 100V/60Hz electricity during tests.

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

Table 2. Description of Tests Performed

Test #	Test Space[a]	Heat Exchanger Model[b]	Heat Exchanger Location	Heat Exchanger Fan speed	Baseboard Heat[c]	Furnace Fan[d]
1	RRH - 1RM	Mits	Central	Low	No	No
2	RRH - 1RM	Sharp	Central	Low	No	No
3	RRH - 3RM	Sharp	Central	High	No	No
4	RRH - 3RM	Sharp	Central	High	Yes	No
5	RRH - 3RM	Mits	Central	Medium	No	No
6[e]	RRH - 3RM	Mits	Central	Medium	Yes	No
7	RRH - 3RM	Sharp	End	High	No	No
8	RRH - 3RM	Sharp	End	High	Yes	No
9	RRH - 3RM	Mits	End	Medium	No	No
10	RRH - 3RM	Mits	End	Medium	Yes	No
11	WCH[f]	Mits.	Lvg. Rm.	Medium	No	No
12	WCH[f]	2-Mits[g]	Lvg. Rm. Mstr. Bdr.	Low Low	No	No
13	WCH[f]	Mits.	Lvg. Rm.	Medium	No	Yes
14	WCH[f]	2-Mits[g]	Lvg. Rm. Mstr. Bdr.	Low Low	No	Yes

[a]RRH = Radon Research House; WCH = Walnut Creek House;
1RM = one-room test; 3RM = three-room test.

[b]Mits = Mitsubishi Model VL-1500; Sharp = Sharp Model GV-120.

[c]Temperature differential of 20 ± 5 °C maintained between inside and outside.

[d]Furnace fan operated 10 minutes on/10 minutes off throughout decay.

[e]Radon gas used simultaneously with SF_6 as a tracer.

[f]Door to Bedroom 1 closed during test.

[g]Two Mitsubishi heat exchangers operating during test.

Table 3. Results of Ventilation Efficiency Tests at the LBL Radon Research House.

Test # Description[a]	Ts-Tin (°C)[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-M1C	2.5	0.23	1.45	2.01	0.61	40	0.54	0.61	0.67	0.46	0.62	0.61
2-S1C	1.1	0.14	1.22	1.64	0.65	34	0.72	0.72	0.63	0.63	0.69	0.65
3-S3C	1.5	0.07	0.49	0.74	0.57	54	0.63	0.54	0.66	0.57	0.63	0.53
4-S3CH	5.8	0.08	0.46	0.74	0.51	48	0.55	0.51	0.61	0.46	0.47	0.47
5-M3C	2.9	0.06	0.49	0.86	0.49	54	0.47	0.43	0.51	0.50	0.48	0.50
6-M3CH	8.2	0.10	0.51	0.86	0.47	52	0.45	0.50	0.47	0.52	0.46	0.49
7-S3E	1.4	0.06	0.47	0.74	0.55	52	0.42	0.61	0.40	0.49	0.75	0.54
8-S3EH	5.6	0.08	0.45	0.74	0.50	48	0.36	0.69	0.38	0.41	0.62	0.42
9-M3E	1.8	0.11	0.51	0.86	0.47	52	0.35	0.51	0.37	0.50	0.56	0.46
10-M3EH	4.0	0.09	0.52	0.86	0.50	55	0.48	0.49	0.49	0.46	0.52	0.48

NOTE: See page following Table 4 for footnotes.

Table 4. Results of Ventilation Efficiency Tests at the LBL Walnut Creek Research House.

Test #- 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
11-1M	0.12	0.41	0.48	0.64	70	0.81	0.69	0.77	0.59	0.80	-0.26
12-2M	0.12	0.47	0.56	0.66	86	0.80	0.73	0.88	0.61	0.73	-0.04
13-1MF	0.31	0.52	0.48	0.44	48	0.35	0.54	0.29	0.50	0.33	0.13
14-2MF	0.31	0.64	0.56	0.58	75	0.71	0.84	0.65	0.69	0.64	0.38

NOTE: See following page for footnotes.

Footnotes for Tables 3 and 4

[a]M = Mitsubishi Heat Exchanger (H.E.); S = Sharp 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; 1M and 2M 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 (air flow 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 air flow rate through H.E. if no cross-stream leakage or short circuiting 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 ventilation rate at six indoor locations divided by the predicted increase assuming perfect mixing.

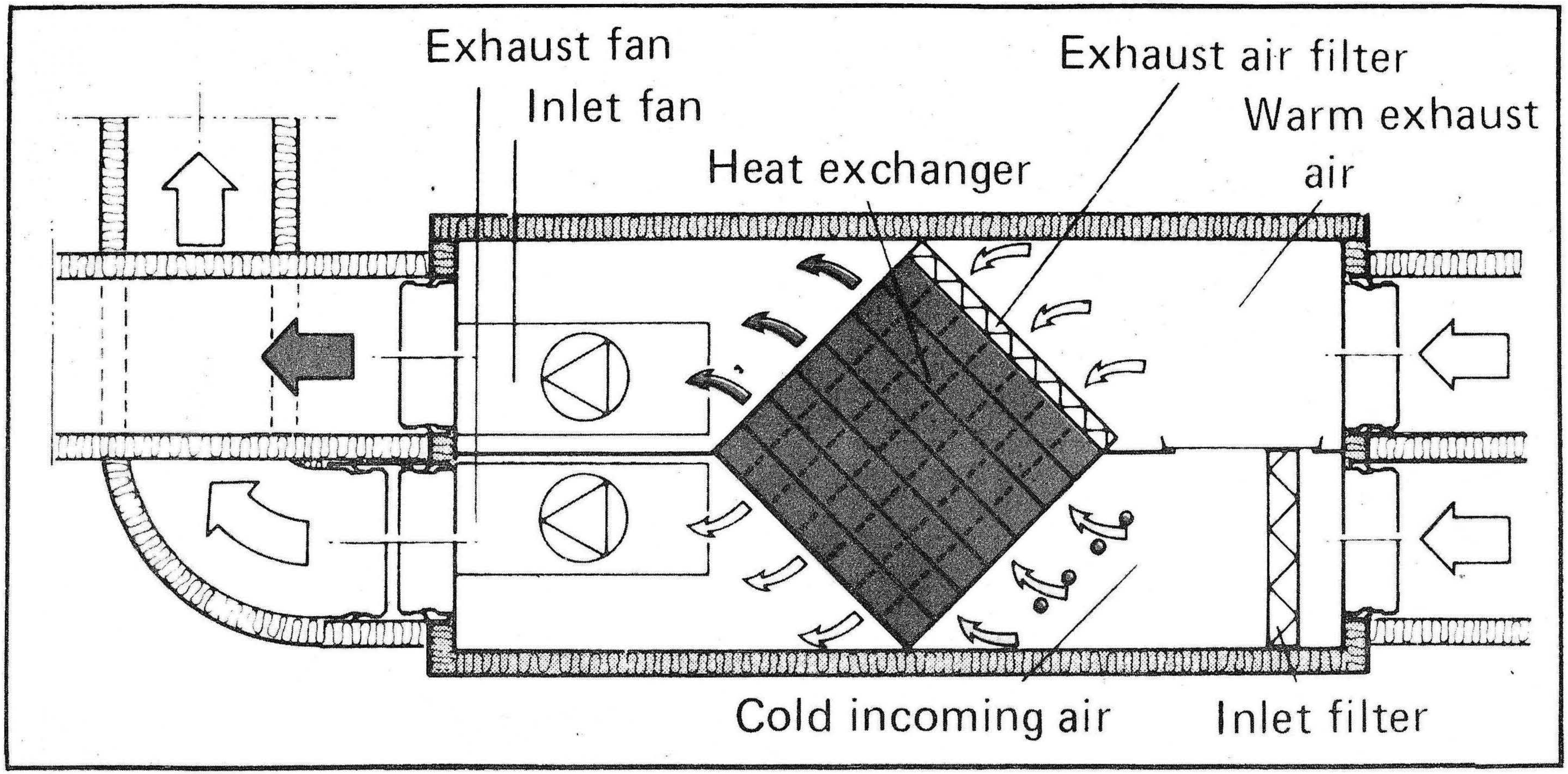
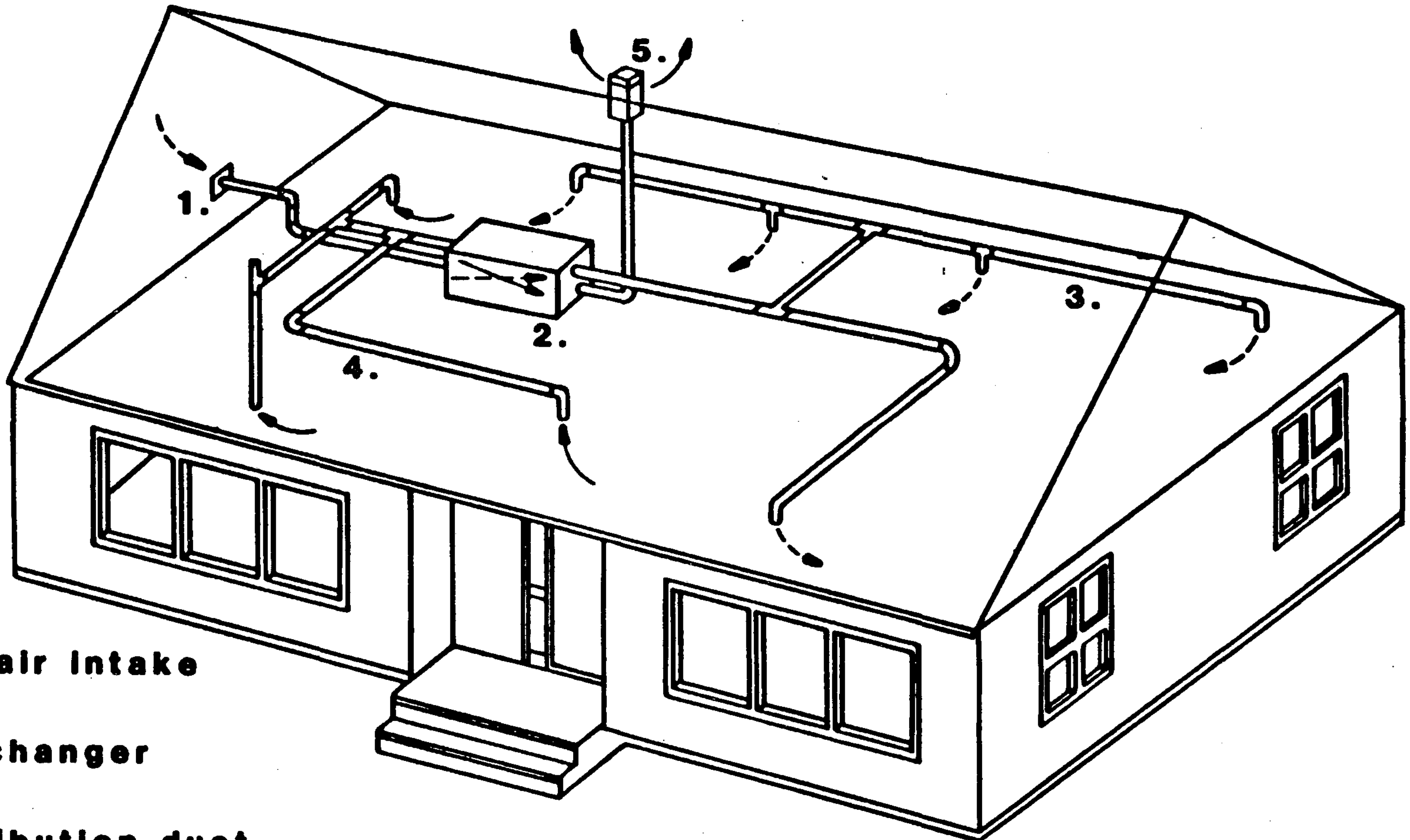


Figure 1. Schematic Diagram of an Air-to-Air Heat Exchanger.



1. Outside air intake

2. Heat exchanger

3. Air distribution duct

4. Exhaust air duct

5. Exhaust air vent

Figure 2. Fully Ducted Installation of Heat Exchanger.

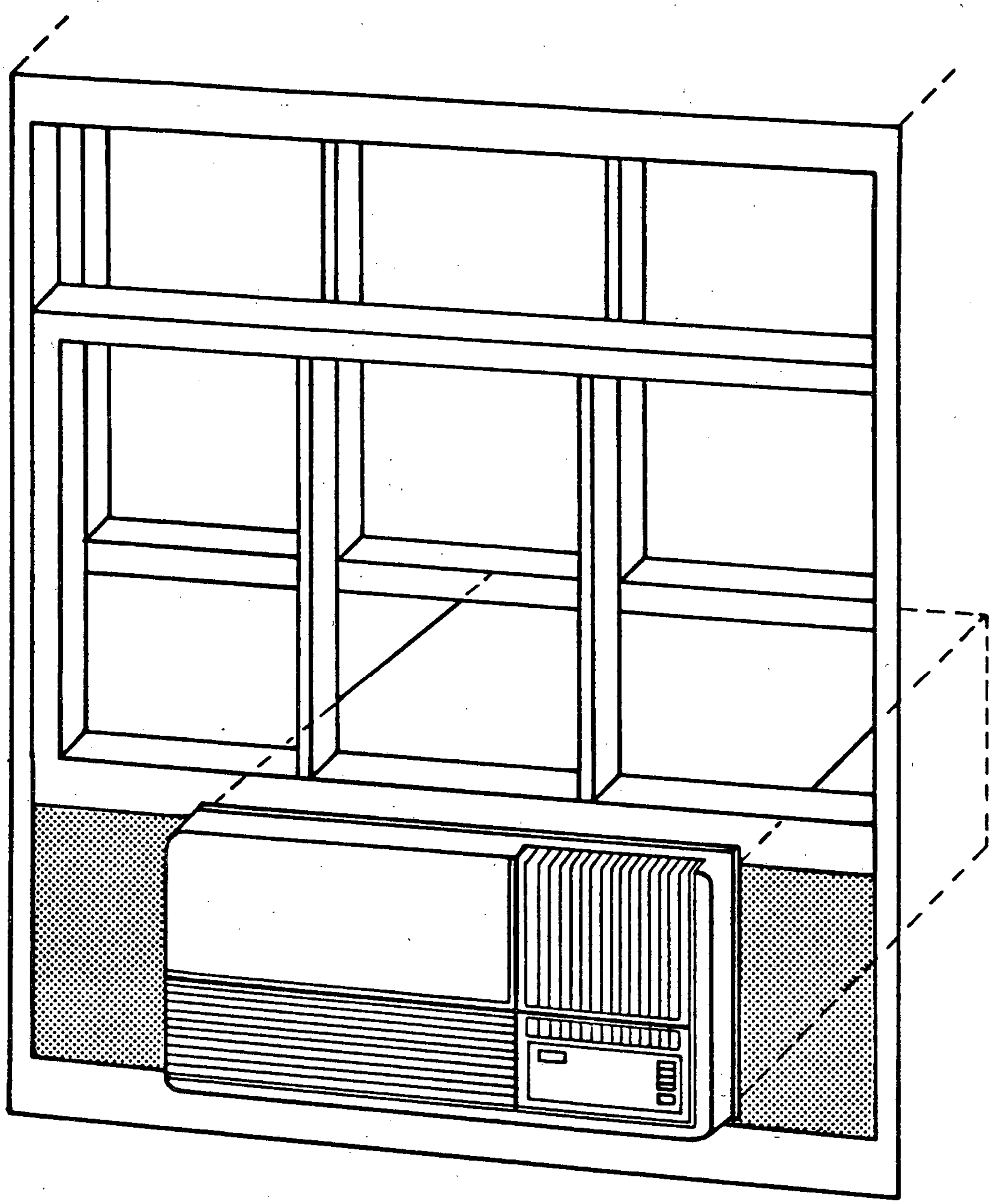
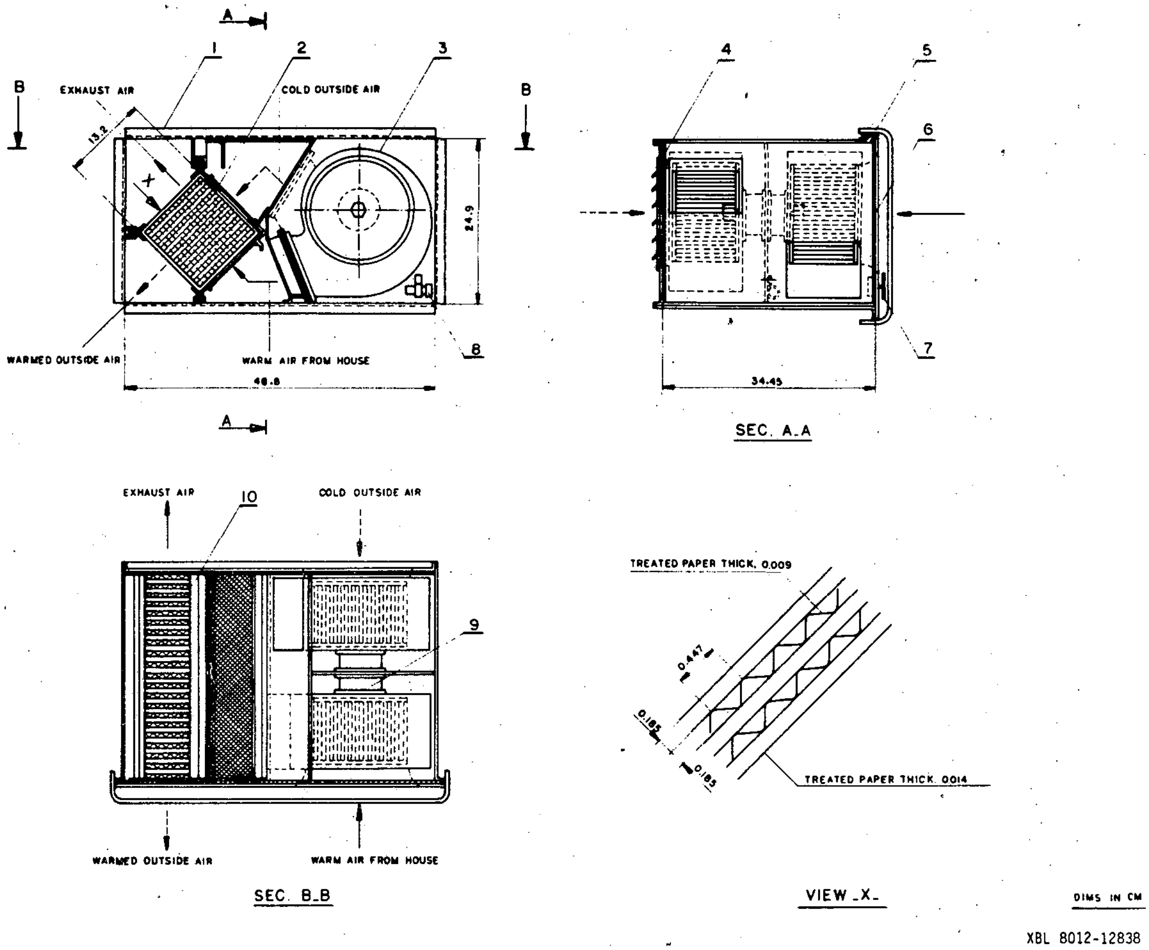


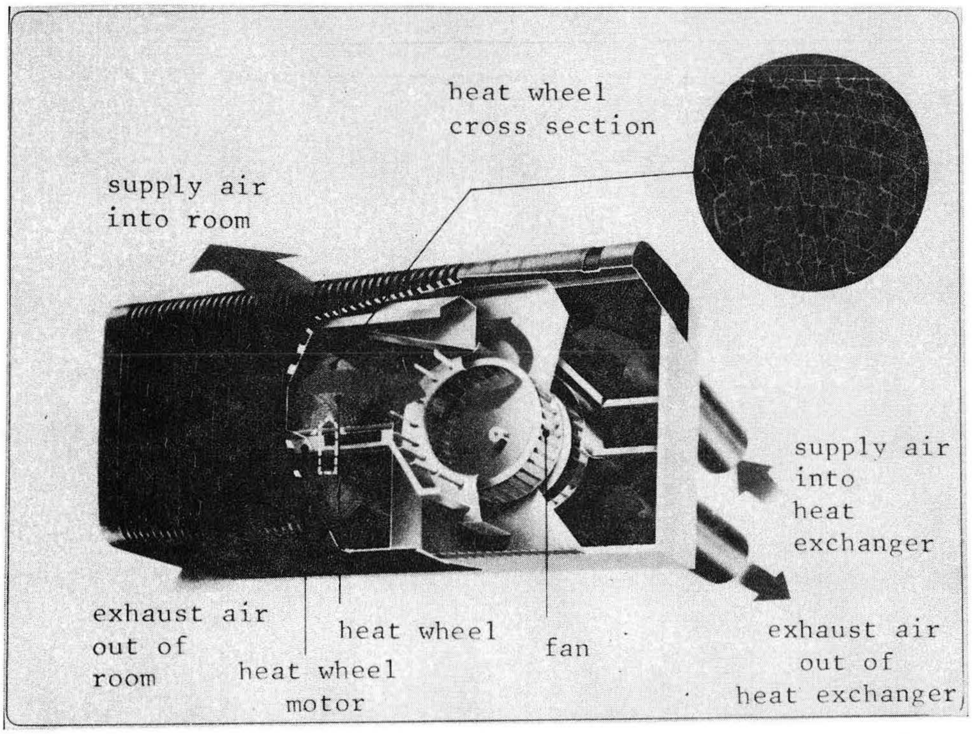
Figure 3. Window Installation of Heat Exchanger.



NOTE: Patented Product

1.	Case mounting flange	6.	Case, front section
2.	Supply air filter	7.	Fan speed indicator
3.	Fan housing	8.	Capacitor
4.	Insulation	9.	Fan motor
5.	Front cover	10.	Core

Figure 4. Mitsubishi Model VL-1500 heat exchanger.



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NOTE: Patented product.

Figure 5. Sharp GV-120 heat exchanger.

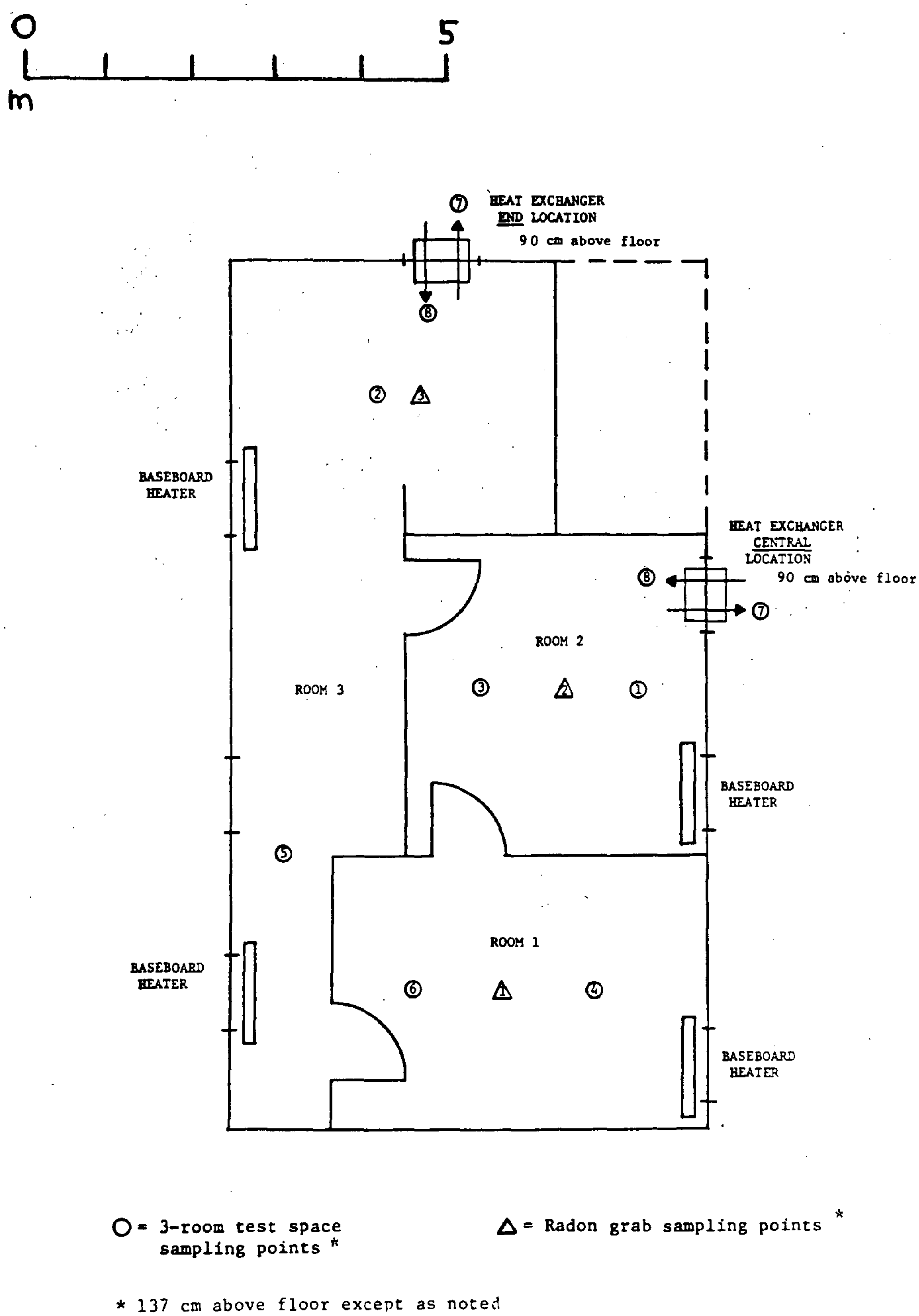
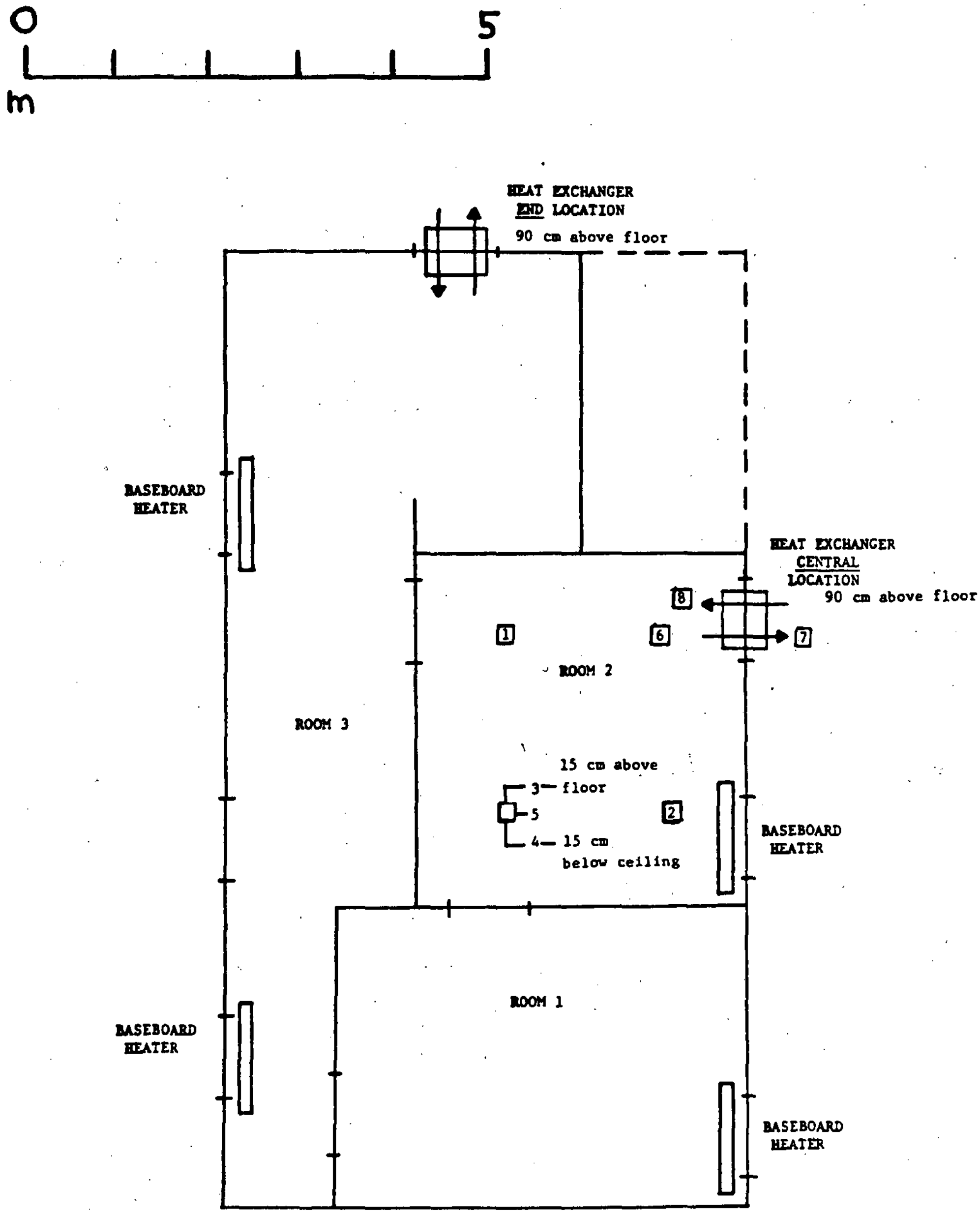


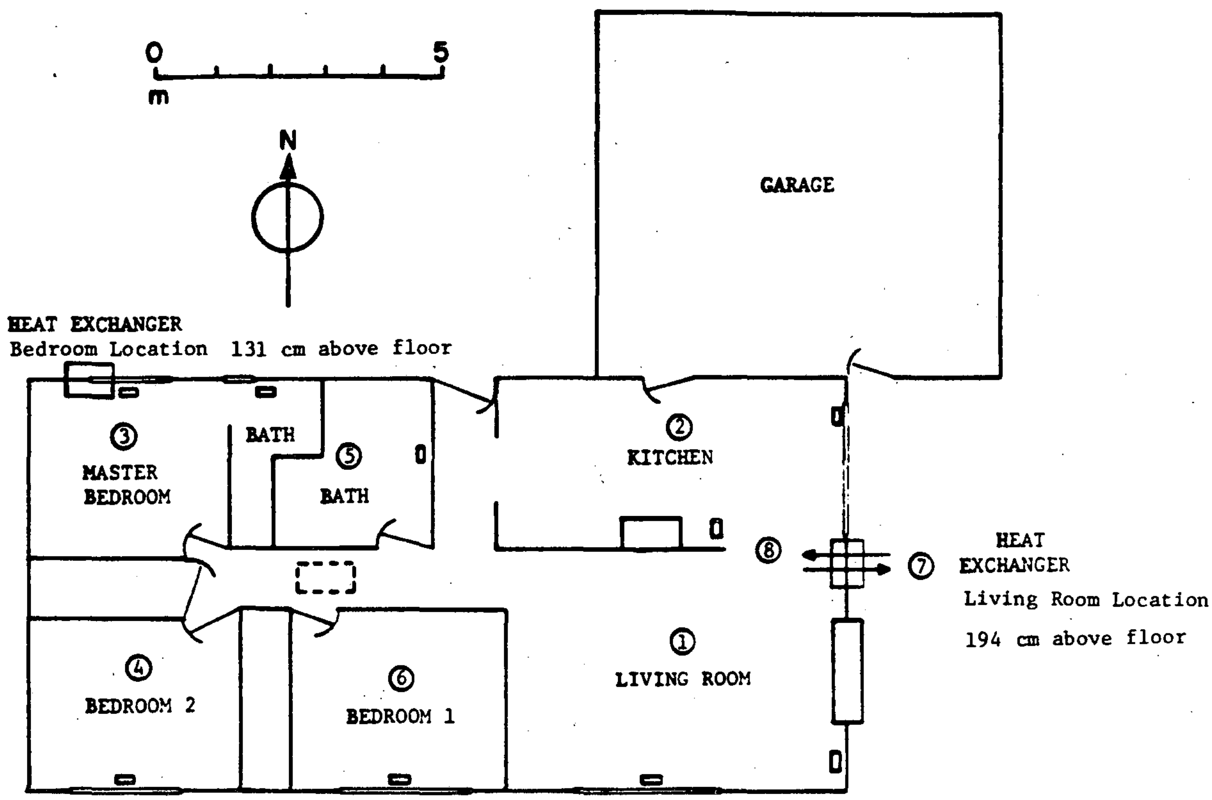
Figure 6. Radon Research House
Three-Room Test Pattern.



□ = 1-room test space sampling points *

* 137 cm above floor except as noted

Figure 7. Radon Research House One-Room Test Pattern



O = Sampling points*

* 137 cm above floor except as noted

Figure 8. Walnut Creek House.

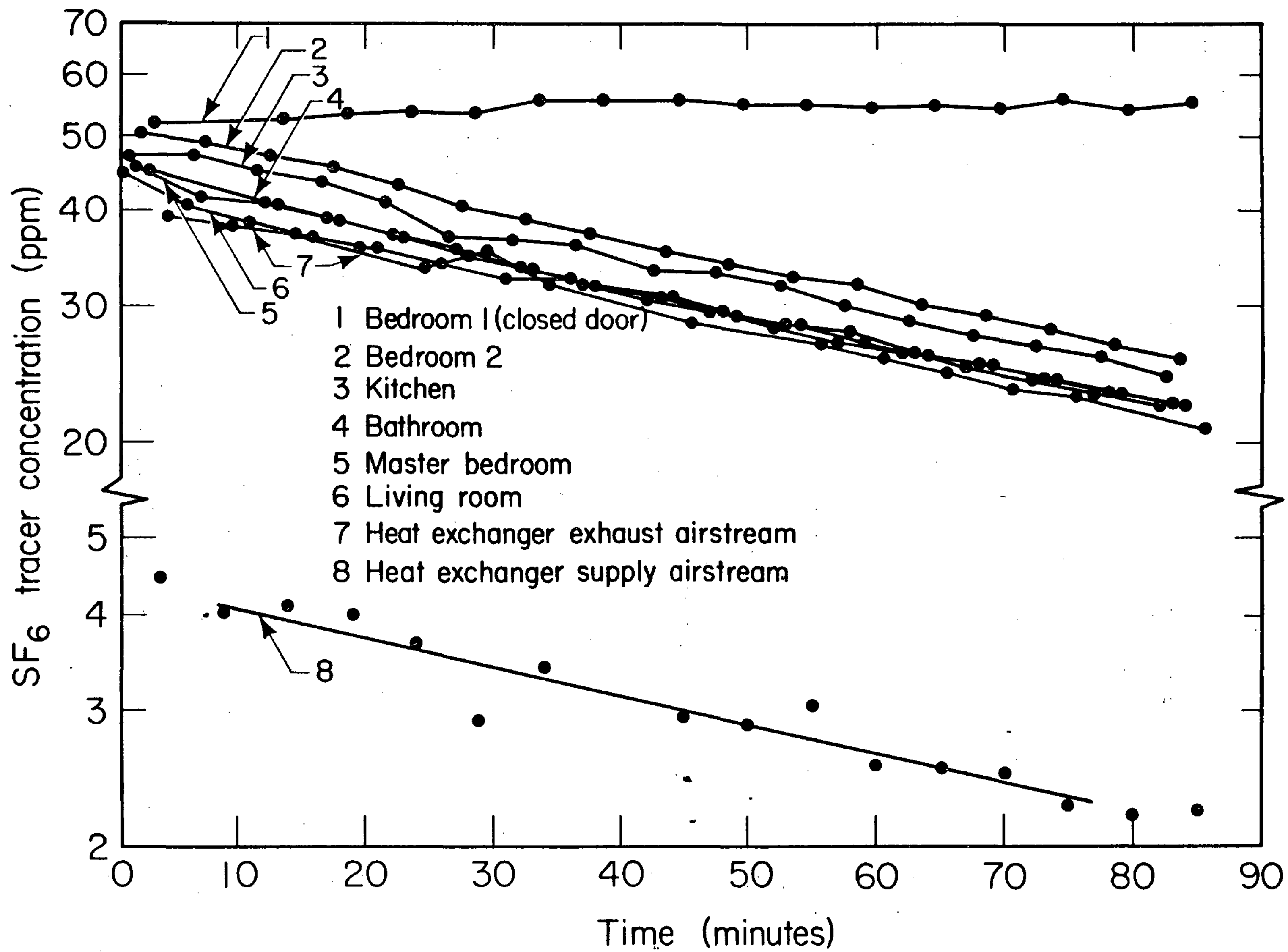


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Figure 9. A semi-logarithmic plot of tracer concentration versus time for 8 locations during test #11 in the Walnut Creek House (Mitsubishi heat exchanger operating at medium speed in a living room location and bedroom 1 closed).

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