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A prototype ventilation system for superinsulated houses using forced air duct systems

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ABSTRACT

The Energy Division of the Minnesota Department of Energy and Economic Development (DEED) recognizes the complex technical interrelationships between energy efficient building construction, design and operation of heating, ventilation and air conditioning (HVAC) systems, and the impact of these two factors on indoor air quality. In response, the Energy Division is working toward development of ventilation systems capable of providing acceptable levels of indoor air quality in superinsulated houses.

The first step in this effort was a competitive contract sponsored by the Energy Division and awarded to Honeywell Inc. of Minneapolis, Minnesota. The Honeywell research was designed to analyze and improve the indoor air quality of a superinsulated retrofit house located in St. Paul, Minnesota. The occupants had encountered "stuffy air" problems after their house was superinsulated and weatherized to reduce heat loss, and uncontrolled air infiltration. An analysis of indoor air quality and energy consumption patterns using real time monitoring techniques and dynamic computer simulation revealed high levels of CO_2 build up indoors, despite the presence of a continuously operating air-to-air heat exchanger. Several modifications were developed and installed on the existing HVAC system to improve air quality. The modifications consisted of a controlled ventilation system and recirculation of the air from an electric cooking stove after passing it through an electronic air cleaner. The other facet of the project used similar analysis techniques to assess the thermal and environmental performance of the new HVAC modifications.

Results of this research effort will aid the Energy Division in its continuing efforts to promote energy conservation and insure that acceptable indoor air quality is maintained in energy efficient houses. The building community will benefit by the development of appropriate technologies which can be effectively applied to superinsulated and other energy conserving houses.

INTRODUCTION

It has been shown that energy efficient thermal envelope designs are feasible, but maintaining effective ventilation and acceptable indoor air quality (IAQ) may be a major, unresolved problem. To address this issue, the Energy Division created a demonstration research project with an overall goal of developing and demonstrating cost effective and energy efficient control of IAQ in a superinsulated house using forced air duct systems. Specific objectives were to:

* Measure and evaluate concentrations of indoor contaminants during the spring of 1984.
* Simulate system performance by means of a verified, dynamic computer program.
* Identify HVAC control options that will improve IAQ and reduce energy consumption.

Research sponsored by the Minnesota Department of Energy and Economic Development. Analysis and diagrams prepared by the contractor, Honeywell Physical Sciences Center of Minneapolis, Minnesota.

* Install selected control options in the test house and re-evaluate the air quality and energy performance during the winter of 1985.

For the project, the Energy Division selected a St. Paul, Minnesota, home which had undergone a superinsulated retrofit during the summer of 1983. Built in 1957, the test house sits on a 6,750 square feet (627 m^2) urban lot. It is a one level, 971 net square feet (90 m^2) ranch-style house with a full, heated basement and detached garage. Figures 1-3 show the house and floor plan. Table 1 describes important occupant and lifestyle parameters.

Assessment of Existing Conditions

To assess the existing IAQ conditions in the test house, the characteristics of the house and the occupancy patterns of the family were determined by completion of questionnaires and daily logs. The infiltration rates, interzonal air exchange rates, and concentrations of indoor contaminants were determined using real time measurements during a 42-hour period in the spring of 1984. Table 2 lists the parameters measured and the instrumentation used.

Air contaminant concentrations in the residence were measured at two locations--the kitchen and a bedroom--and at one outdoor location. To make the indoor measurements, a manifold was constructed to bring air samples to the instrumentation (Figure 4). Once the sample passed through the instrumentation, most of the air was recirculated back to the sample location in the residence. The radon and particle measurements were made only at one location in the basement, which will generally be the area of highest radon concentration.

In addition to the real time contaminant measurements, integrated measurements were also made using palmes tube monitors for nitrogen dioxide (NO_2) over a one week period following the inital test. A tracer gas decay test was used to measure air exchange rates between the two rooms and the outdoor air. A dilute concentration of chemically pure 2ethane (CH_4) was introduced into the house, and the rates of decay of the concentrations were measured.

The air exchange measurements were made at the start of the testing period and at the end of the test period. The results of these two measurements indicate an air exchange rate of about 0.14 ACH for both the kitchen and basement zones during the initial period of the test. The air exchange rates were measured at about 0.24 ACH at the end of the testing period. The existing air-to-air heat exchanger operated continuously during these measurements and is rated to provide 0.5 ACH.

To assess whether outside weather conditions had an impact on the infiltration change, wind speed, direction and atmospheric pressure readings were obtained from the St. Paul Airport, which is located about five miles from the residence. The wind speed readings indicate that about 20 hours into the test the wind became extremely gusty (20-35 kts). At this same time, the wind direction started shifting from the south to the west-northwest. This change in wind speed and direction would have had a definite change in the air infiltration into the house.

The infiltration measurements pointed out a problem in the current ventilation plan of the house. The methane made its way outdoors primarily through the heat exchanger exhaust duct. This produced high outdoor levels (~100 ppm) at this location. Since the fresh air supply to the heat exchanger was also located near this point (about 6 feet away), exhaust air could easily re-enter the ventilation system depending on wind direction.

During the test, the HVAC fan would start when the air conditioner was switched on. The existing electronic air cleaner mounted in the return air duct would also operate during this period.

Results

The concentrations obtained for CO_2 during the test period are shown in Figure 5. The CO_2 results show very little difference between the kitchen and basement test zones. There was a gradual decreasing trend evident in both indoor and outdoor measurements. This might be attributed to weather conditions experienced during the test (e.g. increase in wind velocity and direction). The carbon monoxide (CO) measurements shown in Figure 6 indicate very little difference between indoor and outdoor levels. For CO, the indoor/outdoor ratio remained about 1.0 with some noticeable exceptions producing outside fluctuations. The fluctuations in CO might have been due to occupants parking their vehicles near the kitchen, allowing CO to be pulled into the house through the fresh air intake.

The measurements made in the basement den area for radon indicate very low levels (<0.002 WL) of radon decay products Figure 7). As a comparison, the ASHRAE recommended level or radon is 0.01 WL.

The particle measurements in Figure 8 show high peaks that probably occurred during times of smoking (four of the occupants smoke cigarettes). The average measured particle concentration was 113.20 ug/m^3. Available National Ambient Air Quality Standards are 75 ug/m^3 for an annual average and 260 ug/m^3 for a 24-hour period.

To obtain an estimate of particle generation from sources other than tobacco smoke, the two particle size regions (0.3 - 0.378 and > 3.0 microns) were plotted (Figure 9). Since tobacco smoke usually has an upper diameter of 2.0-3.0 microns, this figure shows the levels of other particle sources such as household dust or pollen.

Temperature and relative humidity readings are shown in Figures 10 and 11, respectively. The kitchen area temperature remained at about 84^{o}F (29^{o}C) with a basement temperature of about 75^{o}F (24^{o}C). The outdoor temperature, of course, varied with the time of day.

The relative humidity plot shows basement humidities ranging between 51-59 percent and lower kitchen humidities of about 30-40 percent. There is a large variation in the outdoor humidity measurements. Superimposed on the indoor humidity readings are oscillations in power usage due to the operation of the air conditioner, which can be seen cycling in Figure 12. If these are compared with the small oscillations in the indoor temperatures and relative humidities, a match can be seen.

Description of Modifications

Based on the results of air quality monitoring, computer simulations of the system performance, and visual inspection of central forced air heating, ventilating, and cooling systems, the following modifications were made:

1) Existing air-to-air heat exchanger was removed and an outside air dilution cycle installed which provides adiabatic mixing of outdoor and return air for thermal and air quality control. Computer simulations indicate that the dilution cycle should provide improved air quality conditions at about the same energy performance as the air-to-air heat exchanger. (Actual performance evaluations were made during February and March of 1985 after this printing).
2) Humidifier originally installed on the forced air furnace was removed.

3) Air was supplied to the downstairs bedroom and the downstairs den, which previously had no heating or cooling ducts.

4) Excess supply air to the other downstairs bedroom was blocked off.

5) Return air grilles and ductwork for child's upstairs bedroom, and new air supply duct for the basement bedroom were installed.

6) Removed existing kitchen range and range hood (which had been deactivated during the 1983 energy retrofit) and installed a new Jenn-air™ kitchen range with a built-in exhaust system. This exhaust system includes an electronic air cleaner and was revised to automatically provide outside dilution air if gaseous concentrations rise above acceptable levels (Figure 14).

7) Installed new ductwork for make-up and exhaust air to the mixing plenum to replace existing ductwork, which could have been causing self-contamination. The new ductwork provides make-up the south side of the house and then exhausts the air to the north side.

8) HVAC control strategies were modified to provide dehumidification in winter by diluting with outside air, and in summer by mechanical refrigeration; to activate the electronic air cleaner or the kitchen range upon increase of particulate concentrations; and to provide dilution control through the kitchen

range exhaust upon increases in gaseous concentrations.

The cost of modifying the original HVAC system was $2,000 U.S. The cost of replacing the range was considered as a separate item, as its function will be evaluated independently from the HVAC system. The range was $1050 U.S.

System Operation

The HVAC control system was modified to provide improved air quality with little or no impact on the energy efficiency of the system. Schematics of the control wiring diagrams are shown in Figures 13 and 14.

The control system has been designed to function in six separate climatic regions of the psychrometric chart as shown in Figure 15:

Region I: Cold/dry outdoor conditions. When the thermostat calls for heating, the furnace is energized and the electronic air cleaner is energized with the supply fan. If the room humidity is below the set point, the outdoor damper in the dilution control package remains in the minimally open position for combustion air, the exhaust damper closes and the return damper opens to recirculate the air. If the room humidity is above the set point, the outdoor and exhaust air dampers open and the return air damper closes to maintain a mixed air temperature at 55°F (12.8°C) and humidity ratio of 0.0036 lb_w/lb_a, or less (i.e. mixed air relative humidity not more than 40 percent at winter design conditions). If the room humidity is above set point and the thermostat does not have the furnace runnning, the supply fan is energized by the humidistat; if recirculation is not sufficient to lower the humidity below the set point in 15 minutes, the dampers modulate to provide mixed air at 55°F (12.8°C) at RH < 40 percent.

Region II: Cool/humid outdoor conditions. Thermostatic control is the same as for Region I. If the humidity ratio of the outdoor air exceeds 0.0036 lb_w/lb_a, the minimum amount of dehumidification available through this strategy is limited by the humidity ratio and temperature of the outdoor air. For example, 45 percent outdoor air (e.g., 265 CFM (125 l/s)) is required to maintain a mixed air temperature of 55°F (12.8°C) if the return and outdoor air temperatures are 71°F (21.8°C) and 35°F (1.7°C), respectively. If the outdoor air is saturated with water vapor, the minimum dehumidification available is 0.81 lb/hr (0.37 kg/hr). For colder outdoor temperatures, the difference in humidity ratios increases, thus increasing the dehumidification capability, although the percentage of outdoor air decreases. For warmer outdoor temperatures, the percentage of outdoor air increases, thus increasing the dehumidification rate assuming the outdoor air humidity ratio remains constant. When the humidity ratio of the outdoor air exceeds that corresponding to the set point of the humidistat, dehumidification cannot be achieved by dilution.

Region III: Isothermal conditions. When the outdoor temperature is above the winter and below the summer set points of the thermostat, the furnace and the refrigeration system are shut down. However, if the set point of the humidistat is above the humidity of the outdoor air, removal and dilution control with outdoor air is provided.

Region IV: Warm/dry outdoor conditions. When the room air temperature exceeds the first stage summer set point, the humidity is below the set point of the humidistat, and the enthalpy of the outdoor air is below the loci of set points of the outdoor dilution cycle control, sensible cooling is provided by dilution with outdoor air supplied at low fan speed.

Region V: Warm/humid outdoor conditions. When the room air temperature exceeds the first stage summer set point and the humidity is above the set point of the humidistat, the dampers in the dilution package are de-energized (i.e., minimum outdoor air provided), the refrigeration system is energized, and air is supplied at low fan speed to decrease the sensible heat ratio for improved dehumidification and energy efficiency.

Region VI: Hot outdoor conditions. When the room air temperature exceeds the second stage summer set point, the dampers remain de-energized and air is supplied at high fan speed to increase the sensible heat ratio required for sensible and latent cooling at summer design conditions.

Schematics of the air mixing plenum, and the HVAC system before and after the modifications, are shown in Figures 16, 17 and 18, respectively.

Sequence of Operation

(Figures 13, 14)

1.0 Heating Cycle

1.1 Temperature Control. Upon a drop in room temperature below the heating set point t_{dbh}, thermostat T-1 energizes gas valve V-1, if limit switch SW-1 is energized.

1.1.1 If HVAC fan switch is set on automatic at T-1, the fan will be energized on low speed when fan switch SW-2 closes (i.e., when supply air temperature exceeds 105°F (40°C)).

1.1.2 Upon an increase in room temperature, thermostat T-1 de-energizes the gas valve V-1, and the fan will de-energize when the fan switch SW-2 opens (i.e., when supply air temperature decreases below 90°F (32°C)).

1.2 Air Cleaner. The electronic air cleaner will be energized whenever the fan is operational.

1.3 Humidity Control

1.3.1 If the room humidity is below set point, the outdoor air damper will remain in its minimum position for combustion air, the exhaust air damper will be closed, and the return air damper will be open (Region I on the psychrometric chart in Figure 15).

1.3.2 If the room air humidity increases above the set point (i.e., Region II on psychrometric chart in Figure 15), the humidistat RH-1 will be energized.

1.3.2.1 The outdoor and exhaust dampers will modulate open, and the return air damper will modulate closed through relay contacts R-2, R-4 and through mixed air thermostat T-2 set for a range of 25-55°F (-4-12.8°C).

1.3.2.2 The HVAC fan will operate on low speed through relay R-3.

2.0 Cooling Cycle

2.1 First Stage. Upon a rise in room dry bulb temperature above the cooling set point t_{dbc}, thermostat T-1 will be energized for the first stage of cooling.

2.1.1 Enthalpy Control. If the enthalpy of the outdoor air is below the set point of the controller, H-1 and the humidity is below the set point of RH-1. (i.e., Region III on the psychrometric chart in Figure 15):

2.1.1.1 The outdoor and exhaust dampers will open, and the return air damper will close through R-2 and mixed air thermostat T-2.

2.1.1.2 The HVAC fan will operate at low speed through relay R-2.

2.1.2 Humidity Control. If the outdoor air enthalpy is above the set point of controller H-1 and the humidity is above the set point of RH-1, (i.e., Region IV on the psychrometric chart in Figure 15):

2.1.2.1 The outdoor and exhausts air damper will be closed, and the return air damper will be opened.

2.1.2.2 The HVAC fan will operate at low speed through relay R-3.

2.1.2.3 The refrigeration system will be energized through contact C-2.

2.2 Air Cleaner. The electronic air cleaner will be energized when the fan is operational.

2.3 Second Stage. Upon a continued rise in room temperature, thermostat T-1 will be energized for the second stage of cooling (i.e., Regions V and VI on the psychrometric chart in Figure 15):

2.3.1 The enthalpy control circuits will be de-energized.

2.3.2 The HVAC fan will operate at high speed through the Fan Center Relay R-1.

2.3.3 The refrigeration system will be energized through contact C-1.

3.0 Kitchen Range Exhaust Control

When the kitchen range exhaust is energized through manual switch SW-3, the internal air cleaner and fan will be energized.

3.1 Normal Operation. The air will circulate through the electronic air cleaner and be returned to the kitchen space.

3.2 Gaseous Contaminant Control. Upon an increase in gaseous contaminants, gas sensor G-1 will be energized.

3.2.1 The air from the range will be locally exhausted by damper and motor which are energized relay R-5.

3.2.2 The outdoor and exhaust damper in the HVAC system will be opened, and the return air damper will be closed through relay R-5.

For purposes of this project, the controls for this device were provided by Honeywell.

Thermal and HVAC Modeling

The superinsulated test house was modeled within the Honeywell General Engineering Modeling and Simulation (GEMS) dynamic computer model structure, based on the house plan and construction information provided by the Energy Division. The existing heating, cooling and distribution system was modeled as the two zone heat delivery system with single zone thermostat control. In the model, the air distribution flows were balanced to simulate thermal conditions observed in each of two distribution zones--main floor and basement. The existing air-to-air heat exchanger was modeled based on measured flows rather than rated flows due to significant flow reductions experienced as a result of air-to-air heat exchanger installation. Furnace cycling under a constant -20°F (-29°C) design condition was plotted to verify that the models were properly interconnected and that space temperature control was within the normally predicted range. The estimated annual heating requirement of the model system was then calibrated to actual fuel consumption for the home. Calibration was achieved by minor adjustment to the modeled level of basement insulation. A model of moisture transfer from zone to zone, from indoor/outdoor air exchange, and from humidification and dehumidification of HVAC equipment was also included.

Air Quality Modeling

A two compartment air quality model was developed, with one compartment representing the basement and the other main level space (Figure 17). The effects of infiltration (including flue stack flow, indoor/outdoor temperature difference, and wind driven sources), mechanical exhaust, interzonal mixing, internal air contaminant generation, removal, and HVAC system cycling were all considered in the model. For the reference case the kitchen range exhaust fan was considered to be not in operation. Contaminant generation from people and processes was handled by inputting a daily schedule of occupancy and activity periods. This was developed based on assumptions of life style, or entered directly when the occupant pattern for a specific case is known.

Open loop simulation of this model was performed to verify that interactions predicted by the model are realistic with actual data. Some interactions are not verifiable by field data, since in real situations the influencing factors cannot be controlled. Specifically, open loop tests were performed where each of the parameters influencing space concentration in the model were studied independently. These included interzone mixing, infiltration, mechanical exhaust, air-to-air heat exchanger operation, and natural air infliltration. Plots of actual concentrations of CO_2 were generated to verify that the dynamic space concentration response to step changes in the parameters were properly represented. Once confidence in the model was established, the thermal and air quality models were integrated so that closed loop overall system interactions could then be examined.

Simulation Methodology

To achieve the program goal of evaluating candidate air quality improvement techniques relative to the existing air-to-air heat exchanger, a matrix of computer simulation runs were established. Having identified a mixed air control system which seemed capable of meeting the goals of providing acceptable indoor air quality with a minimum of energy expended, this became the base line control system for study with a number of implementation options to be evaluated.

Simulation Results

Preliminary results verify that there are indeed situations where the existing air-to-air heat exchanger would not have maintained acceptable air quality, as was actually reported by occupants in the house. Therefore the comparison of candidate strategies cannot directly be compared to the base line system from an energy standpoint alone, since that base line system does not always meet the minimum requirement of acceptable indoor air quality and moisture control. The energy savings or penalty of candidate strategies can only be compared when the system first meets minimum air quality requirements.

The computer runs demonstrated that the introduction and proper distribution of 0.5 air changes per hour (ACH) of outside air is required to maintain acceptable indoor air quality. This brings the modeled residence into compliance with both ASHRAE and Swedish National Standards for minimum acceptable air quality, and establishes the basis for mixed air control comparison.

Conclusions and Recommendations

From the testing, it can be concluded that contaminants such as NO_2, HCHO, and possibly even CO may not be much of a concern in this particular superinsulated house. This statement does not hold true, of course, for all superinsulated houses. High levels of CO_2 were observed initially in the test (>1000 ppm), and this may explain the stuffiness perceived by the occupants. This stuffiness, coupled with what appears to be good mixing within the house and very low air infiltration rates with the outside, could possibly be the cause of occupant concern.

The particle level, on the average, would probably put the occupants at an exposure greater than the EPA allowable annual averages; however, the 24-hour standard was not reached. It should be remembered that the air cleaner was in operation during much of the test period and that the particle levels might have been much greater near the spot sources of generation. The operation of the air cleaner could also account for the low radon progeny concentrations found. From the data, it would appear that the temperatures in the kitchen are on the high side, around 85°F (29°C) average. This could also contribute to the uncomfortable feeling of the occupants.

This study provides evidence that if the indoor air quality impact of energy retrofits is not properly controlled, potential energy savings may not be realized because of the health risks perceived by the occupants. Also evident is that CO_2 concentrations in residences may be sufficiently high for concern, that periods of maximum exposure to indoor contaminants do not necessarily occur at times of maximum heating or cooling requirements, and that air-to-air heat exchangers may not provide sufficient dilution during moderating seasons in certain occupant situations. Thus, if the potential for energy savings is to be maximized, control strategies that are also responsive to changing indoor air contaminant levels are required.

The performance of the HVAC modifications during the winter season are currently being evaluated and were not available for inclusion in this paper.

Figure 1. South elevation.

Figure 2. North elevation.

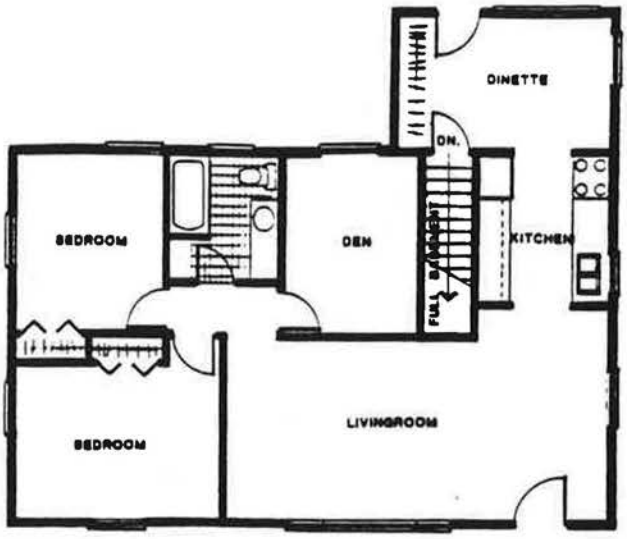

Figure 3. Floor plan.

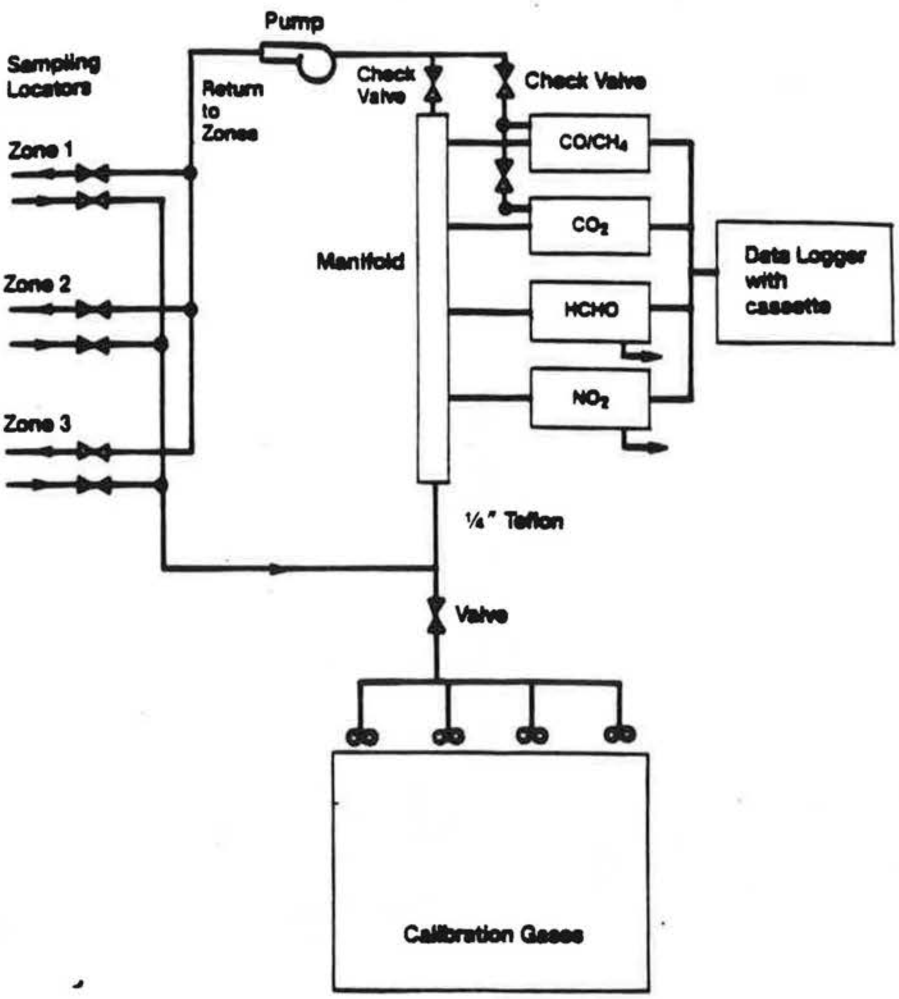

Figure 4. Data sampling system.

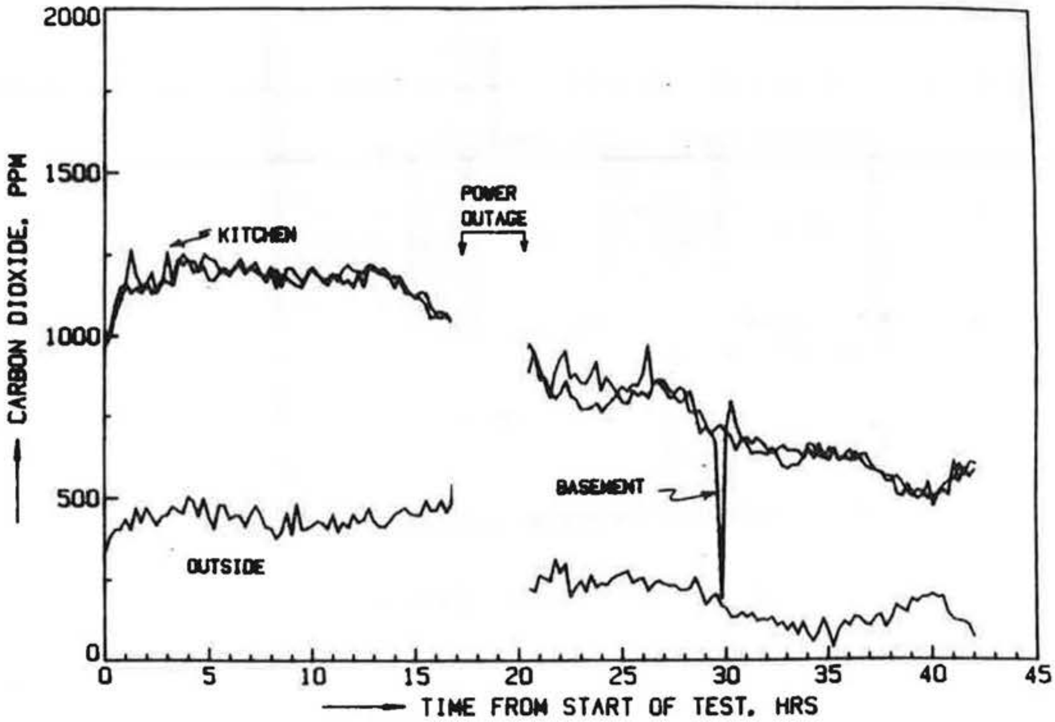

Figure 5. Carbon dioxide measurements.

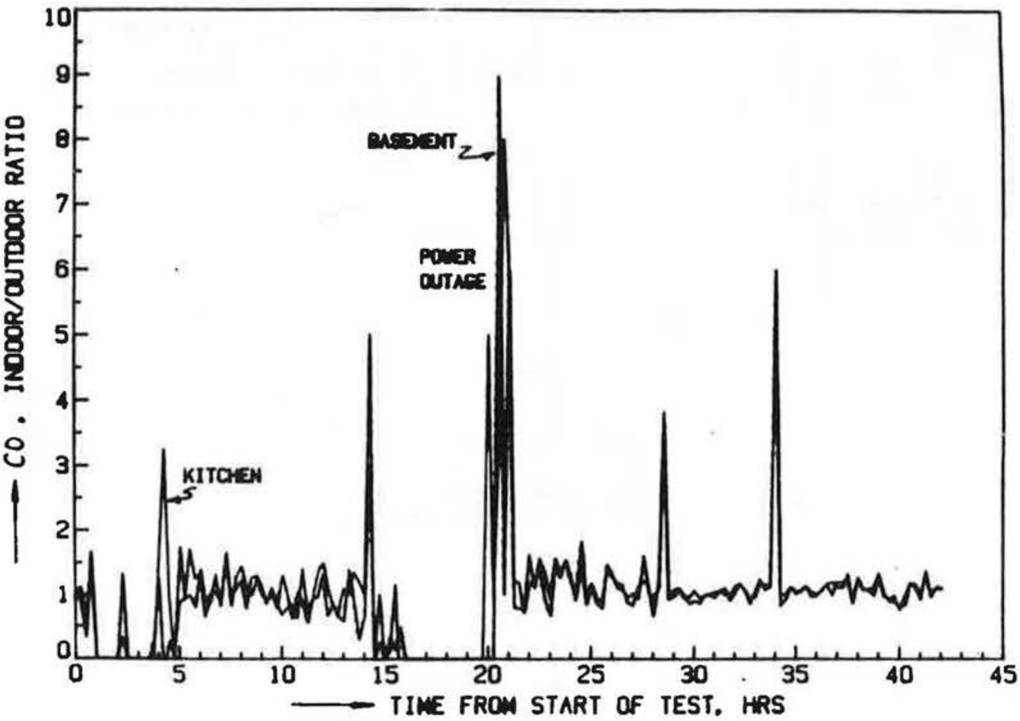

Figure 6. Carbon monoxide indoor/outdoor ratios.

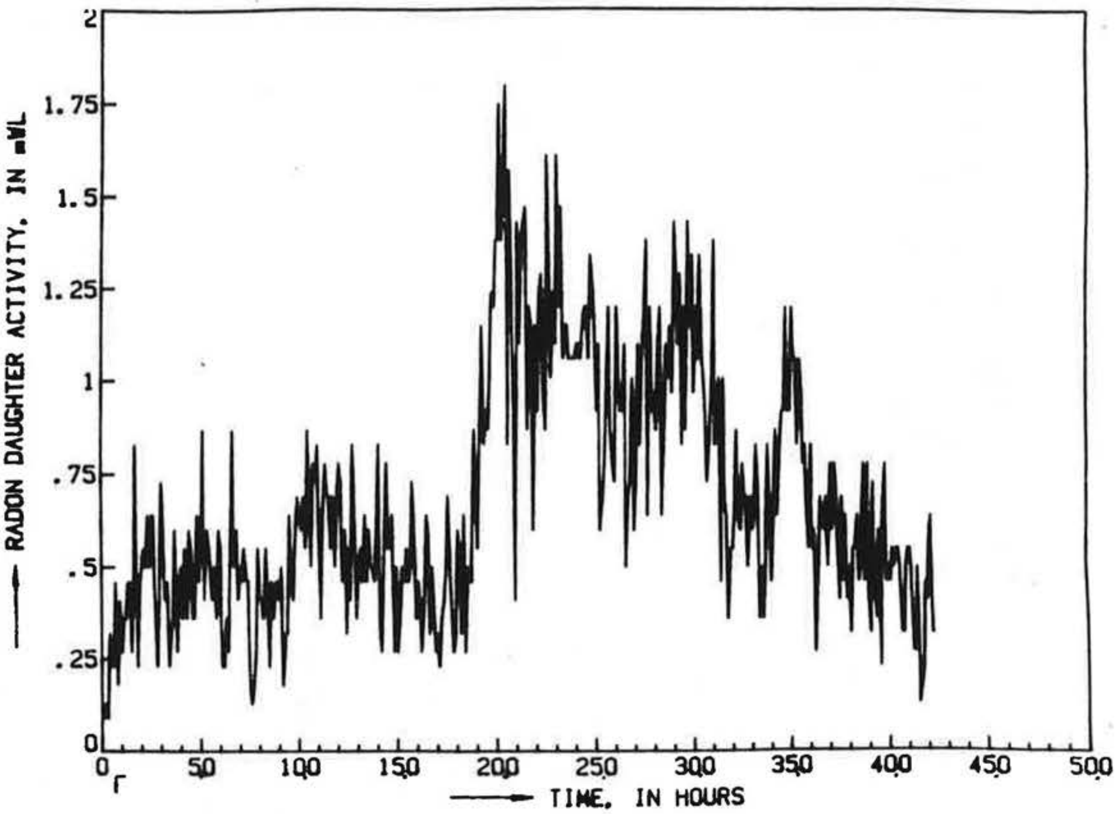

Figure 7. Radon progeny activity measured in basement.

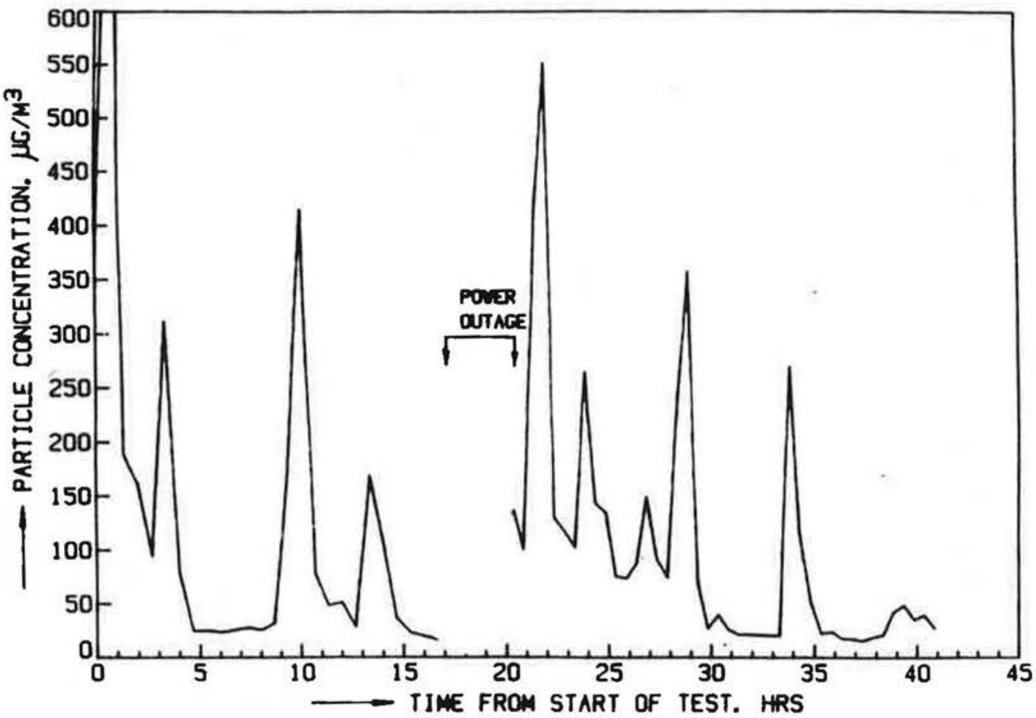

Figure 8. Particle concentration.

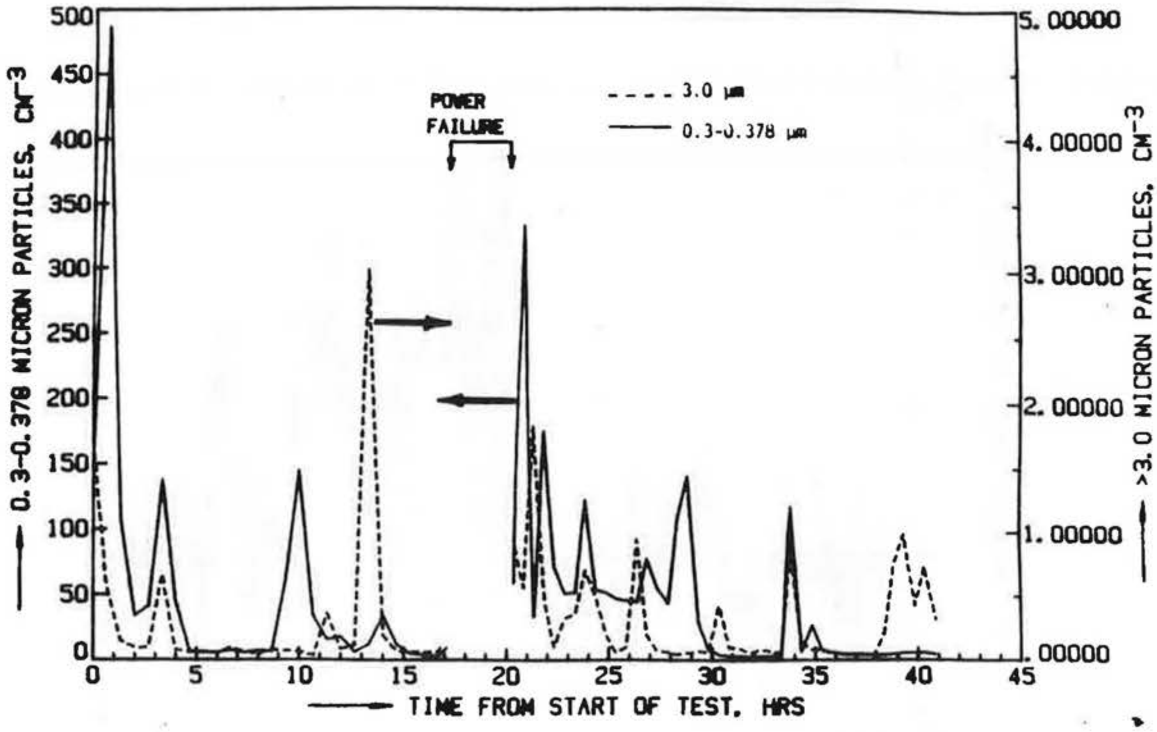

Figure 9. Particle concentrations for two size fractions, 0.3 - 0.378 microns and)3.0 microns.

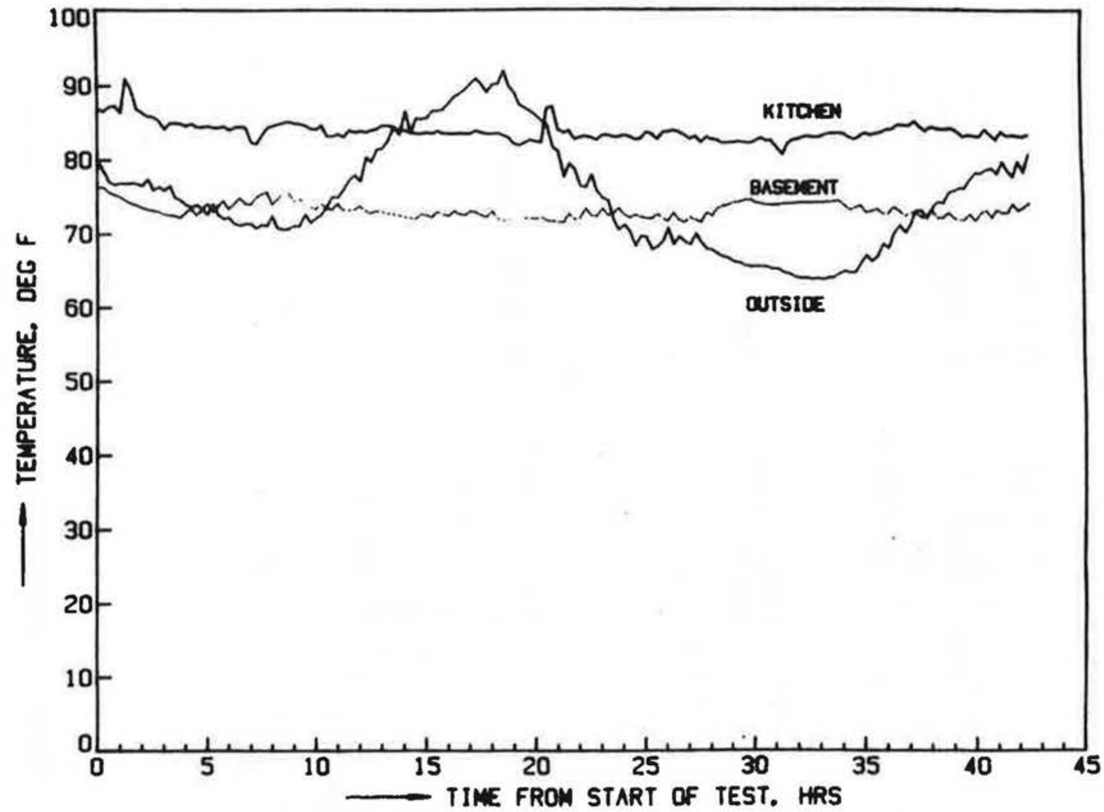

Figure 10. Temperature measurements.

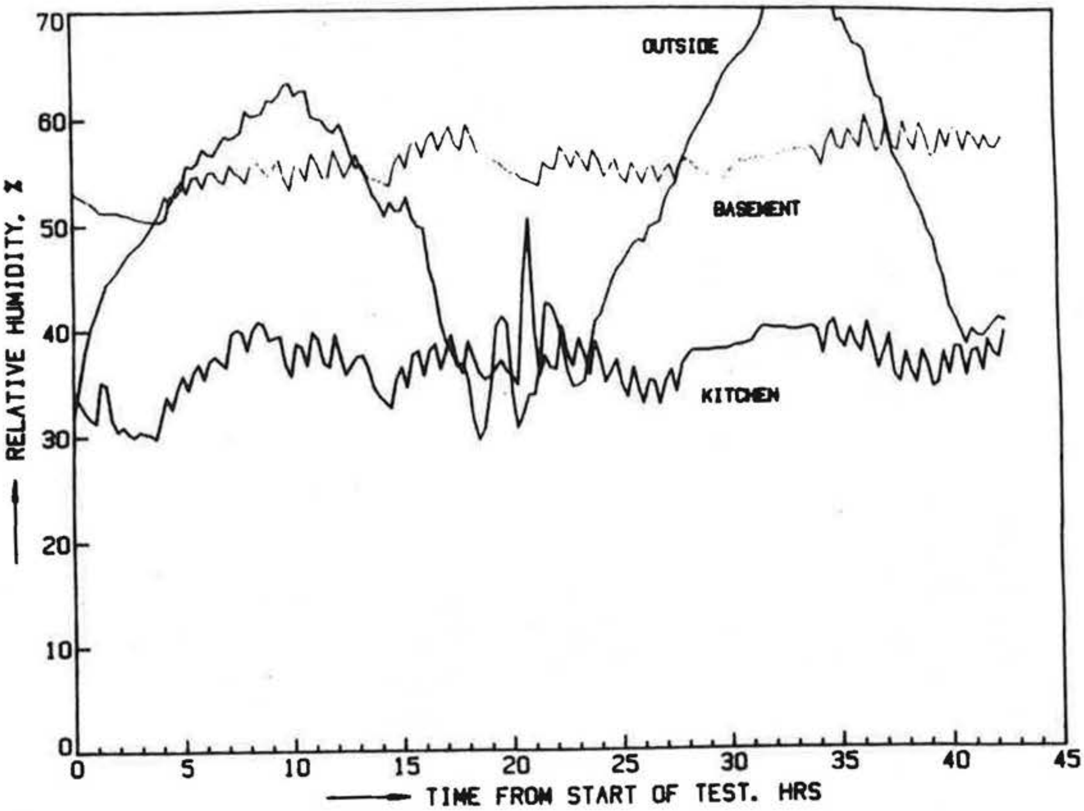

Figure 11. Relative humidity measured.

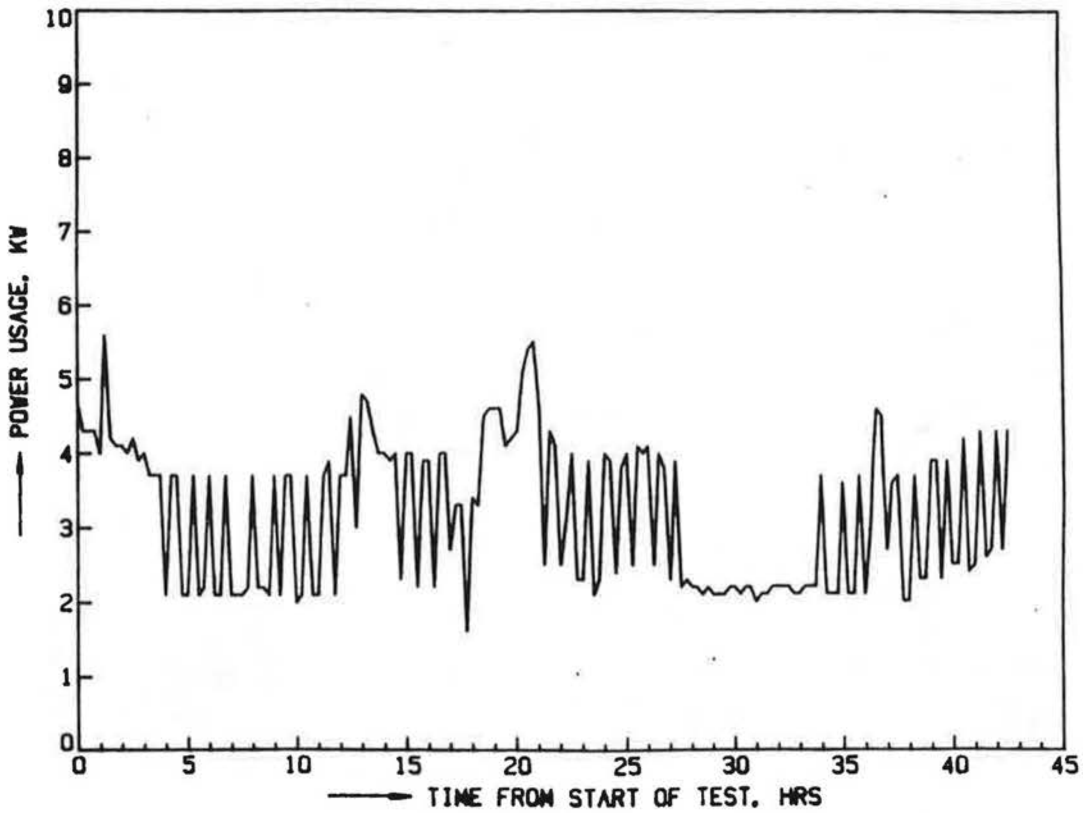

Figure 12. Power usage during measurement period.

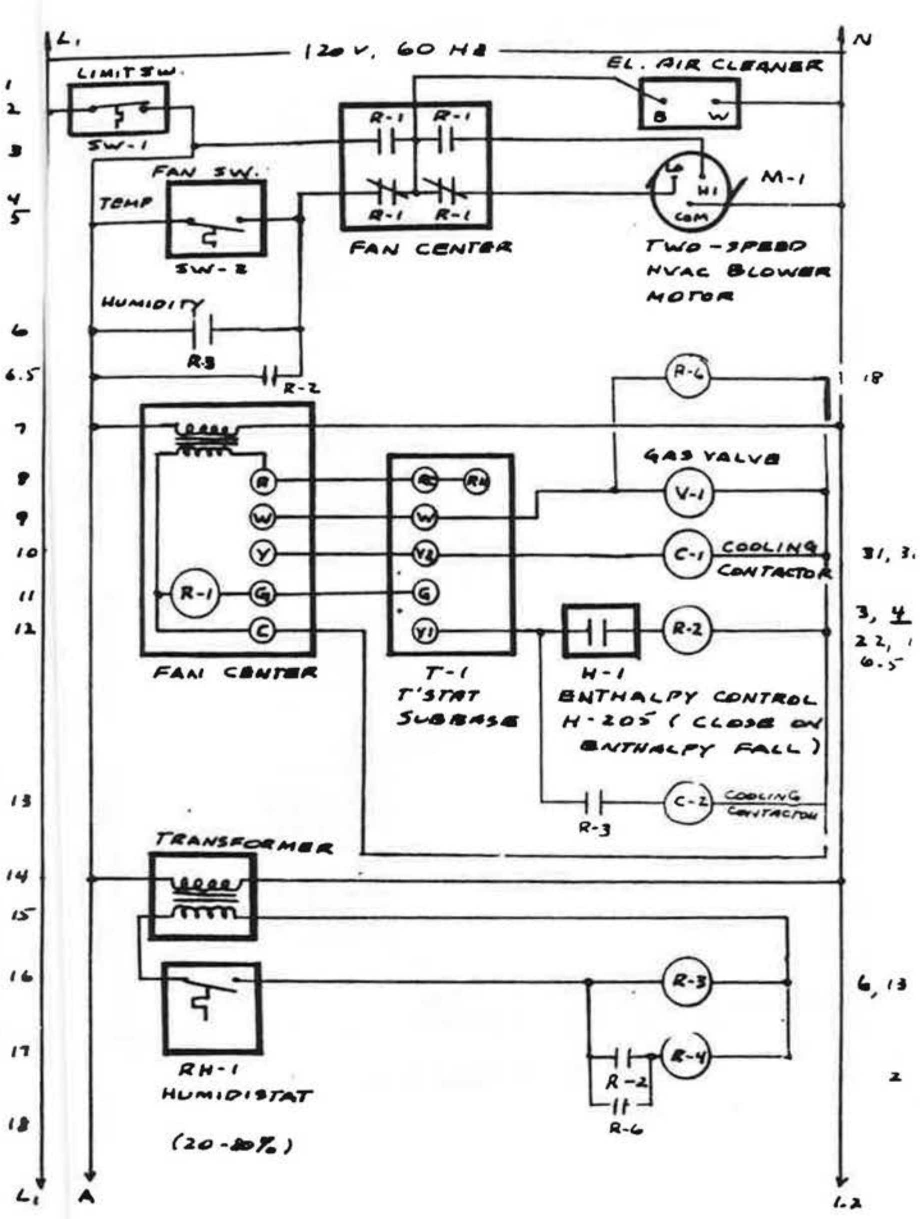

Figure 13. Control ladder diagram.

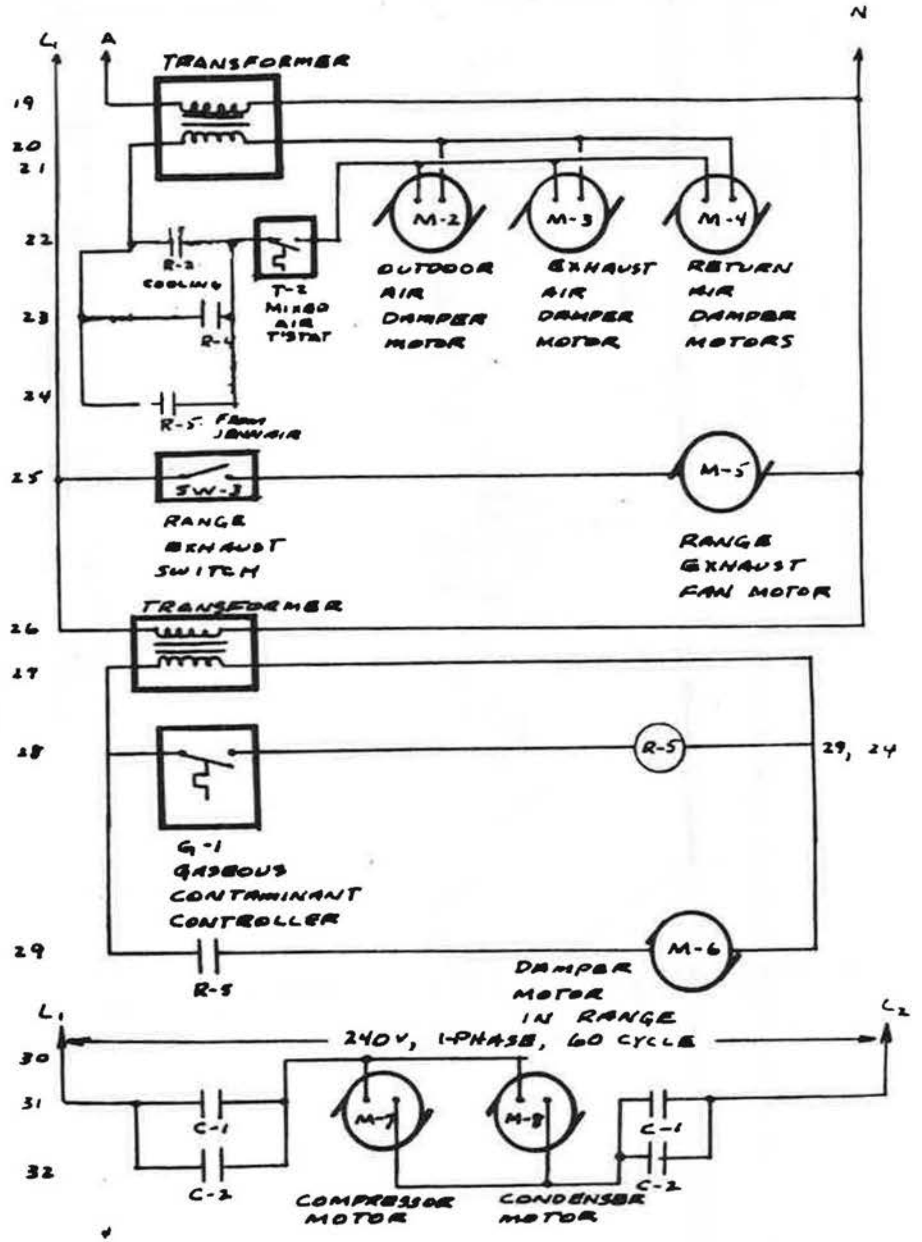

Figure 14. Control ladder diagram.

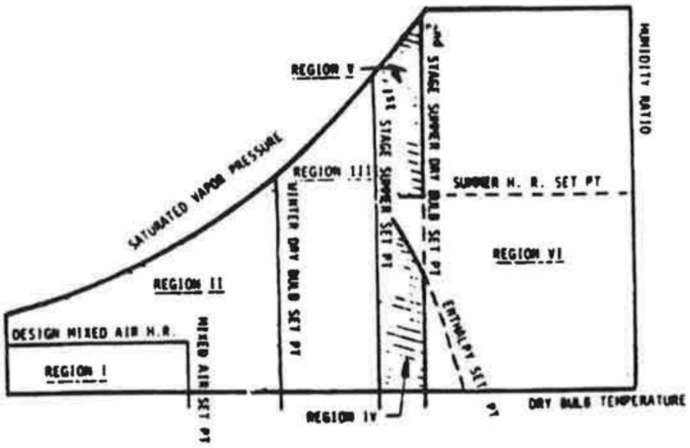

Figure 15. Enthalpy regions of outside air (Woods).

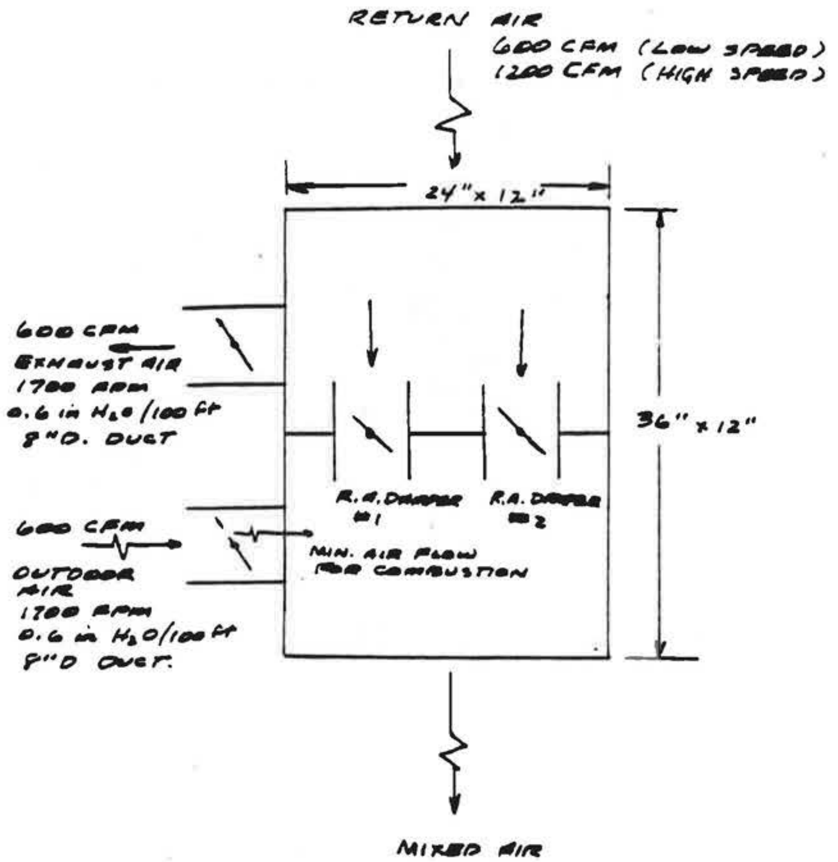

Figure 16. Schematic of air mixing in plenum-system.

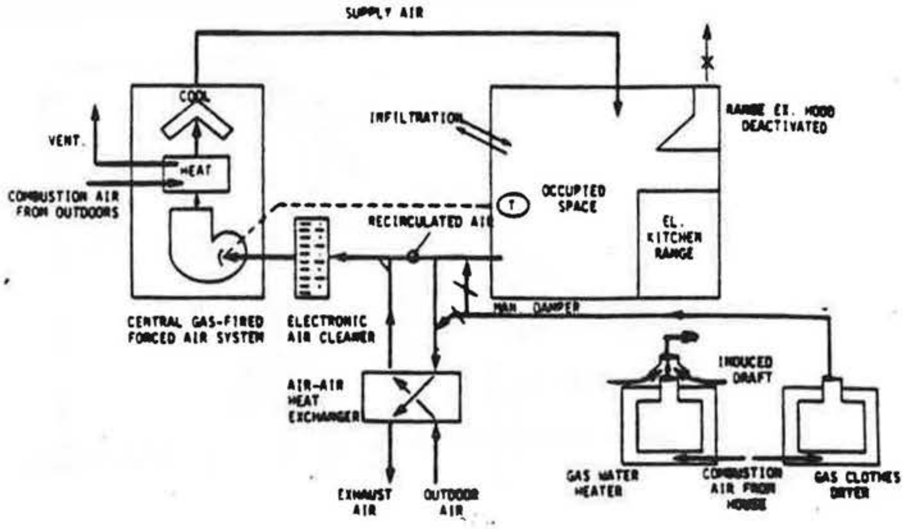

Figure 17. Schematic of system before IAQ modification (Woods)

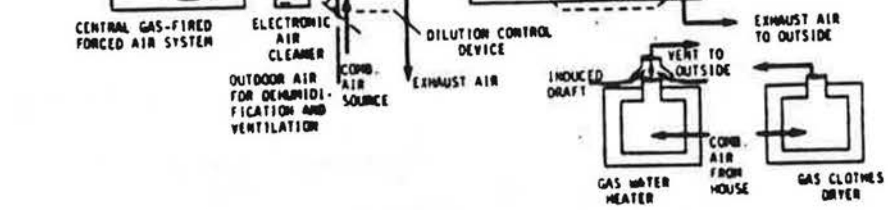

Figure 18. Schematic of system after IAQ modifications (Woods)

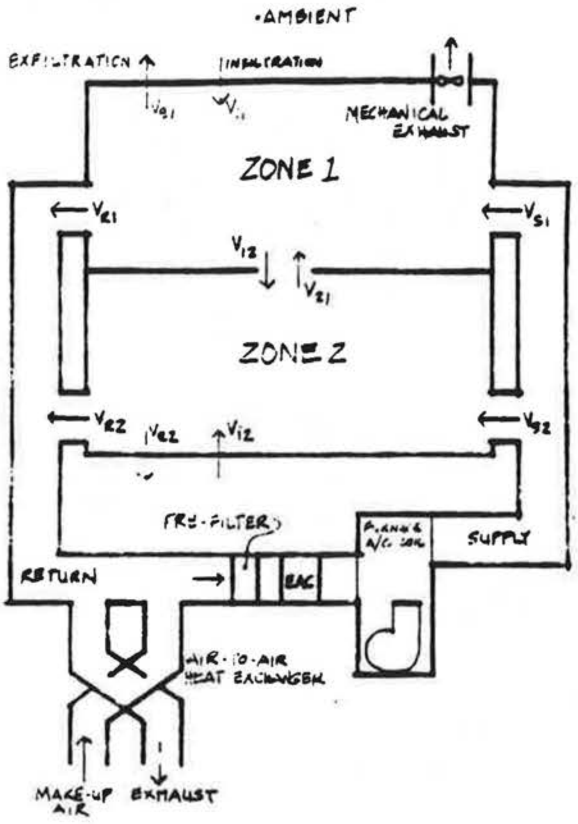

Figure 19. Air Quality model schematic of components and flows considered in the computer model (Baseline case).

Table 1. Occupant and lifestyle parameters.

Occupants:

Husband	Age 42	Employed Daytime
Wife	Age 40	Homemaker/Part-Time Employment
Son	Age 19	Work Days
Son	Age 18	Work Days
Daughter	Age 15	School Days
Daughter	Age 10	School Days

No pets

Approximately 40 showers/baths per week
Approximately 13 cooked meals per week

Appliances

Gas:	Furnace/40 gallon uninsulated water heater; dryer
Electric:	Stove; refrigerator; dishwasher; clothes dryer; central air conditioner; in plenum humidifier; microwave oven

Approximate number of loads of dishes each week:	10
Approximate number of loads of clothes washed each week:	
In hot water	10
In warm water	10
Approximate number of loads of clothes in dryer each week:	15
Woodburning stove or fireplace:	None

Problems:

Home damp with condensation on windows.
Air stuffy.
House overheats in winter. Remedy used was to block off one of the gas burners.
Air humid, cooking odors linger. Occupants often open doors and windows to air out house.
Heat exchanger fan runs continuously; yet poor air circulation through duct system.

Table 2. Data collection instrumentation.

Parameter to be Measured	Instrumentation
NO_2	Thermo Electron Co. Model 14A
CO	Infrared Industries CO Monitor
CO_2	Anarad CO_2 Monitor
Radon	EDA Working Level Monitor Model WLM-300
Particle (>.3 μm)	Royco Model 225
Formaldehyde	CEA Total Gas Monitor Model TGM-555
Humidity	Polyimide relative humidity sensors
Temperature	Thermistors
Energy Consumption	Signal from gas valve and from watt transducer