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AHRI Air-Conditioning, Heating, and Refrigeration Institute

DOCKET
10-BSTD-1
DATE MAY 17 2011
RECD. MAY 17 2011

May 17, 2011

Mr. Bruce Wilcox
bwilcox@lmi.net

Re: AHRI Comments on April 12, 2011 Residential Zoned AC Presentation – Stakeholder Meeting #2

Dear Mr. Wilcox:

The Air-Conditioning, Heating and Refrigeration Institute (AHRI) is the trade association representing manufacturers of heating, cooling, water heating, and commercial refrigeration equipment. Over 300 members strong, AHRI is an internationally recognized advocate for the industry, and develops standards for and certifies the performance of many of the products manufactured by our members. In North America, the annual output of the HVACR industry is worth more than $20 billion. In the United States alone, our members employ approximately 130,000 people, and support some 800,000 dealers, contractors, and technicians.

AHRI is writing regarding your study on Residential Zonal A/C. One of our sections, the Zone Controls Technology, has great concern over your study's findings as it relates to the performance of zoning and by-pass. While we do not dispute your actual findings, we are concerned with the 7-Zoned Split Systems that were used in this study. The data you collected does not coincide with the previous studies that show overall zoning savings, as much as 25%.

AHRI and specifically the Zone Controls System Technology Section would like to work with you and other stakeholders to show that zoning can in fact provide substantial energy savings. We are greatly concerned over the removal of zoning credit from the Title 24 Code. Zoning can and will save energy when installed and operated properly. Just like you have a light switch for every room, zoning provides a thermostat for each room, or more practically each zone. We think you would agree that having one light switch for the whole house would not save any energy, just like having only one thermostat wasting heating and cooling in areas not used, or are already at set point, will waste energy as well. With this common sense energy principal in mind now we need to insure that the performance of the HVAC Equipment is not affected.

The AHRI Zoning Members stress the importance of maintaining adequate airflow (CFM) through the A/C Unit at all times. We have reviewed your April 12, 2011 presentation on Zonal A/C and we feel that the zoning systems tested are not representative of proper zoning practice. This shows the need for guidance on zoning which will be published in the upcoming ACCA Manual Z_R. AHRI members have discussed the pictures in the presentation and are concerned about the installation and size of the by-pass, as well as the rest of the duct system. The data indicated, even for some non-zoned systems, that there was inadequate airflow on a majority of the systems. When you include zoning on a poorly designed duct system, the poor performance is multiplied. All AHRI members have witnessed poor installations and those referenced in the presentation are not ones to base the validity of zoning savings.

With respect to slide 4 of the presentation, we have the following comments on single system with dampered supply ducts:

- Return ducts are not zoned – ***This is correct and there is no appreciable value to do this.***
- Single speed compressors and fans can't modulate to track load – ***Even as two speed compressors and variable speed fans can modulate to a degree, or track the load, as in any building the load can move from zone to zone. Zoning directs the capacity to the zone with the load, often matching the load with a low speed capacity.***
- Supply air flow is low, particularly with one zone calling - ***This is not how zoning is supposed to work. Supply airflow must be maintained in order to satisfy the load. Any testing with this fact distorts the ability of zoning. Typically with zoning we try to increase the supply air to any single zone calling, not decrease it.***
- Bypass ducts (short circuit from supply into return) are common - ***While true, proper bypass sizing/installation/setup is extremely important for zone control system performance.***
- Results – low EER – ***Slide 7 only shows 2 zoning systems that went well below the others in lowering the EER. Three of the 5 were only marginally below the EER performance line and in practical terms this drop in performance is offset with overall energy savings.***

Slide 6 states -

- AC systems have low capacity and efficiency
- Cooling air flow lower than the standard in 60% of systems
- Cooling duct pressures are very high
- Cooling Fan Watts are high

If 60% of the systems exhibit cooling airflow lower than the standard, and homes with zoning are subject to similar issues, removing the zoning will obviously improve the performance as shown on Slides 18 and 19. Zoning added to a poorly designed system will decrease system performance. We believe that the installations discussed in your study are poor test cases, and that California Title 24 should address installation practices rather than zoned systems.

Slide 7 states there are 7 zoned systems whereas slide 8 shows that only 5 zoned systems have low EER. Why does slide 8 not account for the two additional zoned systems mentioned in slide 7? We would like to point out that 3 of the 5 zoned systems are just barely below the EER line. There are 11 non-zoned systems that have a lower EER than the lowest zoning system. The data clearly indicates that zoning is not the cause in the other 11 systems. The zoning industry, for 50 years, has had to overcome the misinformation that zoning systems cause such problems. The facts are that these systems are not designed and/or installed properly and as the other 11 non-zoned systems prove, design and installation have a lot to do with performance and it is not just zoning.

Slides 9 and 10 bring the overall basis for the conclusions of these tests into question. These slides show an increase in fan wattage when all zones are calling. This is totally counter to what should actually occur. If all zones are open, the fan wattage should decrease as there should be ample airflow and no restrictions to cause the wattage to increase. This leads us to believe there is system design issues not associated with a proper zone system.

Slide 22 states an acceptable method of zoning are a separate system for each zone. While this has been an accepted practice for quite some time, the offset to this is not a reduction in energy costs for your stakeholders. Each unit requires an electrical disconnect which the utility must account for on its grid. Having a sub-division with 100 homes and two A/C units per house versus one may require utilities to provide more electricity, which is not what they are looking to do. We believe they would rather accept a slight decrease in A/C unit efficiency if it means reducing their load requirement.

Any revision to the current code must look at the current and future state of A/Cs. The new energy codes are requiring higher efficiency equipment and most all of this newer equipment will have variable speed blowers and, in most cases, two speed compressors. The combination of zoning with variable speed blowers and two speed air conditioners will overcome any drawbacks outlined in your study. When the new code actually takes effect, it should reflect what can be accomplished using high efficiency equipment and properly designed duct systems. The code should not be based on the average system, which shows more homes without zoning being just as poor, if not poorer performers. In these older homes, a review of the overall system (including ducts) will have to be conducted to ensure that a high efficiency HVAC is delivered by a high efficiency zoned duct system. We are sure you will agree that there are many poorly designed and installed duct systems in California.

Attached, please find two separate studies showing that zoning will save substantial energy when designed and properly installed. The HVAC Industry is leading the way in developing systems for energy efficient homes. The HVAC equipment is almost at its peak in efficiency. The next step is to look at the distribution system, its design and installation. This is where zoning has its biggest potential for the utilities.

The Air Conditioning Contractors of America (ACCA) has produced an ANSI standard Quality Installation Manual. ACCA is also producing a Zoning Manual, now in its final stages. These two manuals should be referenced in the next update of Title 24. AHRI and its Zone Controls System Technology Section are ready to assist you, the California Energy Commission (CEC) and any others in establishing an effective HVAC Energy Efficiency Code. Zoning must be a crucial part as to reduce the carbon footprint from multiple systems and the energy they require. There is tremendous potential with regards to energy savings if every home or office with multiple systems could be reduced to one HVAC unit and zoning. We estimate that the increase of capacity to consumers would be as much as 20% from the residential market.

We look forward to working with you and hope the data attached provides the bigger picture of the substantial energy savings zoning does provide. We would be happy to clarify any questions you may have with regards to zoning and share the energy savings opportunities that exist. If you have any questions or wish to discuss this further, please do not hesitate to call me at (703) 600-0383.

Sincerely,

Aniruddh Roy
Regulatory Engineer
Air-Conditioning, Heating, and Refrigeration Institute
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ENERGY IMPLICATIONS OF BLOWER OVERRUN STRATEGIES FOR A ZONED RESIDENTIAL FORCED-AIR SYSTEM

P. Oppenheim, Ph.D., P.E.
Member ASHRAE

ABSTRACT

A zoned, forced-air distribution system was designed using industry-accepted methods and installed in an unoccupied research house. A variable-air-volume cooling system was used, and it included a two-speed compressor, a variable-speed blower, dampers, zone thermostats, and prototype hardware for zone temperature and humidity control. Instrumentation was designed and installed to evaluate the delivered comfort and energy performance of the system. A personal-computer-based data acquisition system was used to record data. The zoned system was modified by deactivating the zoning components to represent a conventional unzoned system as a baseline for comparison. A comprehensive system to characterize the thermal performance and the delivered comfort conditions of the distribution system was developed.

The blower on a residential forced-air system typically cycles off when the condensing unit shuts down. The purpose of blower overrun is to take advantage of the cold evaporator coil while not adversely affecting space conditions by re-entrainment of moisture off the coil and moisture in the condensate pan into the airstream.

Using conventional operation (central thermostat, no zoning or thermostat control strategies, and no blower modulation) as a baseline for energy consumption, three other options were investigated. The comfort setpoint was 75°F (24°C), and the setup setpoint for each zone was 85°F (30°C). The energy consumption for zoning with blower modulation and overrun with no thermostat control strategy was 120% of the baseline. The energy consumption for zoning with thermostat control strategies and blower control strategies with overrun was 75% of the energy consumption of the baseline. The energy consumption for zoning with thermostat control strategies (with blower modulation but no overrun) was 84% of the energy consumption of the baseline test. The effect of blower control on humidity levels was evaluated because of the possibility of re-evaporating moisture of the cooling coil. This effect was not seen in the data collected, but a very strong correlation between ambient absolute humidity and moisture removed from the indoor air was observed.

INTRODUCTION

There is a high probability that modulating equipment will become extremely important in residential space conditioning in coming years. Legislation mandating minimum efficiency levels for climate-control equipment is making it increasingly difficult to achieve the required efficiencies while maintaining comfort conditions with single-speed, constant-volume equipment.

For example, a potential problem exists in controlling latent loads with high seasonal energy efficiency ratio (SEER) cooling equipment that uses a "warm" evaporator coil. Raising the temperature of the evaporator coil increases the suction pressure of the system. A higher suction pressure increases the density of the refrigerant and can also reduce the compression ratio. Both of these effects result in a higher equipment operating efficiency. However, this condition elevates the dew point of the coil and can subsequently decrease the dehumidification ability of the unit. A solution to this problem is the development of variable-volume-delivery residential equipment. A central forced-air unit with a variable-speed indoor blower coupled to a variable-speed compressor could adjust to varying loads and would be able to respond to both sensible and latent load efficiently.

The technology for variable-speed indoor blowers and for two-speed and variable-speed compressors is available and is currently in use by several manufacturers in their product lines. Assuming that variable-speed indoor blowers become the standard of the future for cooling, there are many potential advantages for the heating plant as well. Indeed, the need for modulating central units for latent control in cooling may propel the use of modulating units for heating, especially in conjunction with zoned systems.

Modulating airflow over the indoor cooling coil requires control of the refrigerant flow rate. By effectively controlling both airflow over the evaporator coil and the refrigerant flow, an air conditioner can operate efficiently over a wide range. The advantages of a modulating air-conditioning system can be summarized as follows:

- Oversizing is virtually eliminated because the unit modulates to respond to the load when two-speed or variable-speed compressors are used.
- Run time increases during mild ambient conditions, thereby decreasing room air stratification and room-to-room temperature variation.
- The ability to zone a house for both comfort enhancement and energy reduction is dependent on having a modulating unit. A constant-volume system with a "dump zone" is not an energy-efficient alternative. This strategy involves delivering air to a normally unconditioned space (dump zone) to allow a constant-volume system to continue to operate at a normal system static pressure when a damper to a conditioned area closes.
- Ventilation strategies for indoor air quality are

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possible when used in combination with a central delivery system with variable-air-volume delivery capability.

The objectives of this work were to quantify the fuel cost savings provided by a zoned, forced-air distribution system compared to a conventional unzoned system and the effects of blower overrun strategies. The basic premise supporting this investigation is that a zoned, forced-air system offers better control of comfort conditions at lower energy costs than a conventional, unzoned house.

LABORATORY FACILITY

The laboratory house used in this study was completed in the fall of 1987. The house was designed and constructed by a national building research group. The house was built in Prince Georges County, Maryland, approximately 10 miles (16 km) east of Washington, DC. Data from a national builder practices survey were used to develop specifications for the design of the laboratory house. The objective was to incorporate trends so that the research house is representative of homes that will be built in the 1990s.

The house is one and one-half stories with a total living area of 2,225 ft^2 (207 m^2). It has a full basement with cast concrete foundation walls. Open-web floor trusses were used for the first floor, and plywood joists were used for the second-floor framing. The roof was built with prefabricated scissor trusses to provide a cathedral ceiling over the living area. Exterior walls were framed with 2 in. by 4 in. (5 cm by 10 cm) wood studs on 16-in. (41-cm) centers. R-13 friction-fit mineral fiber insulation with plastic foam sheathing was used in the exterior walls. The ceiling was insulated with R-30 glass-fiber batts. Vinyl siding was used on the side and back walls, and the front wall was faced with a brick veneer.

The house was divided into three zones for cooling. Zone 1 was the second-floor bedrooms, Zone 2 was first-floor bedrooms, and Zone 3 was the first-floor living area. The basement was not conditioned for these tests. A description of the components used in the laboratory house is given in Table 1.

TABLE 1
Laboratory House Characteristics

Location	Bowie, Maryland
Constructed	1987
Style	One and one-half story detached with full basement Four bedrooms, two and one-half baths Two-car attached garage (used as data acquisition area)
Floor Area	1,600 ft^2 (149 m^2) first floor 625 ft^2 (58 m^2) second floor 1,550 ft^2 (144 m^2) basement
Construction	Exterior finish -- brick veneer front with balance in vinyl siding Poured concrete basement walls with 2 in. by 4 in. (5 cm by 10 cm) furring to accommodate R-11 batt insulation Open web floor trusses for first floor Plywood floor trusses for second floor Exterior walls 2 in. by 4 in. (5 cm by 10 cm) studs on 16-in. (41-cm) centers insulated with R-13 friction-fit insulation with plastic foam exterior sheathing Roof insulated with R-30 fiberglass batt insulation Low-emission insulated glass used for all window and door glazing
Space Conditioning	Modulating prototype furnace 73,500 Btuh (77,543 kJ) input, 82% efficiency Two-speed condensing unit Electrically commutated direct current indoor blower motor Round butterfly dampers

EXPERIMENTAL METHODOLOGY

The objective of this work was to quantify the fuel savings and the moisture-removal capability of a variable-air-volume delivery system. The basic premise supporting this investigation is that a zoned, forced-air system offers better control of comfort conditions at lower energy consumption than a conventional, unzoned house. A test plan, measurement parameters, and a data analysis procedure were developed to test this premise. The tests that were conducted are shown in Table 2.

TABLE 2
Description of Tests Conducted

Test #	Description	Thermostat Schedule
1	Characterize energy consumption in the house using a conventional two-speed condensing unit (no zoning, no indoor blower modulation, no humidity control).	75°F all day
2	Cooling test using a two-speed condensing unit with indoor blower modulation to accomplish both zoning and humidity control with blower overrun (physical isolation between zones).	75°F all day
3	Cooling test using a two-speed condensing unit with indoor blower modulation to accomplish both zoning and humidity control with blower overrun (physical isolation between zones).	Schedule according to Table 3
4	Cooling test using a two-speed condensing unit with indoor blower modulation to accomplish zoning and humidity control with no blower overrun (physical isolation between zones).	Schedule according to Table 3

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TABLE 3
Thermostat Schedule for Tests 3 and 4

Zone #	Description	Time	Thermostat Setting
1	2nd floor bedroom area	11 p.m. - 8 a.m. 8 a.m. - 11 p.m.	75°F all week 85°F all week
2	1st floor bedroom area	11 p.m. - 8 a.m. 8 a.m. - 11 p.m.	85°F all week 75°F all week
3	1st floor living area	11 p.m. - 8 a.m. 8 a.m. - 11 p.m.	85°F all week 75°F all week

Humidity Control with Blower Overrun

The procedure for humidity control with blower control strategies is described below:

1. A call for cooling at the central zone controller opens the appropriate dampers, sets the blower speed according to zone requirements, and turns on the condensing unit.

2. First-stage dehumidification (humidity above 55% RH)—drops the normal volumetric flow by 200 cfm (6 m^3/min). This slows air movement over the evaporator coil and allows for better dehumidification.

3. Second stage dehumidification (humidity above 65% RH)—increases airflow by 400 cfm (11 m^3/min). This is a net gain of 200 cfm (6 m^3/min) over normal requirements. This additional air goes through a bypass loop from the supply plenum to the return. The bypass allows the air another pass over the evaporator coil, thereby reducing its humidity. The reason for the increase in airflow is to maintain system static pressure, thereby maintaining airflow to the zones as required.

Either Step 4, 5, or 6 will happen, depending upon the humidity level in the house.

4. When the thermostat is satisfied, the blower will shut down immediately if second-stage dehumidification is in effect. This is done because any air passed over the evaporator coil once the condensing unit has shut off will evaporate water on the coil and aggravate an already high humidity condition.

5. When the thermostat is satisfied, the blower will run for two minutes at a reduced flow rate of 200 cfm (6 m^3/min) if first-stage dehumidification is in effect. This is done because the evaporator still has the ability to do cooling while not adding significantly to the latent load.

6. When the thermostat is satisfied, the blower will run for four minutes at a normal flow rate. This period has been determined as the optimum run time after condensing unit shutdown to recover work that is available in the evaporator.

Humidity Control Without Blower Overrun (Test 4)

Humidity control by varying the blower speed with no fan overrun is done because of manufacturers' concerns over the reintroduction of moisture into the air after the condensing unit shuts off. Steps 1 through 3 from above apply to this test. The blower will stop at the same time as the condensing unit for this test.

Measurement Parameters

Performance of a climate-control system is measured by the energy efficiency of the system and the degree of indoor comfort provided, including the dynamic response of the system to changing outdoor conditions and different indoor conditions. Testing protocols were designed to provide data to evaluate the performance of different cooling systems with scheduled indoor settings over the range of outdoor conditions in the Washington, DC, area. A variety of parameters defining outdoor weather conditions, system response, indoor comfort, and energy consumption were monitored.

Air temperature was measured at a height of 43 in. (109 cm) from the floor at the geometric center of each room of the house. Other parameters related to comfort, including mean radiant temperature, relative humidity, and room air velocity, were also measured at a 43-in. (109-cm) height at the geometric center of one designated room in each conditioned zone. These four comfort parameters provided the basis for calculating comfort indices with appropriate values for clothing insulation and metabolic rate. Additionally, air temperature was measured at a 4-in. (10-cm) height from the floor and 4 in. (10 cm) below the ceiling.

Measurement parameters used in this study are summarized in Table 4. Indoor/outdoor parameters are conditions that influence interaction of the building envelope with outdoor or unconditioned spaces. HVAC parameters are measurements that describe the operational conditions of the space-conditioning systems. Status parameters are the on/off status of appliances. Outdoor and indoor parameters were scanned by the data acquisition system every 60 seconds and averaged on the hour. Data observations from the HVAC system were conditional on furnace fan status and supply damper position. If the furnace fan was on and the damper position was open for a particular zone, then that information was recorded on the 60-second scan and averaged for the hour. On/off status parameters of the furnace were taken every 10 seconds and totaled by hour. Energy consumption registered by electric meters was also tallied by hour.

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TABLE 4
Measurement Parameters

1. Outdoor Measurement Parameters
   - Wind speed
   - Wind direction
   - Solar radiation
   - Relative humidity
   - Air temperature
   - Barometric pressure
   - Precipitation
   - Ground temperatures
2. Indoor/Outdoor Measurement Parameters
   - Air infiltration
   - Interzonal air flows
   - Air temperature of unconditioned areas
3. Indoor Measurement Parameters
   - Air temperature at thermostat
   - Wall temperature at thermostat
   - Stratification in room
   - Relative humidity
   - Mean radiant temperature
4. HVAC Measurement Parameters
   - (Main) Supply and Return
     - Static pressure differential between supply and return
     - Temperature
     - Humidity
     - Velocity
   - Supply Registers -- Temperature
5. Electric Monitoring
   - House total
   - Forced-air blower for furnace
   - Laboratory
   - Outdoor lights
   - Zone controller
6. Specifications for Status Parameters
   - Furnace fan
   - Water heater
   - Dampers

RESULTS

A subset of the 1988 test year was used to develop the characteristic fuel consumption lines for each of the tests conducted. Data points were collected over the range of ambient summer conditions so that the predicted line for each test would provide an accurate characterization of the electric consumption of the climate control system.

Test bins were filled on a weekly flip-flop basis, back and forth between the central and zoned delivery systems; however, the schedule was adjusted between central and zoned delivery tests in order to capture run-time hours for each test in each bin. The minimum period for each test was five days in order to minimize "edge" effects that might occur in shorter-term tests. All switches between tests were made at midnight.

The ambient weather conditions for each test are presented in Table 5. The tests conducted are shown by calendar day along with average outdoor air temperature, relative humidity, wind speed, solar insolation, barometric pressure, and rainfall. These ambient parameters were useful in explaining outliers in the characteristic energy consumption lines developed for the condensing unit and the blower for the four tests (Figures 1 through 8).

Historical weather data from Andrews Air Force Base, which is 10 miles (16 km) from the test house site, were used with the characteristic fuel consumption regression lines developed for each of the system configurations in this study to estimate fuel consumption weighted by temperature bins.

The information from Figures 1, 2, 3, and 4 (condensing unit electric consumption by test) and Figures 5, 6, 7, and 8 (blower unit electric consumption by test) was used to derive Table 6, which is a comparison between the unit power consumption for each of the four tests for a historical year. This information is presented graphically in Figure 9.

Zoning with a no-thermostat setup (Test 2) used more electricity for cooling than the system in a central configuration (Test 1) with no thermostat setpoint scheduling. The reason is that by having temperature control at three points instead of just one, the air-conditioning unit was more responsive to the house load. Since thermostat scheduling was not used in Tests 1 and 2, it is reasonable that the zoning system would use more electricity while maintaining more comfortable indoor conditions.

Test 3 was the most aggressive energy-conserving strategy. In addition to using the thermostat strategy as specified in the test plan, this zoning strategy used fan overrun. Thus, when the condensing unit cycled off, the indoor blower continued to run based on indoor humidity levels as specified in the test plan. Thus, air-conditioning unit power consumption for Test 3 was only 75% of that for Test 1. Test 4 had the same thermostat control strategy as Test 3 but did not have the blower overrun algorithm. The air-conditioning unit power consumption was 84% of the power consumption used in Test 1. Thus, optimum control of comfort conditions in different zones with no regard to occupancy schedules comes at an energy penalty of 120% of centrally sensed demands. Consideration of occupancy schedules and indoor blower operating schedules had an air-conditioning unit power consumption that was 75% of the consumption of the central system, and not taking advantage of blower control strategies changed the air-conditioning unit's power consumption to 84% of the power used by the condensing unit in the central mode.

The total power used for cooling was less for Test 3 (blower overrun) than for Test 4 (no blower overrun). All other parameters were held constant for this comparison. Low R^2 values for condensing unit power consumption (0.77 for Test 3 and 0.73 for Test 4), coupled with the low R^2 values for blower power consumption (0.59 for Test 3 and 0.46 for Test 4) make the margin of error greater than the numerical difference seen between the tests. Both tests were successful in maintaining indoor relative humidity levels according to the test plan.

The high R^2 values in Figures 1 and 2 indicate that outside air temperature is a very strong predictor of air-conditioning power consumption. However, since Test 3 and 4 use setback strategies, a daily ambient average temperature is not as good a predictor of power consumption as indicated by the low R^2 (Figures 3 and 4).

SUMMARY AND CONCLUSIONS

A zoned, forced-air system was designed using industry-accepted methods, and was installed in an unoccupied research house. Instrumentation was installed to allow evaluation of the delivered comfort and energy performance of the system. A personal-computer-based data acquisition system was set up to record data points. The zoned-air delivery system was modified by deactivating the zoning components to represent a conventional unzoned system as a baseline for comparison.

1. Using conventional operation (central thermostat, no zoning or thermostat control strategies) as a baseline for energy consumption, three other tests were conducted. Using the energy consumption for

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TABLE 5
Daily Ambient Weather Conditions for the Four Tests

Calendar Date	Test #	Ave Outdoor Air Temp °F	Ambient RH %	Wind Speed mph	Solar Insolation Btu/hr ft² day	Barometric Pressure	Rain Inches
6/09	2	56	87	3.4	1111	29.7	0.3340
6/10	2	58	63	2.5	3069	30.0	0.0000
6/11	2	63	56	2.7	3037	30.0	0.0000
6/12	2	72	51	4.5	2934	30.0	0.0000
6/13	2	78	54	3.0	2863	30.1	0.0000
6/14	2	79	59	2.7	2765	30.2	0.0000
6/15	2	80	59	3.3	2812	30.1	0.0000
6/16	1	79	68	4.4	2349	29.9	0.0000
6/17	1	74	78	2.3	1753	29.9	0.1500
6/18	1	77	67	3.2	2684	30.0	0.0000
6/19	1	74	81	3.5	1815	30.1	0.0000
6/20	1	80	66	4.2	2561	30.0	0.0000
6/21	1	85	63	4.2	2750	29.9	0.0000
6/22	1	86	63	4.5	2479	29.8	0.0000
6/23	3	84	61	4.9	2269	29.8	0.0000
6/24	3	72	51	4.9	2550	30.1	0.0000
6/25	3	67	79	4.4	109	29.9	0.0000
6/26	3	77	63	4.9	1741	29.7	0.0000
6/27	3	70	53	3.5	2922	29.8	0.0000
6/28	3	72	52	3.9	2913	29.9	0.0000
6/29	3	75	51	3.1	2489	29.8	0.0000
6/30	3	67	45	3.6	2830	29.7	0.0000
7/01	4	63	58	4.0	2632	29.8	0.2330
7/02	4	68	60	3.4	3011	29.9	0.0000
7/03	4	73	59	4.3	2727	29.9	0.0000
7/04	4	77	61	4.5	2861	30.1	0.0000
7/05	4	77	59	3.9	2869	30.2	0.0000
7/06	4	80	57	3.4	2358	30.2	0.0000
7/07	3	87	51	5.0	2532	30.0	0.0000
7/08	3	81	62	4.2	2359	30.0	0.0000
7/09	3	80	71	3.4	1887	29.9	0.0000
7/11	3	80	74	3.2	227	29.9	0.0170
7/12	3	76	92	2.5	1349	29.9	0.2010
7/13	3	82	63	3.4	2820	29.9	0.2010

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TABLE 5
Daily Ambient Weather Conditions for the Four Tests (continued)

Calendar Date	Test #	Ave Ambient Air Temp °F	Ambient RH %	Wind Speed mph	Solar Insolation Btu/hr ft^2 day	Barometric Pressure	Rain inches
7/14	3	72	86	2.3	143	29.9	0.0000
7/15	2	88	55	3.4	2695	29.9	0.0000
7/16	2	87	68	4.8	2674	30.0	0.0000
7/17	2	86	77	5.4	2463	29.9	0.9830
7/18	2	83	76	3.8	2765	29.9	0.0170
7/19	2	79	86	5.3	1558	29.9	0.2000
7/20	2	77	90	4.8	1806	30.0	0.1670
7/21	2	78	88	7.5	1849	29.9	0.8330
7/22	1	75	93	2.6	1243	30.0	0.0670
7/23	1	74	84	3.7	1323	30.0	0.8140
7/24	1	77	79	3.5	2699	29.9	0.0170
7/25	1	79	72	3.1	2657	30.0	0.0000
7/26	1	78	82	4.6	2282	30.0	0.3090
7/27	1	73	92	2.6	1702	30.0	0.1670
7/28	1	76	88	2.8	1960	30.1	0.0000
7/29	1	83	74	4.1	2624	30.1	0.0000
7/30	4	85	70	3.5	2686	30.0	0.0000
7/31	4	83	74	2.4	2471	29.9	0.0000
8/01	4	82	79	2.5	2337	30.0	0.0000
8/02	4	83	77	3.1	2472	30.1	0.0000
8/03	4	82	78	3.9	2686	30.2	0.0000
8/04	3	82	78	4.3	2794	30.2	0.0000
8/05	3	81	79	5.3	2738	30.1	0.0000
8/06	3	79	82	2.9	2408	29.9	0.3000
8/07	3	81	75	2.8	2614	29.9	0.0000
8/08	3	80	67	2.7	2804	29.9	0.0000
8/09	3	80	75	3.2	2583	30.0	0.0000
8/10	3	83	79	3.0	2636	30.0	0.0000
8/11	3	82	84	4.7	2192	30.1	0.0000
8/12	2	84	78	4.7	2448	30.1	0.0000
8/13	2	85	75	4.9	2379	30.1	0.0000
8/14	2	85	70	6.8	2783	30.0	0.0000
8/15	2	87	70	7.7	2344	29.9	0.0000

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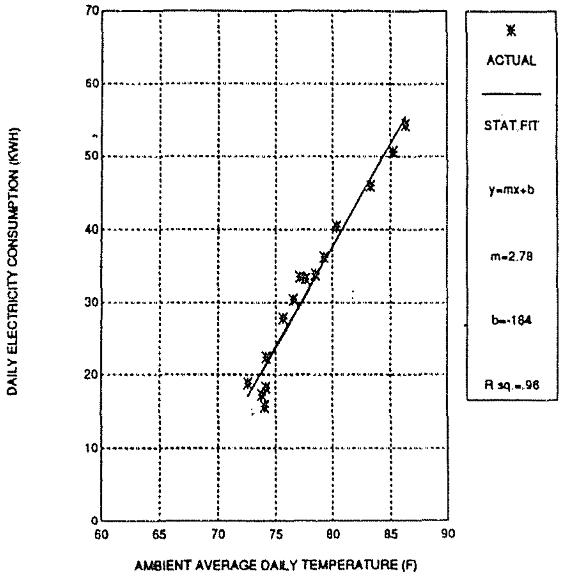

Figure 1 *Daily average condensing unit power consumption for test 1. Test 1 was conventional operation with no zoning, thermostat setup, or blower modulation.*

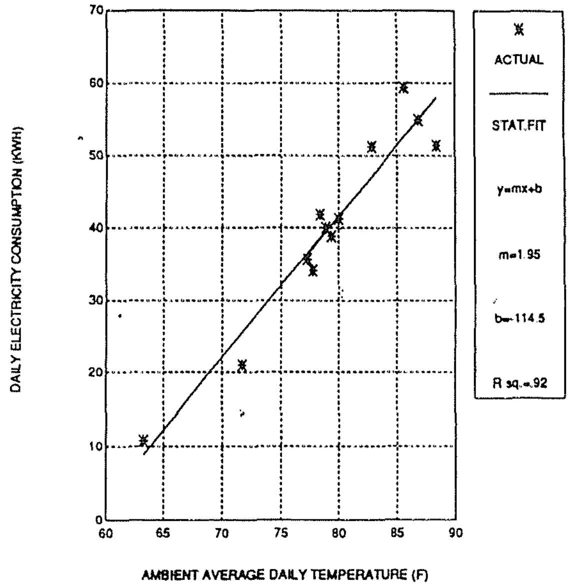

Figure 2 *Daily average condensing unit power consumption for test 2. Test 2 had zoning, blower modulation, and no thermostat setup.*

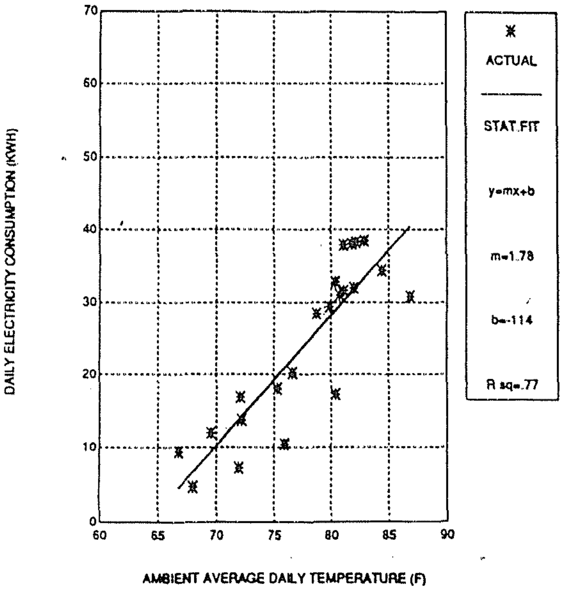

Figure 3 *Daily average condensing unit power consumption for test 3. Test 3 had zoning, blower modulation, and thermostat setup.*

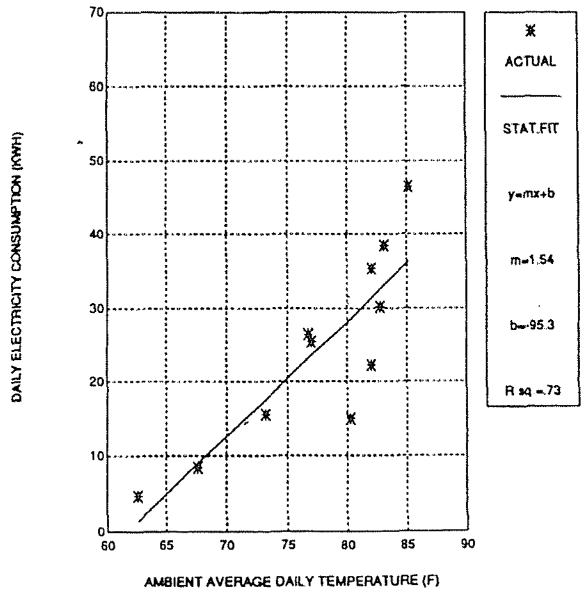

Figure 4 *Daily average condensing unit power consumption for test 4. Test 4 had zoning, blower modulation without overrun, and thermostat setup.*

conventional operation (Test 1) as a baseline, the energy consumption for zoning with no thermostat control strategy (Test 2) was 120% of that for Test 1. The energy consumption for zoning with thermostat control strategies and blower control strategies (Test 3) was 75% of the energy consumption of Test 1. The energy consumption for zoning with thermostat control strategies but no blower strategy was 84% of the energy consumption of Test 1.

2. Tests 3 and 4 were designed to measure how effective certain blower control strategies were at maintaining indoor humidity levels. Even though the blower control strategy of Test 3 incorporated fan overrun—and thus the potential to re-evaporate moisture off the cooling coil—this effect was not seen in the data collected. Less energy was consumed in Test 3 than in Test 4. However, the difference was not statistically significant.

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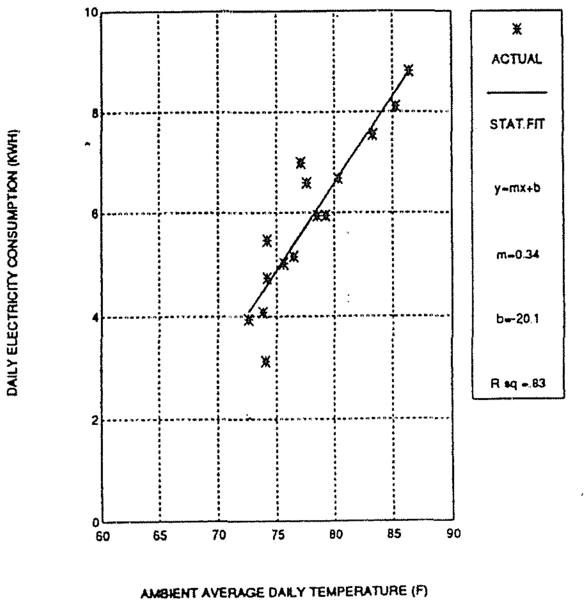

Figure 5 *Daily average blower power consumption for test 1*

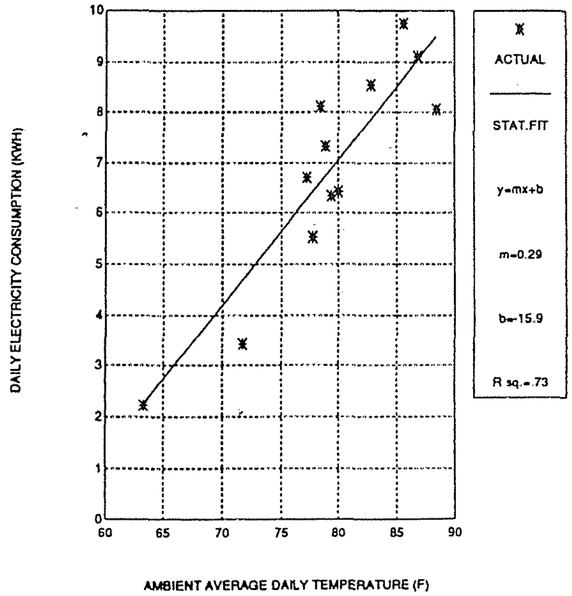

Figure 6 *Daily average blower power consumption for test 2*

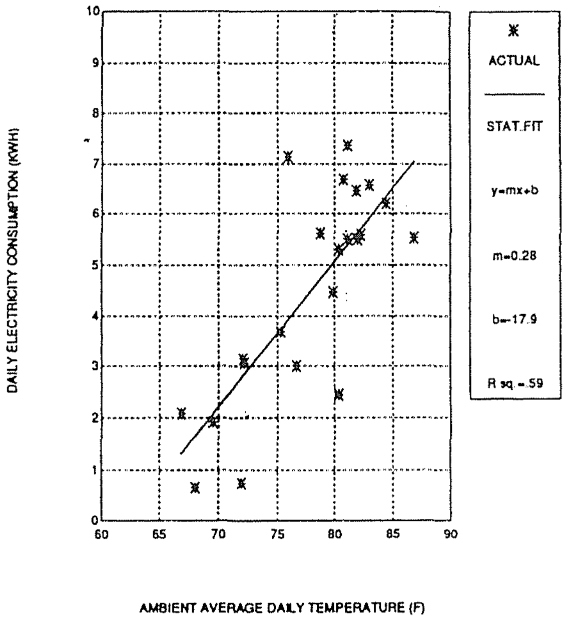

Figure 7 *Daily average blower power consumption for test 3*

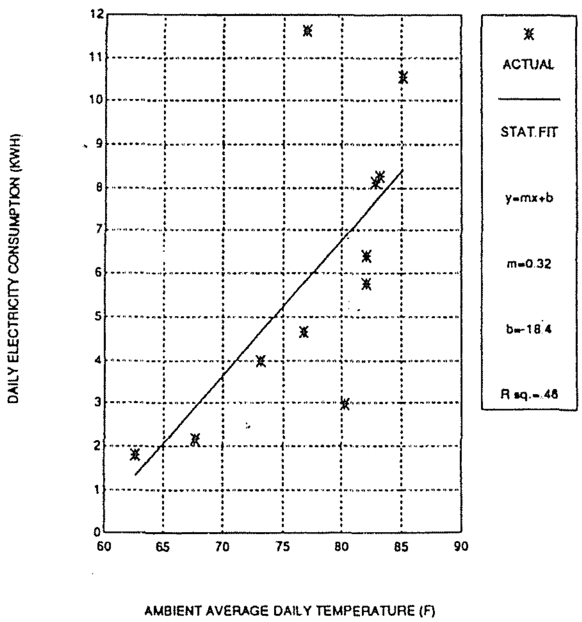

Figure 8 *Daily average blower power consumption for test 4*

3. A strong correlation exists between ambient absolute humidity and moisture removed from the indoor air. This observation is useful since the outdoor absolute humidity/indoor air condensate removal relationship is another assessment of infiltration. In addition, this relationship supplies information that normalizes the effect of climatic conditions on the effectiveness of the evaporator coil in removing moisture from the indoor air.

ACKNOWLEDGMENTS

This project was funded by the Gas Research Institute and the work was conducted at a laboratory facility of the NAHB Research Center. The author was an employee of the NAHB/RC between 1986 and 1989. Special thanks go to Kenneth Kazmer of the Gas Research Institute and Thomas Kenney, Larry Zarker, and Donald Luebs of NAHB National Research Center.

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TABLE 6
Electrical Power Consumption of the Four Tests for a Historical Cooling Season

	Test 1		Test 2		Test 3		Test 4	
	KWH	% of total	KWH	% of total	KWH	% of total	KWH	% of total
Condensing Unit Power Consumption	2010	82.5%	2488	84.6%	1570	86.3%	1605	78.6%
Blower Power Consumption	424	17.5%	454	16.4%	250	13.7%	437	21.4%
Total Power used for Typical Cooling Season	2434		2942		1819		2042	
Power Consumption using Test 1 as the Baseline	1		1.2		0.75		0.84	

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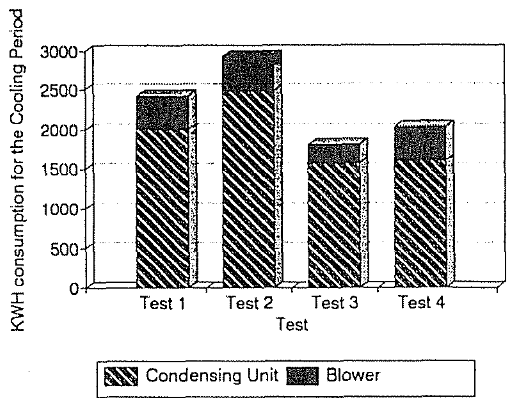

Figure 9 *kWh consumption for cooling season created with historical data and regression lines generated for condensing unit and blower electrical power consumption*

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FINAL REPORT

Field Investigation of Carrier Residential Zoning System

Prepared for

Carrier Corporation
P.O. Box 70 (46206)
7310 West Morris Street
Indianapolis, IN 46231

by

NAHB Research Center, Inc.
400 Prince George's Boulevard
Upper Marlboro, MD 20772-8731

August 31, 1994

RESEARCH SUMMARY

Title	Field Investigation of Carrier Residential Zoning System
Sponsor	Carrier Corporation Agreement No. MKT-13285
Project Manager	Joseph Summa
Contractor	NAHB Research Center, Inc. Project No. 2174
Principal Investigators	Thomas M. Kenney, P.E. C. Edward Barbour

Purpose The purpose of this study was to monitor and evaluate the performance of the residential zoning system in a house that is characteristic of 1990's construction. The primary focus of this field evaluation was:

- operating cost implications of zone control strategies;
- thermal comfort attributable to zoned thermal distribution;
- equipment sizing considering zonal heating and cooling load diversity; and
- electric demand diversity.

Scope The residential zoning system was evaluated for energy consumption and thermal comfort during the summer of 1993 and the winter of 1993/94 in the Home Systems Research House. The evaluation was based on comparing performance data from when the house operated with a zoned distribution system and with a central distribution system. The zone system operated with a 5°F thermostat setup/setback strategy and the central system had a constant thermostat setpoint.

Objective The objective of this study was to demonstrate the advantages of zoned distribution and develop recommended operating practices that Carrier could use to help homeowners operate their zoned heating and cooling systems effectively.

Facility This study was conducted at the Home Systems Research House, a fully instrumented and unoccupied house located at the NAHB Research Home Park in Upper Marlboro, MD. The Research House has dedicated ducting to five zones; including two basement zones, two first floor zones (living areas and bedroom), and one second floor bedrooms. The basement mechanical and laundry areas were passively conditioned. One basement zone in addition to three upstairs zones were comfort conditioned in this study. Each zone was monitored for mean radiant temperature, drybulb temperature and relative humidity to characterize thermal comfort.

Occupancy simulation was provided in each of the zones by adding latent and sensible loads of a three-person family; as well as daily operation of appliances such as dishwasher, range, clothes washer and dryer, and shower. Approximately 150 data points were monitored every minute and selected data summarized into hourly averages.

Technical Approach

The zone and central duct configurations operated on alternating weeks throughout each season. This study design scheme assures similar climatic conditions for each data base. Analytical work includes graphical and statistical methods applied to the monitored data. Data was normalized to permit modelling the house in other climatic regions and companion between the two distribution systems.

Technical Perspective

Forced air heating and cooling is the most popular method of comfort conditioning. This type of system is typically controlled with a central thermostat. Thermostat temperature setup for cooling and setback for heating are effective energy conservation strategies, but implementation is limited to unoccupied periods. Thermal discomfort in remote areas is common because thermal requirements in these areas are not detected by the central thermostat.

Zoned distribution systems can provide improved thermal comfort and encourage energy conservation by conditioning areas only when they are occupied. Multiple temperature sensing used by zone controls provides conditioned air to areas that are inadequately conditioned with a central thermostat.

Zoned systems are known to encourage energy conservation. This has resulted in agencies such as the California Energy Commission to provide performance credits for zoned heating and cooling systems. It is therefore important to introduce this energy code trade-off into other energy codes, such as the Council of American Building Officials (CABO) Model Energy Code. It is equally important that homeowner's operating instructions be provided to achieve the energy effectiveness that zoning offers. Studies have shown that operating cost are strongly influenced by the occupant habits. Unnecessarily high utility bills can result from haphazard thermostat settings with either zoned or central systems. Moreover, zoning can cause higher operating costs if thermostat temperature setup/setback is not used; however, the level of comfort is dramatically increased over the central thermostat.

TABLE OF CONTENTS

EXECUTIVE SUMMARY

A year long study was conducted by the NAHB Research Center, Inc. The study quantified the human comfort and energy savings resulting from operating a properly designed zoning system. The intent of this report is to lay the foundation for manufacturers, distributors, contractors and utility companies to realize the impact of the role of zoning in the future.

Operating cost utilizing zoning with thermostat setup/setback strategies had a 29 percent energy savings over the central system during the cooling season and 27 percent energy savings during the heating season. The thermal comfort attributable to zoning was dramatically improved over the central thermostat system. The test results showed that operating a zoning system without setup/setback strategies could cause higher operating cost compared to a central thermostat system, however, the level of comfort is consistently superior.

The study was conducted in a research house that's typical of the building methods and home characteristics found in the 1990s. An extensive Data Acquisition System was used to monitor and log data from over 150 input sensors. Thermal comfort was quantified for each zone. Two control schemes were tested and measured. The first, a single zone system utilizing one thermostat centrally located and a central duct configuration. The second was a zoned system utilizing four thermostats and a zoned duct configuration.

The cooling test results showed the zone system took advantage of electric demand diversity. The daily average outdoor temperature is a reliable predictor of energy consumption. This is used in determining time-of day and standard electric rates. The intent is to reduce or shift electric loads to off-peak periods. For every degree rise in outdoor temperature, daily average energy consumption increases approximately 2 KWH. Balance point temperature is another factor that influences energy consumption. This is the temperature at which no consumption occurs. The zone thermostat system showed a balance point approximately 3°F warmer than the central system. This is a result of the system dynamics of zoning dampers and thermostat setup/setback settings.

The test results indicated the load estimate overpredicted the cooling load by 24 percent for the zoned system and 16 percent of the central system. However, the results measured a significant difference in the comfort levels indicating the ability of the zoning system to take advantage of cooling load diversity.

1.0 INTRODUCTION

Forced air heating and cooling is the most popular method of comfort conditioning. This type of system is typically controlled with a central thermostat. A central thermostat limits the opportunity for implementing energy conservation through thermostat temperature setup or setback. Also, thermal discomfort in remote areas of the house is common because thermal demands are not detected by a central thermostat.

Zone systems place individual thermostats in various areas of the house. These thermostats help to maintain thermal conditions in remote areas. Zoning allows a house to be separated into distinct conditioned zones based on occupancy patterns and location of rooms. Zoning, combined with thermostat setup/setback strategies, allows homeowners to best satisfy their thermal comfort needs while keeping their system energy consumption costs to a minimum. Occupants' varying schedules can be accommodated by conditioning areas only when the areas are occupied. Thermal comfort in remote areas is enhanced with multiple temperature sensing and the equipment response provided by zone control.

The potential of zoning to encourage and provide energy conservation has resulted in agencies such as the California Energy Commission to allow benefits/credits for zoned heating and cooling systems. It is therefore important to encourage other model energy codes to provide similar requirement benefits for zone systems. It is equally important that clear and concise homeowner's operating instructions be provided to achieve the optimum energy effectiveness that zoning offers. Actual performance and operating costs are strongly influenced by occupants' habits and thermostat schedules. Unnecessarily high utility bills can result from zoning with conventional thermostat setting strategy.

Studies have demonstrated that a multizone system will use more energy than a central thermostat system when a constant setpoint is used. A 35 percent increase was documented (Oppenheim 1991) as a direct result of a multi-zone system being more responsive to the cooling needs of the entire house, since temperature is sensed and responded to at several different locations rather than one. While there is an increase in energy consumption, a zone system does provide more uniform temperatures and better thermal comfort throughout the house than that offered by a central thermostat.

Zonal distribution allows some zones to be placed in *setup* in cooling an elevated thermostat setpoint, or *setback* in winter, a decreased thermostat setpoint, while other zones are maintained at the comfort temperature setting. This energy conservation strategy of cooling and heating only the occupied rooms was studied previously in the Home Systems Research House (Research House) and investigated in this study. Seasonal energy savings attributed to thermostat setup was estimated to range between 2.2 (Oppenheim 1991) and 3.2 percent for every degree setup. These estimates are based on unoccupied setup periods of fifteen hours for bed rooms and nine to fourteen hours for living areas.

1.1 Purpose

The purpose of this study was to monitor and evaluate the performance of the residential zoning system in a house that is characteristic of 1990's construction. The primary focus of this field evaluation was:

- operating cost implications of zone control strategies,
- thermal comfort attributable to zoned thermal distribution,
- equipment sizing considering zonal cooling and heating load diversity; and
- electric demand diversity.

1.2 Test Objective

The objective of this study was to develop documentation to demonstrate the advantages of zoned distribution systems and to recommend practices that Carrier could use to help homeowners operate their zoned heating and cooling systems effectively.

The residential zoning system was evaluated during the summer of 1993 and winter of 1993/94 in the Home Systems Research House. The evaluation was based on comparing performance data from when the house operated with a zoned distribution system and when it operated with a central distribution system. For cooling, the zone system operated with a 5°F thermostat setup strategy and the central system had a constant set point. For heating, the zone system operated with a 5°F thermostat setback strategy and the central system had a constant setpoint.

1.3 Test Implementation

The Research House provides a realistic, occupancy-simulated laboratory test environment for evaluating the Carrier equipment's operating characteristics. Test procedures and schedules followed the protocol for research houses as set forth in the Research House Utilization Plan (RHUP)(Geomet 1988). The house was divided into five thermal zones. Zone 1 was the equipment room located in the basement and was left unconditioned during the test period. Zone 2 was also located in the basement and was conditioned during the test. Zone 3 was the first-floor master bedroom area. Zone 4 was the living room area, and Zone 5 was the upstairs bedroom. A floor plan and duct layout for the house is in Appendix A.

Occupancy simulation was provided in each of the above grade zones by adding latent and sensible loads of a three person family as well as daily operation of appliances such as dishwasher, range, clothes washer and dryer, and shower. Energy consumption was monitored for each major appliance as well as the status of the equipment. In all, approximately 150 data points are monitored every minute, and were combined into hourly averages. Occupant simulation and appliance use schedules are located in Appendix B.

The study design for these tests are based on a comparative analysis. For cooling, the Carrier residential zoning system was operated as central distribution system with a constant thermostat setting located in the living room (Zone 4); and as a zoned distribution system using a setup operating strategy. The test period was divided in half for central thermostat operation with a constant 75°F thermostat setpoint and the zone system used a 5°F thermostat setup strategy during the remaining portion of the season. Division of the season was accomplished by alternating the system between zone and central on a weekly schedule. Thermostat setup schedule employed was a setup of 5°F (80°F setpoint) during unoccupied periods. For heating, the Carrier residential zoning system was operated as a central distribution system with a constant thermostat setting located in the living room and as a zoned distribution system using a setback operating strategy. The test period was divided in half for central thermostat operation with a constant 72°F thermostat setpoint and the zone system used a 5°F thermostat setback operating strategy the other half of the season. Division of the season was accomplished by alternating the system between central and zone on a weekly schedule. Thermostat setback schedule employed was a setback of 5°F (setpoint of 67°F) during "unoccupied periods".

Hourly average and minute-by-minute detail data were taken to characterize and compare the overall performance of each configuration. The primary format used in the database was hourly averages. Minute detail data were captured at selected intervals to evaluate transient conditions such as room temperature recovery from thermostat setup/setback. A detailed listing of the instrumentation is in Appendix C.

1.4 Home Systems Research House Description

Background

The Research House (Figure 1.4.1) was commissioned by the Gas Research Institute (GRI) in the fall of 1987. Carrier Corporation sponsored research in this house during 1993/94. The methodology, measurement parameters were designed in accordance with the Research House Utilization Plan (Geomet, 1991). The purpose of this plan is to ensure that research conducted in all GRI research homes is carried out consistently and uniformly.

Figure 1.4.1
Front View of the Home Systems Research House

The house was designed, constructed, and is owned and operated by the Research Center. The house is located in the NAHB Research Home Park in Prince George's County, Maryland, approximately 25 miles east of Washington, D.C. Data from the Research Center's annual Builder Practices Survey were used to develop specifications for the design of the Research House. The design and construction incorporated trends that are representative of homes to be built throughout the 1990s.

The house is one and one-half stories, with a total living area of 2,225 ft^2 (Table 1.4.1); it has a full basement with poured concrete walls. Open web floor trusses were used for the first floor and plywood trusses were used for the second floor. The roof was built with prefabricated scissors trusses to provide a cathedral ceiling over the living area. Exterior walls were built with 2x4 wood studs on 16-inch centers. R-13 fiberglass batt insulation with extruded polystyrene foam sheathing was used in the exterior walls. The ceiling was insulated with R-30 fiberglass batts. Vinyl siding was used on the side and back walls, and the front wall was faced with a brick veneer. The house is divided into five conditioning zones, one on the second floor, two on the first floor, and two in the basement. The duct system was designed based on the house heating and cooling load in each zone.

Data Acquisition System (DAS)

The DAS consisted of various sensors and signal conditioning boards that scanned approximately 200 input channels and a personal computer for data logging. Data were recorded on the computer's hard-disk drive and later transferred to the Research Center laboratory facilities for processing and evaluation. A list of quality assurance objectives, instrument specifications, data acquisition boards, and sensor wiring codes is provided in Appendix C. Sensor calibration procedures are included in Appendix D.

Comfort monitoring stations equipped to monitor dry bulb temperatures from 4 inches to 12 feet above the floor, mean radiant temperature, humidity, and a sulfur hexaflouride (SF_6) tracer gas sampling tube to measure air infiltration rates were located in each of the five house zones.

Table 1.4.1
GRI Home Systems Research House Characteristics

Location	16001 Pennsbury Drive Mitchellville, Maryland 20716
Constructed	1987
Style	One and one-half story, detached with full basement. Four bedrooms, two and one-half baths. Two-car attached garage (used as data acquisition area).
Floor Area	1,600-square foot first floor 625-square foot second floor 1,550-square foot basement
Construction	Exterior finish -- brick veneer front with balance in vinyl siding. Poured concrete basement walls with 2x4 partitions to accommodate R-11 batt insulation. Open web floor trusses for first floor. Plywood floor trusses for second floor. Exterior walls constructed of 2x4 studs on 16-inch centers insulated with R-13 fiberglass batt insulation with extruded polystyrene foam exterior sheathing. Roof insulated with R-30 fiberglass batt insulation. Low-emissivity, double-pane insulated glass used for all window and door glazing.
Space Conditioning	• Furnace 60,000 Btuh Model 58SXC060 • AFUE 91.5 percent • Residential zoning control system • 3 Ton Single Speed Condenser, Carrier Model 38TKB036301 with 10 SEER • Air Handler Coil, Carrier Model CD5A036 • Thermostatic Expansion Valve, Model TXV • Barometric Bypass Damper

Quality Assurance Program

A quality assurance program was implemented in accordance with the RHUP to assure a reliable database. The program consisted of daily instrument checks for reasonableness and accuracy, comparisons of manual meter readings with on-screen DAS readings, logical DAS operation checks, performance checks, and multipoint calibrations. An audit is performed on the house data acquisition system to verify instrumentation accuracy at the beginning of each test season. The audit procedures are provided in Appendix D.

2.0 1993/94 HEATING RESULTS AND DISCUSSION

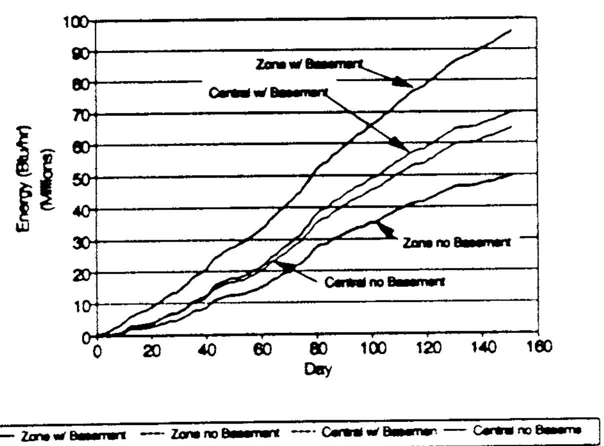
2.1 1993 Heating Season and Record Year Climate

The Washington, D.C., area heating season spans the 30-week period between October 1 and April 27. To compare the test period heating season data with that of a typical heating season, weather data collected for this report were normalized with the Typical Record Year (TRY) weather data, shown in Appendix E. The TRY data was provided by the National Climatic Center, Asheville, North Carolina. TRY data from other locations can be used with these to estimate performance and energy consumption.

2.2 Heating Equipment Operating Characteristics and Gas Consumption

Seasonal energy consumption and operating cost was estimated from measured test data and TRY weather data. Energy consumption depicted in Figure 2.2.1 estimates energy required for heating the laboratory house during the TRY heating season. It includes electrical energy for the air handler and gas energy supplied to the furnace. Four lines plotted on the graph show the difference in energy consumption of the zone and central systems; with and without energy delivered to the basement.

Figure 2.2.1
Energy Consumption



Cost Analysis

The predicted seasonal energy expenditures are shown in Table 2.2.1 along with associated operating cost. Electric power tariff of 7.075 ¢/KWH and natural gas tariff of 0.711 $/Therm were used to determine operating cost. Operating cost differences between the systems were determined using the central system as a reference point.

When the basement was conditioned the zone system operating cost was significantly more than the central system. This higher operating cost is attributable to superior comfort provided by the zone system in the basement area. The basement area required approximately 12,000 Btuh to maintain thermostat setting of the zoned system. The basement did not attain comfort conditions when operating with a central thermostat because a thermostat was not located in the basement.

Table 2.2.1

Control Strategy	Gas		Electric		Total		Cost Difference
	Energy (Therm)	Cost($)	Energy (KWH)	Cost($)	Energy (MMBtu)	Cost($)	%
Zone w/Basement	956	680	594	42	97.6	722	+34
Zone wo/Basement	494	351	591	42	51.4	393	-27
Central	695	494	620	44	71.6	538	Base
Central wo/Basement	646	459	512	36	66.4	495	-8

Heating energy provided to the basement was determined and removed to estimate the impact of the basement on operating costs. With the basement heating removed from the zoned system consumption estimate, the zoned system operating cost was 27 percent lower than the central system (including the basement). For parity, energy provided to the basement by the central system was estimated and subtracted from the central system seasonal consumption estimate. In this case, the operating cost of the zone system was 21 percent less than the central system. This cost savings includes the effects of zoning and 5°F thermostat setback. Aggregated over the season, it represents a savings of 4 percent per degree of setback.

In another study conducted at the Research House (Oppenheim 1991), thermostat schedules and zoned control strategies were evaluated with the basement unconditioned. Zoned distribution was determined to have a 12 percent reduction (one percent reduction/degree setback) over the central system for seasonal energy consumption. Both systems operated with a 12°F setback in the living room area. For the zone system, additional setback of bedrooms was implemented for ten hours per day (9 a.m. to 7 p.m.). Physical isolation between zones (closed doors) may have also contributed to the effectiveness of the zone distribution system. A third zone setback strategy increased the bedroom setback time by eight hours (11 p.m. to 7 a.m.) to a total of eighteen hours of setback per day but this additional setback time did not result in more energy savings.

Disaggregated Heating Demand

Gas consumption for space heating was evaluated using average daily outdoor temperature and average daily gas and electric consumption. Energy consumed by each zone was determined by apportioning the total energy consumed. The allocation was made by applying a mass/energy balance of measured air flow in the zone ducts. The duct system was designed using the static-regain method for each individual run to the five zones. The static regain method is described in detail in the ASHRAE Handbook of Fundamentals, 1993, Chapter 32. Velocities in each branch never exceeded 600 fpm, and the static pressure across the fan ranged from 0.6 to 0.8 in. WC throughout the test.

Figure 2.2.2 shows the sum of linear regressions for the zone system; with and without the basement heating energy. The difference between these two lines is a 12,000 Btuh heating load. An interesting observation is at 60°F outdoor temperature, both the central and zone system without basement have the same balance point. This demonstrates that the basement is ground coupled and therefore requires heating even for outdoor temperatures above the balance point.

Genesis of the previous figure is found in Figures 2.2.3 and 2.2.4. These graphs depict the rate of heating energy consumption according to outdoor temperature. They provide additional insight to the responsiveness of each zone to climatic conditions. The zone graph, for instance, indicates that the living room heating load is the most sensitive to outdoor temperature. There is a corresponding increase in energy consumption with decreasing outdoor air temperature. The

Figure 2.2.2
House Heating Load

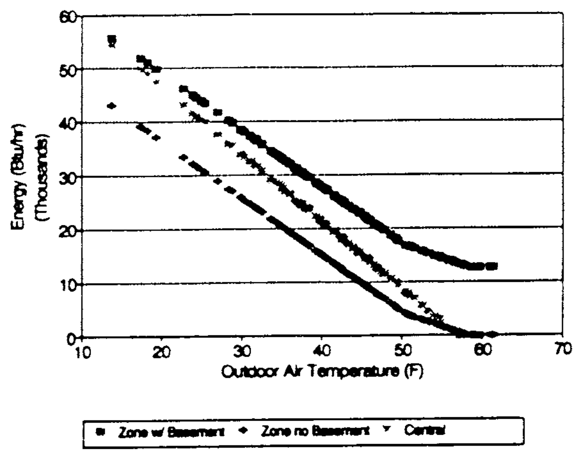


Figure 2.2.3
Zone System
Zone Heating Load

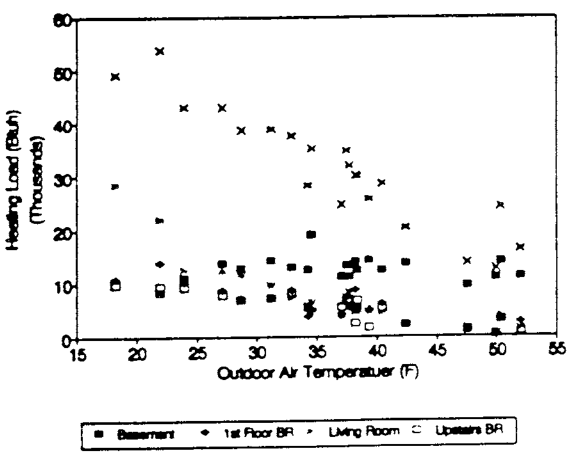
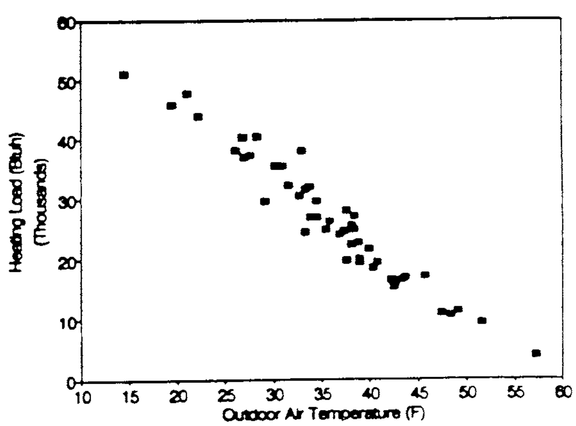


Figure 2.2.4
Central System
Whole House Heating Load



other zones also respond to outdoor temperature, but to a lesser degree. The basement zone heating load, however, does not show any influence from outdoor temperature.

2.3 Frequency of Room Temperature Occurrences

Figures 2.3.1 and 2.3.2 display room air temperature frequencies for occupied periods. Occupied periods are defined as the hours of the day that the thermostat schedule is programmed for the 72°F temperature. Each datum point on the graphs is an hourly average calculated from sixty observations made at one-minute intervals.

Figure 2.3.1
Central System
Room Temperature

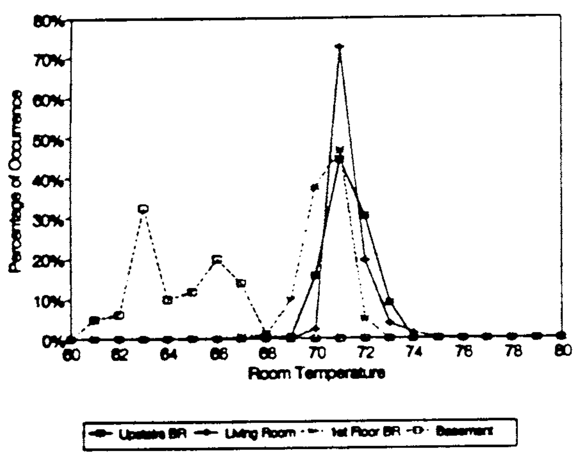
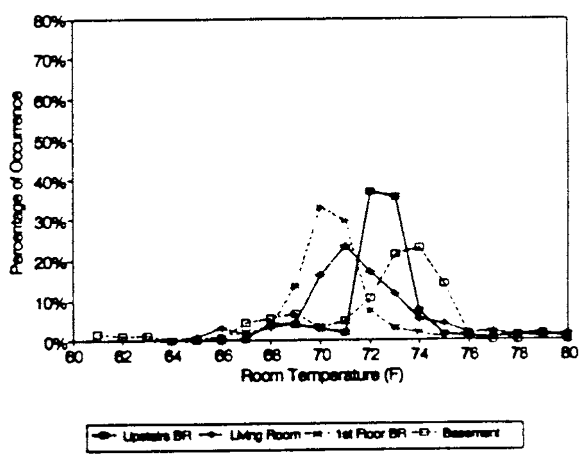


Figure 2.3.2
Zone System
Room Temperature Occurrences



For the central system, the first and second floors had tight control as evidenced by most of the observations occurring near the setpoint temperature. Unlike the upper floor zones, the basement zone very seldom attained setpoint temperature; with most observations within the range of 60° to 68°F. This deficient heating condition will be discussed further in the thermal comfort section of this report.

The zone system demonstrated better control in the basement. For the upper floors, the central system had somewhat better control than the zone system as evidenced by the width of the bell shaped curves. The wider base of the zone curves is a result of the thermostat setback schedule for the zone system. The central system had a constant thermostat setpoint throughout the test period, whereas the zone data included periods where room temperatures were recovering from thermostat setback.

There is evidence of some overheating of the basement zone for the zone system. This overheating situation is attributable to uncontrolled heat loss from ducting located in the basement. It is notable that the central system never attained temperatures above 68°F even with the contribution of this passive heat from the ducts.

Table 2.3.1 lists statistical information regarding the level of temperature control provided to the zones. This information was derived from data presented in Figures 2.3.1 and 2.3.2. The numerical values represent the interaction and responsiveness between the heating load and thermal capacitance of the house, the heating system capacity, and control system. For instance, some overheating may have occurred in the upper floors during daylight hours from solar gain and floor-to-floor stratification. Occurrences of temperatures lower than set point can be in part attributed to recovery periods, furnace capacity and distribution system capacity. Standard deviation listed in the table is a measure of the tightness of temperature control. Mean temperature is the average temperature for the observations and symbolizes the ability of the system to achieve thermostat set point.

Table 2.3.1
Temperature Distributions

	CENTRAL		ZONE	
	MEAN TEMP °F	STD DEV °F	MEAN TEMP °F	STD DEV °F
Basement	70.9	3.5	63.9	1.8
Living Room	69.1	1.8	69.9	0.8
1st Floor Bedroom	70.4	3.1	70.8	0.7
2nd Floor Bedroom	71.0	2.0	70.8	0.8

2.4 Thermal Comfort

The most widely accepted studies on the characterization of thermal comfort have been conducted by Professor P.O. Fanger of Denmark and by Kansas State University for ASHRAE. These studies define indices, predicted mean vote (PMV), and predicted percent dissatisfied (PPD), that characterize thermal comfort in terms of six personal and environmental factors, including metabolic rates, clothing levels, dry bulb temperature, mean radiant temperature, humidity, and room air velocity (Rohles, 1974; Fanger, 1970). A detailed description of these factors is presented in Appendix F, Table F1.

ASHRAE Standard 55-1992 considers conditions environmentally thermally acceptable when 80 percent of a given population in a given area is comfortable. A more detailed description of thermal comfort and a Fortran program for calculating the predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) comfort indices, from ISO 7730, are provided in Appendix F.

Thermal comfort for both tests was characterized using the PMV and PPD indices. PMV depicts the thermal direction of thermal comfort; as indicated on the PMV graph as thermal neutral (zero), warm (positive number) and cool (negative number). PPD index depicts the overall effect relative to the population including cool and warm conditions. The following thermal comfort graphs were developed from measured data during periods of occupancy. Even though the central system was at a constant set point temperature, the same time periods were used to analyze thermal comfort in both zone and central systems to maintain consistency.

Figures 2.4.1 and 2.4.2 depict thermal comfort throughout the heating season for occupied periods. The magnitude and sign of the plots are significant attributes. The graphs contrast the differences between the thermal comfort of the two systems.

For the central system, all of the zones except for the basement was thermally neutral (zero predicted mean vote). The basement as discussed in the previous section never attained set point. This underheated condition is seen on the graph as large negative PMV values.

Figure 2.4.1
PMV - Central System

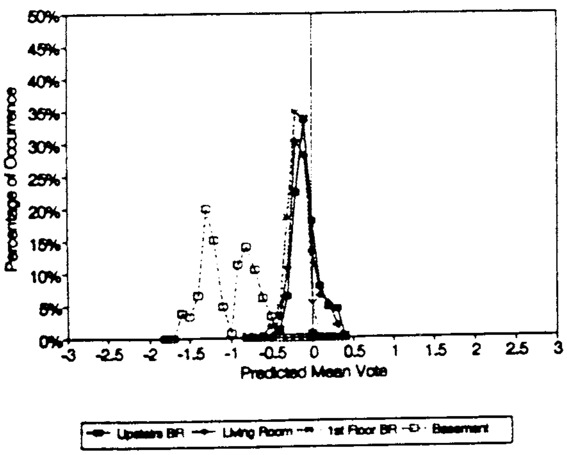
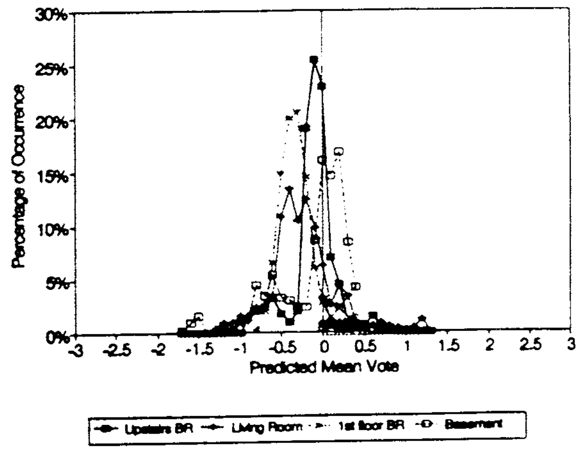


Figure 2.4.2
PMV - Zone System



The zone system has a less orderly display of PMV occurrences as compared to the central system. This characteristic is not an indication of a deficiency. Rather it shows how the zones operate independently. The peaks of curves for the zones are off set from each other and the magnitude is about 10 percent lower than the central system. The resulting larger range of PMV values and shift to the left (cool) for the above grade floor zones are attributable to recovery from thermostat setback. Thermal comfort in the basement zone is much improved over that provided by the central system.

Figures 2.4.3 and 2.4.4 display the percent of people dissatisfied (PPD) for the two systems. PPD accounts for both too cool and too hot conditions in a single number. The target value for acceptable comfort is 20 PPD or less.

As demonstrated earlier, the central system has acceptable control in the above grade zones. The majority values of PPD for the basement zone were 30 and greater indicating thermal discomfort for nearly all occupants.

Figure 2.4.3
PPD - Central System

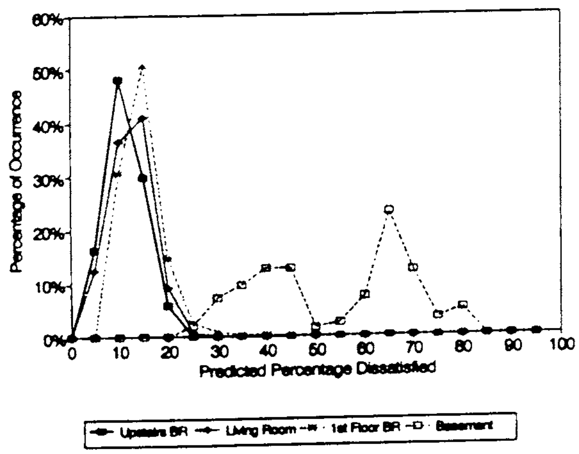
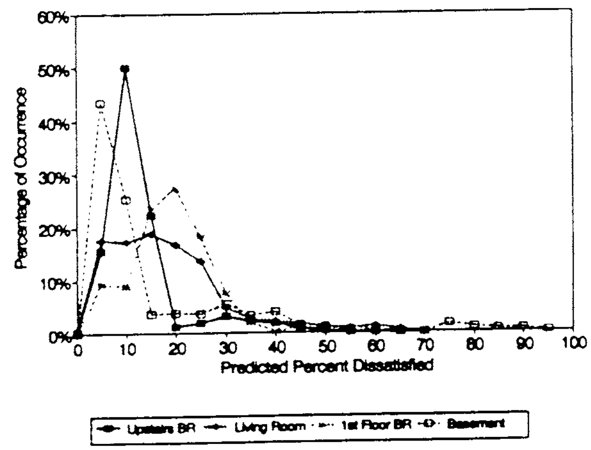


Figure 2.4.4
PPD - Zone System



The zone system had thermal comfort mostly in the acceptable range as shown where PPD is 20 or less. However, there were occurrences on the first floor bedroom and living room where PPD was between 20 and 30. This again is attributable to recovery time and is not to be associated with zoning. A remedy to the recovery time issue would be to start the recovery time earlier to achieve thermostat set point at the desired time. Occupants would normally compensate for this dynamic by readjusting the thermostat schedule.

Combining the findings of the energy consumption and thermal comfort sections of this report provides the following:

- Zoning can improve thermal comfort, especially in areas that are underheated or ground coupled. However, increased operating cost is required to achieve higher levels of thermal comfort.
- Recovery time from thermostat set back should be considered when determining thermostat schedules. Adaptive thermostats relieves this burden from the occupant.
- Set back schedules can significantly reduce operating cost, however some degree of thermal discomfort should be expected.

2.5 Stratification

Air temperature in an enclosed space generally varies from floor to ceiling. If vertical stratification varies more than 5.4°F thermal discomfort can occur according to ISO Standard 7730-1984. To avoid this type of discomfort, the Standard recommends thermal stratification less than 5.4°F, measured at 4-inches and 43-inches from the floor.

Tables 2.5.1 and 2.5.2 display the occurrences of vertical temperature stratification for the occupied periods. The tables show many observations of vertical stratification larger than 5.4°F for the zone system in the basement and living room zones. This stratification is characteristic during recovery from set back and is not necessarily related to zoned systems. Central systems using thermostat set back schedules also exhibit similar stratification. Typically, deeper set backs and higher air delivery temperatures result in more stratification. Stratification occurrences were not extreme and they were short lived. Notice that the all of the zone averages were below 5.4°F. The basement heated with the zone system had the most number of occurrences over 5.4°F. Ducts located in the unfinished ceiling/floor joists area contributed heat whenever the upper floor zones required heat. This caused passive overheating and warmer than desired ceiling temperatures.

Table 2.5.1
Vertical Stratification Between
4-Inch and 43-Inch from Floor
Central System

	Central Distribution 43-4"	Basement	1st Floor Bedroom	Living Room	2nd Floor Bedroom
No. of Observations	5.4°F Differential	464	369	455	418
	over 5.4°F Differential	0	49	9	0
	Min Differential °F	0.2	0.5	0.7	0.1
	Max Differential °F	3.1	7.4	5.9	5.3
	Average Differential °F	1.3	3.7	2.9	1.9
	% Observations over 5.4	0%	12%	2%	0%

Table 2.5.2
Vertical Stratification Between
4-Inch and 43-Inches from Floor
Zone Distribution

	Zone Distribution 43-4"	Basement	1st Floor Bedroom	Living Room	2nd Floor Bedroom
No. of Observations	5.4°F Differential	525	531	596	581
	Over 5.4°F Differential	114	50	43	0
	Min Differential °F	0.1	-1.2	0.6	-3.9
	Max Differential °F	6.9	8.3	8.0	3.5
	Average Differential °F	3.6	2.6	3.3	-1.4
	% Observations over 5.4	18%	9%	7%	0%

2.6 Recovery from Setback

Recovery from thermostat setback is defined as the time required for a system to reheat a house to the point where 80 percent of a random sample of people surveyed would feel comfortable. Stated another way, room temperature must reach approximately 71°F. Factors influencing a heating system's ability to recover from thermostat setback include outdoor temperature, solar radiation, percent heat plant oversizing, house dynamics, and distribution system parameters. The typical acceptable time for recovery is two hours, and time span was monitored using minute by minute detail data to evaluate how long the furnace took to recover. Only in mild temperatures, outside air greater than 51°F, did the zones recover from the five degree setback. In all other cases, the zones did not recover to 71°F in the allotted two hours. This can be attributed to the lack of oversizing which is not recommended in zoning applications.

3.0 1993 COOLING RESULTS AND DISCUSSION

3.1 1993 Cooling Season and Record Year Climate

The Washington, D.C., area cooling season spans a 14 week period between June 6 and September 11. A comparison, shown in Figures 3.1.1 and 3.1.2, between the Typical Record Year (TRY) and the test periods show that the temperature patterns for the test periods were approximately 5 percent cooler than the "typical" year. The TRY data in Appendix E was provided by the National Climatic Center, Asheville, North Carolina, and was used to normalize consumption data to compare the performance between the zone and central systems. TRY data from other locations can be used to estimate the performance and energy consumption. Figure 3.1.3 is a comparison between the climatic conditions that occurred during operation of the zone and central systems. The comparison demonstrates that both systems experienced approximately the same percentage of hours in each temperature bin. Thus, the comparison of the two control schemes in this study is not biased.

Figure 3.1.1
Outside Air Temperature vs. Typical Record Year
Central Test

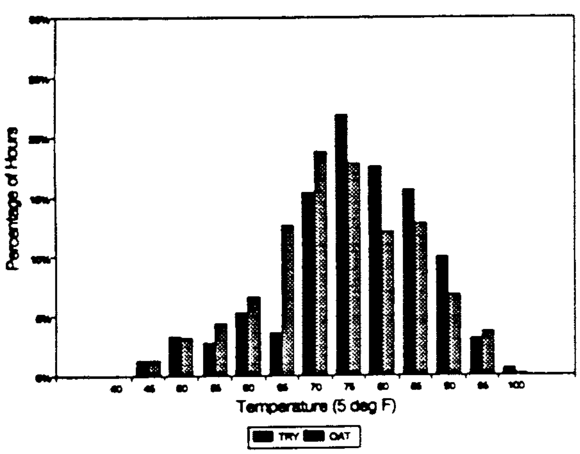


Figure 3.1.2
Outside Air Temperature vs. Typical Record Year
Zone Test

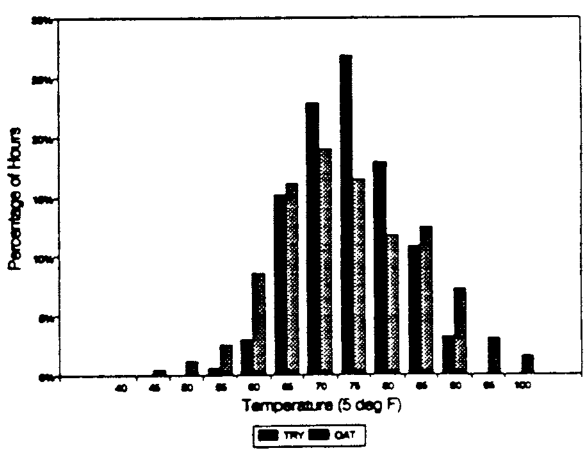
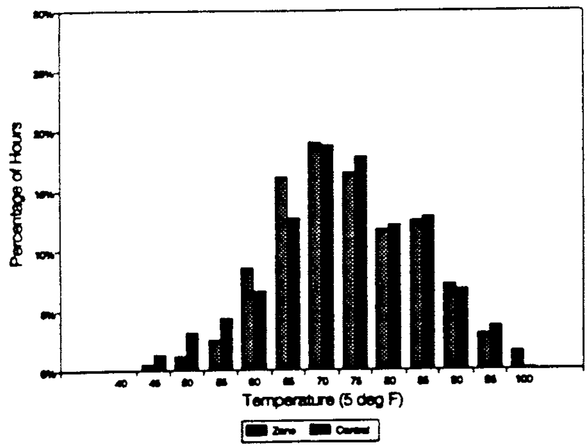


Figure 3.1.3
Outside Air Temperature
Zone vs Central



3.2 Cooling System Operating Performance

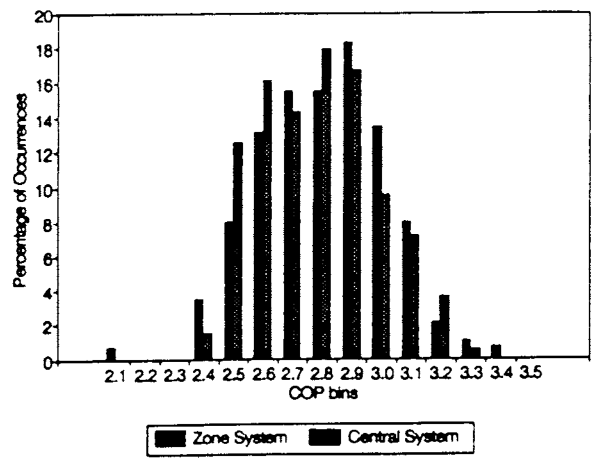
Cooling Equipment Description

The air conditioning unit used in the study was a 10 SEER, single speed, 3 ton condensing unit, Carrier model number 38TKB036301, connected to a Carrier model number CD5A036 air conditioning coil, utilizing the Carrier residential zoning control system. A barometric bypass damper was installed to eliminate possible over pressurization of the supply plenum resulting from closed zone dampers. The air handler operated with constant air circulation.

Performance Observations

Coefficient of Performance (COP), a term analogous to efficiency, is a unitless number and defined as the ratio of the cooling energy produced (in Btu) to the electric energy consumed in Btu. COP is affected by many parameters such as return air conditions, compressor efficiency, and outdoor air temperature. Figure 3.2.1 displays the occurrences of COP for both central and zone tests. The distribution of COP was similar for both systems.

Figure 3.2.1
Coefficient of Performance



Figures 3.2.2 and 3.2.3 show how COP is effected by outdoor air temperature (OAT). The zone system COP displayed a pattern of decreasing COP with increasing outdoor temperature. This represents diminishing cooling capacity with increasing outdoor air temperature. By contrast, central system operated within a band of COP and without a systematic relationship to outdoor air temperature. The zone system operated with a seasonal average COP of 2.8 while the central system operated at an insignificantly lower COP of 2.7. Differences in seasonal energy expenditures discussed later in this report is therefore not attributable to COP variability.

Figure 3.2.2
Central COP vs. Outdoor Air Temperature

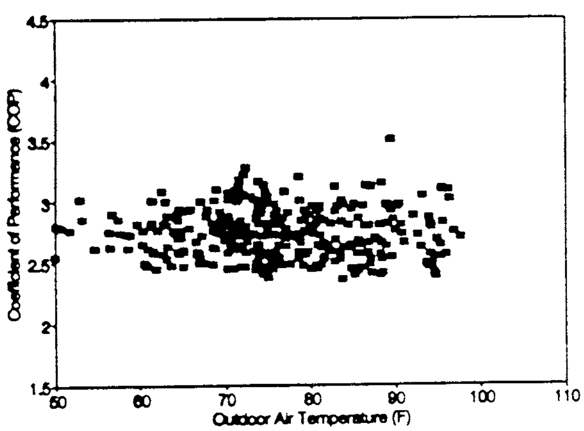
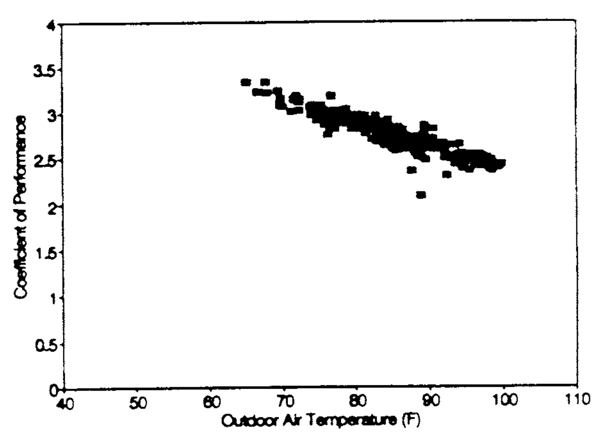


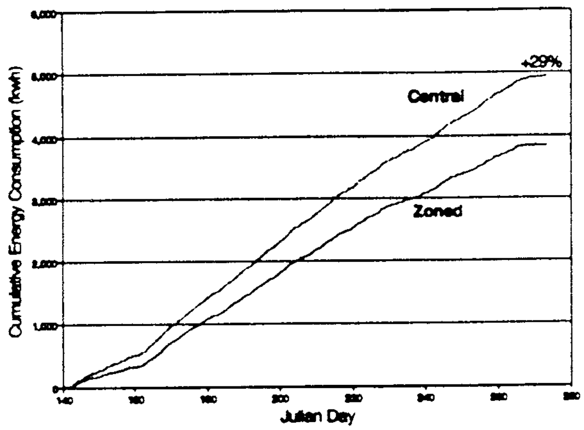
Figure 3.2.3
Zone COP vs. Outside Air Temperature



3.3 Electric Consumption and Demand

Energy consumed by the air conditioning system was analyzed to compare the performance between the zone and central systems. The monitored energy consumption data was normalized with TRY weather data to estimate the seasonal energy expenditure for each system. This estimate is depicted in Figure 3.3.1 where the cumulative energy consumption estimate for the Washington, D.C., area cooling season is 29 percent more energy used by the central system.

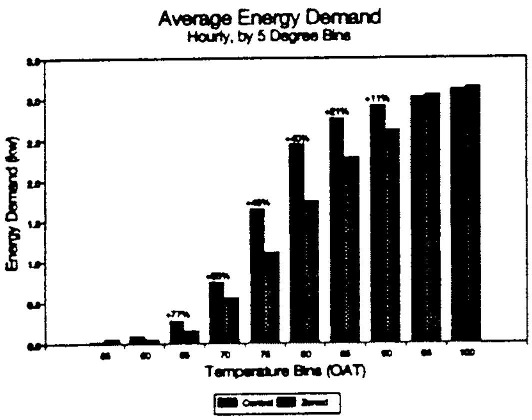
Figure 3.3.1
Cumulative Energy Consumption



The seasonal energy consumption estimate was evaluated for cost with time-of-day and standard-fleet electric rates. Often, electric utilities are able to influence consumer habits with these rates. The intent is to reduce or shift electric loads to off-peak periods. Time-of-day rates provide monetary incentive to homeowners to alter their electric use habits, while builder incentive programs offered by utilities and some energy codes, provide credits or rebates to stimulate the use of peak-shifting technologies.

From an operational perspective, utilities predict system peaks from outdoor temperature forecasts and cycle-off large blocks of loads, e.g., residential air conditioners with radio controlled switches to maintain manageable capacity. Air conditioning controls such as programmable thermostats, zoning, etc., that shift or reduce electric demand during peak periods are inherently important to electric utilities. However, utility control of load shedding is preferred over homeowner control for reasons of reliability. Not withstanding the above, utilities are often motivated to encourage consumer responsiveness to their programs. To that end, zoning provides a level of convenience previously unavailable in houses with conventional equipment. The advantage of zoning for the utility is depicted in Figure 3.3.2. Difference in electric demand between the zone and central systems are shown relative to outdoor air temperature. The graph shows a general trend of lower demand for the zone system. At high outdoor temperatures, electric demand for both systems converge because both are operating at/near 100 percent capacity.

Figure 3.3.2
Average Energy Demand



Energy and Cost Estimate Methodology

Electric consumption was evaluated to discern differences between zone and central systems. Graphical representations of system performance was used to illustrate the differences. The following analysis includes energy used for the air handler motor, condenser fan, compressor and other related parasitic electric loads.

Displayed in Figures 3.3.3 and 3.3.4 is the relationship of outdoor air temperature to energy consumption for the space conditioning. As illustrated by the regression line, the daily average outdoor temperature is a reliable predictor of energy consumption. The correlation coefficient, represented as R-squared on the graph, is a measure of the error between the regression line and the measured data. A perfect correlation, or no error, occurs when all the measured data lie on the linear regression line, is represented by an R-squared of one. R-squared values greater than

0.8 are considered a good fit for estimating with the linear regression coefficients. The regression coefficients were used in a linear equation and with the aid of a spreadsheet, daily energy consumption was determined using temperature data from the TRY weather database. Daily energy consumption was then summed to estimate a seasonal energy bill. Energy consumption equations used in the analysis are denoted on the graphs.

Slope and off-set of the regression lines are characteristics that describe system performance. The lines have nearly identical slopes. This indicates that for increasing outdoor temperatures, energy consumption increases the same amount for both zone and central systems. For every degree rise in outdoor temperature, daily average energy consumption increases approximately by 2 KWH. The vertical off-set between the lines accounts for the difference in energy consumption. Balance point temperature is another factor that influences energy consumption. The temperature at which no consumption occurs is the balance point. The graphs indicate that the zone system has a balance point approximately 3°F warmer than the central system. This shift in balance point is attributable to system dynamics and thermostat schedule/settings.

Figure 3.3.3
Zone Daily Energy Consumption

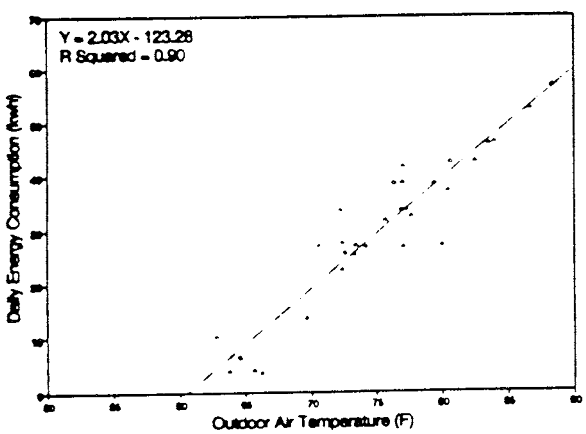
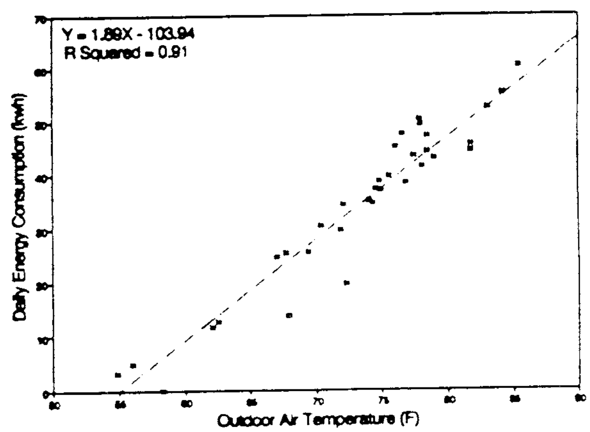


Figure 3.3.4
Central Daily Energy Consumption



Time-of-day rates influence consumer behavior due to their cost structure. The rates are highest during peak periods and lowest during off peak periods. Some utilities offer multi-tiered time-of-day rates such as Baltimore Gas and Electric (BG&E):

Table 3.3.1
Electric Rates

Peak	10 a.m. to 8 p.m.	17.5¢/KWH
Mid-Peak	7 a.m. to 10 a.m. 8 p.m. to 11 p.m.	4.6¢/KWH
Off-Peak	11 p.m. to 7 a.m.	2.8¢/KWH
Standard	All Periods	8.4¢/KWH

BG&E time-of-day and standard rates were used to demonstrate the effect of these rates on seasonal cost for air conditioning for the Research House operating with a zone and central system. Table 3.3.2 displays the estimated costs.

Table 3.3.2
Seasonal Operating Cost

	ZONE SYSTEM	CENTRAL SYSTEM
TIME-OF-DAY RATE	$569	$707
FLAT RATE	$320	$416

House Cooling Load

A Right-J computer summary estimates that cooling load for the Research House is 31,128 Btuh at design outdoor and indoor dry bulb temperatures of 91°F and 75°F, respectively. The measured house cooling load based on daily averages at design condition was 23,600 and 26,100 for the zone and central systems respectively. The Right-J estimate overpredicted the load by 24 percent for the zone system and 16 percent for the central system. The Right-J summary printout is provided in Appendix F.

3.4 Thermal Comfort

The most widely accepted studies on the characterization of thermal comfort have been conducted by Professor P.O. Fanger of Denmark and by Kansas State University for ASHRAE. These studies define indices, predicted mean vote (PMV), and predicted percent dissatisfied (PPD), that characterize thermal comfort in terms of six personal and environmental factors, including metabolic rates, clothing levels, dry bulb temperature, mean radiant temperature, humidity, and room air velocity (Rohles, 1974; Fanger, 1970). A detailed description of these factors is presented in Appendix E, Table E1.

ASHRAE Standard 55-1992 considers conditions environmentally thermally acceptable when 80 percent of a given population in a given area is comfortable. A more detailed description of thermal comfort and a Fortran program for calculating the predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) comfort indices, from ISO 7730, are provided in Appendix E.

Thermal comfort for both tests was characterized using the PMV and PPD indices. PMV depicts the thermal direction of thermal comfort; as indicated on the PMV graph as thermal neutral (zero), warm (positive number) and cool (negative number). PPD index depicts the overall effect relative to the population including cool and warm conditions. The following thermal comfort graphs were developed from measured data during periods of occupancy. Even though the central system was at a constant set point temperature, the same time periods were used to analyze thermal comfort in both zone and central systems to maintain consistency.

Figures 3.4.1 and 3.4.2 show the system's effect on thermal comfort throughout the cooling season. The shape of the curves as well as the location of their peaks are distinguishing features. The zone system peaks are shifted to the left (cooler) as compared to the central system. The zone PMV peaks are also closer to thermal neutrality on the PMV scale. This indicates an ability of the zone system to take greater advantage of cooling load diversity. The width of the base of the curves are wider for the central system indicating a less precise control of temperature as compared to the zone system.

Figure 3.4.1
Central PMV Distribution

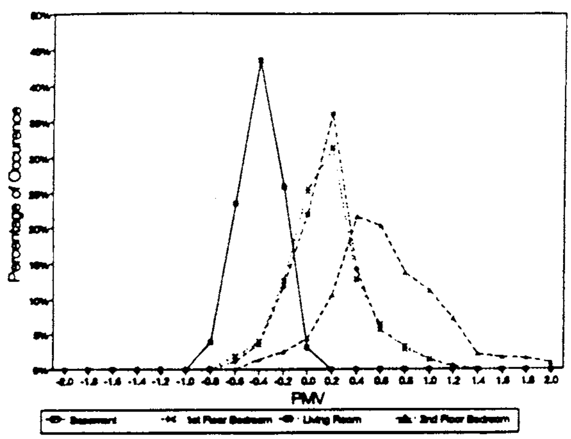
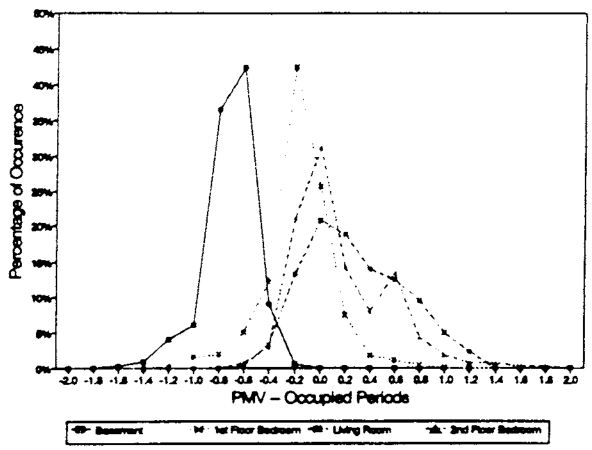


Figure 3.4.2
Zone PMV Distribution



Figures 3.4.3 and 3.4.4 differentiate the performance of the two systems. ASHRAE Standard 55 sets limit of twenty percent of people dissatisfied as the upper limit for thermal discomfort. Other than the basement zone, the zone system operated with tighter control as indicated by narrower curves and more comfort as noted by a shift to the left. The second floor bedroom was under-conditioned by the central system as depicted by the area under the curve beyond PPD of 20. The zone system also had some occurrences of discomfort (PPD 20 and greater) in the second floor bedroom, but not to the extent of the central system. The basement zone seldom ever called for cooling since it was thermally isolated from ambient conditions. Over-cooling of the basement was more severe with the zone system, which was unable to restrict the flow of air when not needed.

Figure 3.4.3
Central PPD Distribution

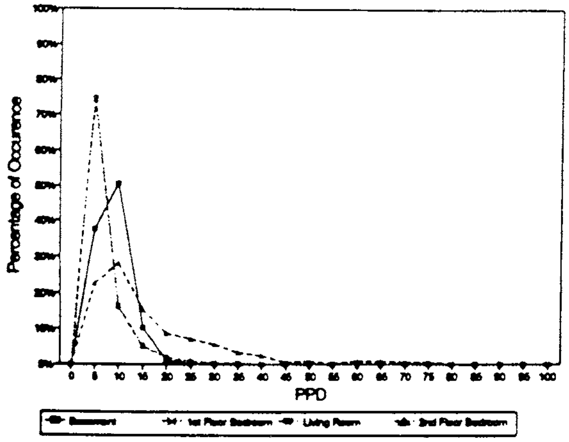
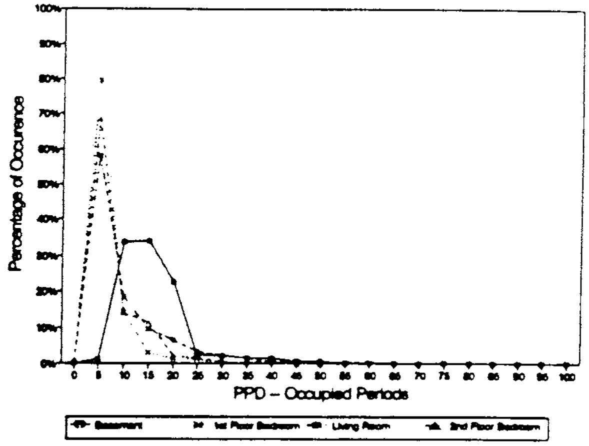


Figure 3.4.4
Zone PPD Distribution



3.5 Humidity Control

In addition to temperature, thermal comfort is strongly influenced by relative humidity. Acceptable levels of relative humidity for comfort is prescribed in ASHRAE Standard 55 (Appendix E). Moisture level in the comfort region is between 40 and 80 grains of moisture. This corresponds to approximately 20 to 60 percent relative humidity.

Moisture was monitored in each zone of the house and plotted in Figures 3.5.1 and 3.5.2. As with the PMV and PPD analysis in the previous section, humidity was evaluated for periods of occupancy. The graphs show that there were periods when humidity exceeded the recommended limit. There was not a discernable difference between the zone and central systems' ability to remove moisture.

Figure 3.5.1
Central Indoor Humidity Occurrences

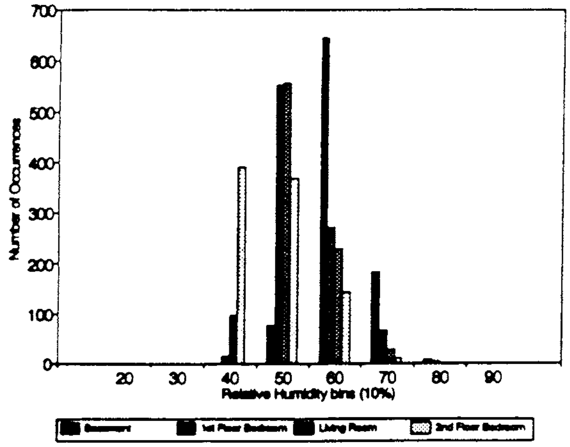
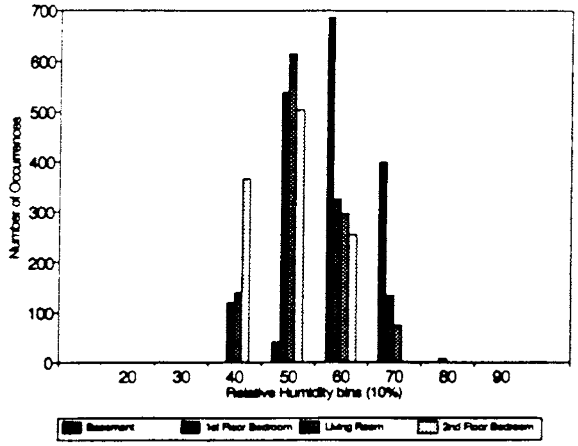


Figure 3.5.2
Zone Indoor Humidity Occurrences



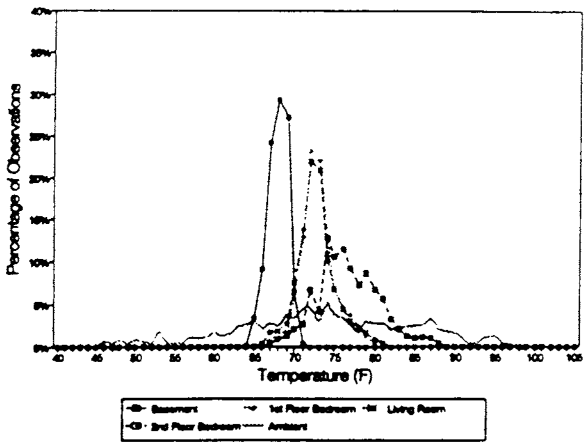
3.6 Frequency of Room Temperature Occurrences

Analysis of the frequency of room temperature occurrences by temperature bins is a technique used to determine how well a thermostat maintains a set temperature in a specific area. Temperature control depends on thermostat location, room size, heating system supply locations, the number and location of doors and windows, and thermostat characteristics

Figure 3.6.1 uses the thermostat sensor temperature to track temperature occurrences. For the central thermostat, the effective temperature setting was below 75°F, being approximately 72°F. The basement and upstairs bedroom show little control of temperature in those zones for different reasons. Due to its location, the basement was consistently cooler than the setpoint in the living room. For the basement, the maximum number of observations were at a temperature of 68°F.

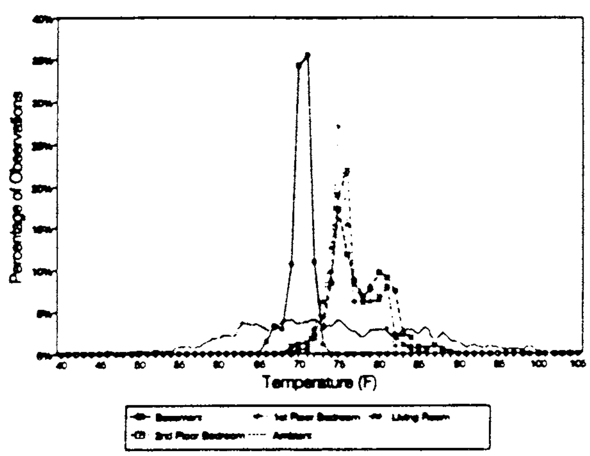
The "stack-effect" contributed excess heat to the second story, and the temperature occurrences for the upstairs bedroom vary widely, with the distribution showing temperature above the setpoint in the living room. The maximum number of observations for the upstairs bedroom occurred at 74°F.

Figure 3.6.1
Central Room Temperature Occurrences



For the zone system, in Figure 3.6.2, the temperature profiles for the first and second floors mimic each other, displaying signs of good control. However, the basement, being cooler, remains out of control, as the temperature never meets the setpoint of 75°F, and the maximum number of observations occurs at 71°F.

Figure 3.6.2
Zone Room Temperature Occurrences



The "hump" to the right in the distributions for the first and second floors is a result of both set up and an insufficient capacity of the cooling system to recover from set-up. For example, in the upstairs bedroom, 56 percent of the temperatures that fall into bins between 73° and 77° during the hours of 10:00 p.m. and 7:00 a.m., the times of set down and the lowest cooling demand. By contrast, 75 percent of the temperatures that fall into the bins in the "hump" - 78° to 82°, occur during the hours of 7:00 a.m. to 10:00 p.m., the set up period and the hours of highest demand.

4.0 REFERENCES

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APPENDIX A

MECHANICAL SYSTEM DESCRIPTION
AND FLOOR PLAN

The Research House was designed to provide a suitable laboratory environment for testing innovative heating and cooling systems. The forced-air distribution system includes special provisions for laboratory testing that were designed to allow manipulation of the system to explore different subsystems and components. Floor plans of the house are presented in Figures A1 and A2.

A microcomputer version of the Air Conditioning Contractors of America (ACCA) Manual J load analysis was run on the Research House (Appendix G). The analysis provided estimated heating and cooling load characteristics and the volumetric flow rate to each room necessary to maintain comfort conditions.

The house was divided into five zones for climate control. There were two zones in the basement, two on the first floor, and one on the second floor. Two zones were used in the basement to separate the furnace room from the remainder of the basement. Two zones on the first floor, one zone on the second floor, and one zone in the basement were conditioned in this study. When a zone requested conditioned air, the appropriate zone damper opened.

Room registers in the basement zones are located one foot from the floor on the perimeter walls. First-floor registers are located in the floor below perimeter glass. Second-floor registers are located both high (seven feet from the floor) and low (one foot from the floor) on interior walls. All registers have operable grills. Returns are located both high (seven feet from the floor) and low (one foot from the floor) on interior walls in the second-floor bedroom zone and the living room zone. The first-floor bedroom zone has only a low return, located one foot from the floor. Returns are located in the basement ceiling for the basement zones.

Figure A1
Five-Zone Air Distribution System
Basement Area of the GRI Home Systems Research House
(copy of engineer's drawing)

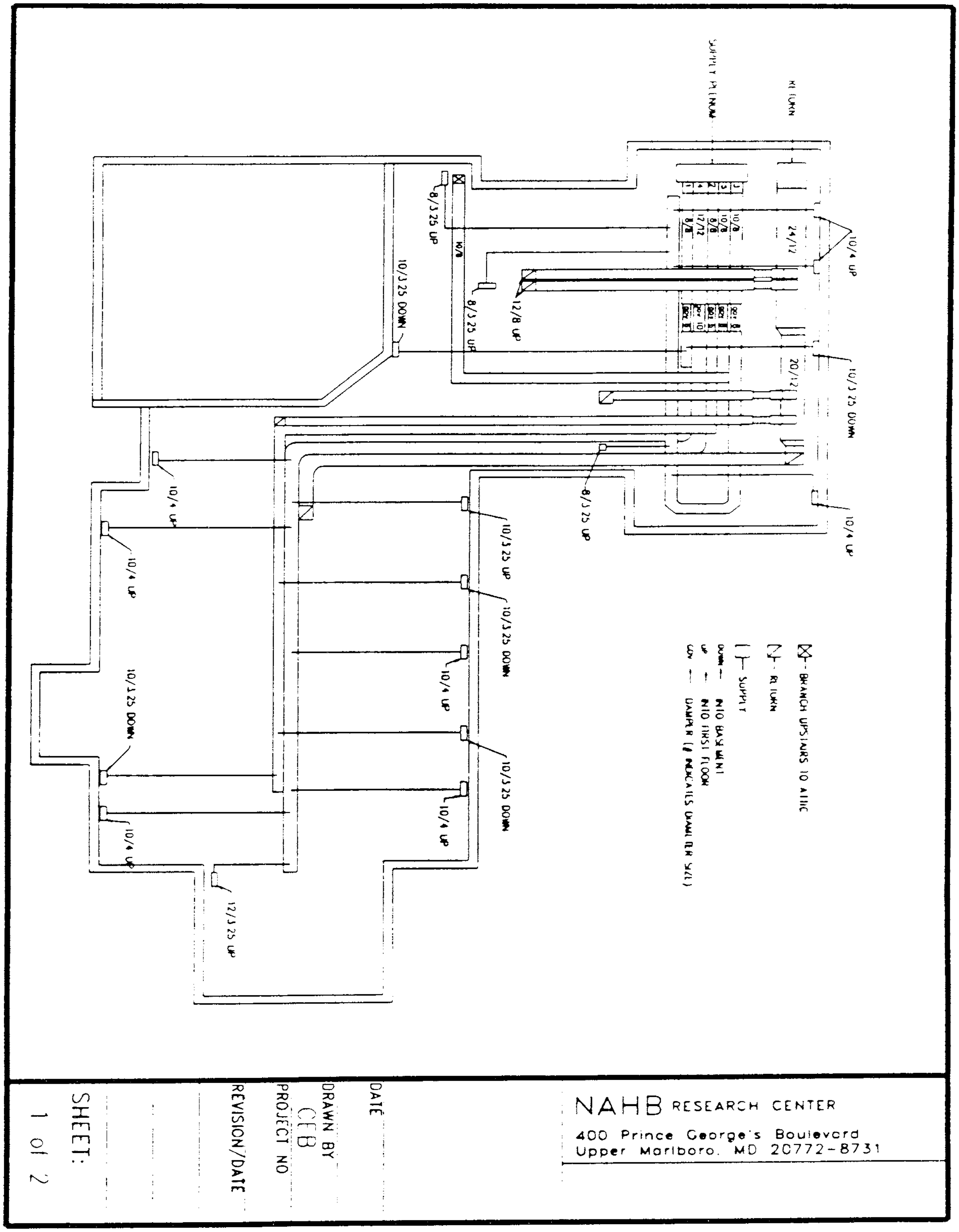
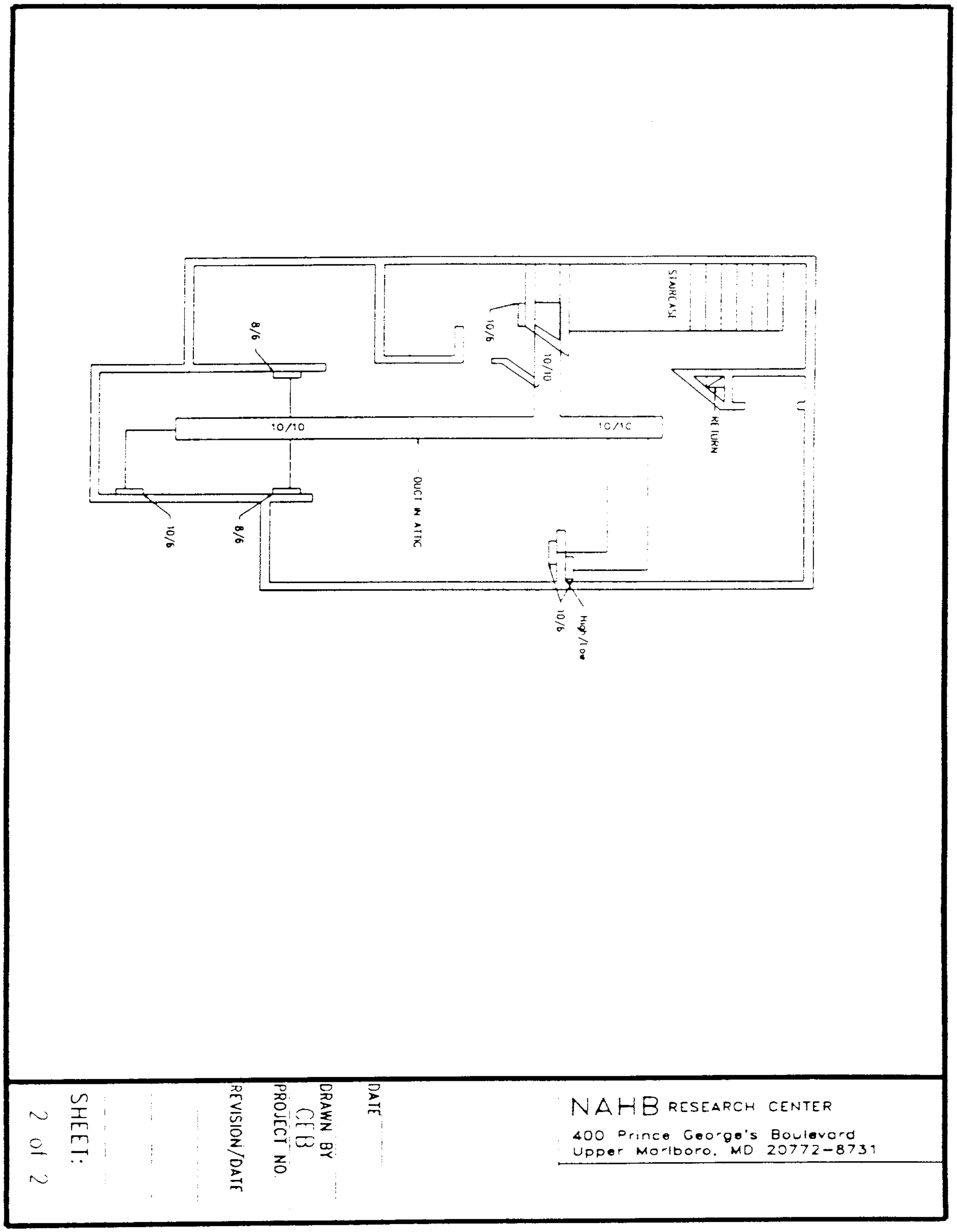


Figure A2
Second-Floor Air Distribution System and Floor Plan
(copy of engineer's drawing)



APPENDIX B

OCCUPANCY SIMULATION PROTOCOL

Schedule of Appliance and Lighting Activities

Appliance Use (Monday through Friday)		
Activity	Time Period	
	On-Time	Off-Time
Shower (1)	0700	0710
Range	0730	0800
Shower (2)	0830	0840
Washing Machine	1000	1030
Clothes Dryer	1200	1300
Oven	1730	1830
Shower (3)	2000	2010
Dishwasher	2100	Cycle Time

Activity	Time Period
TV -- Child's Bedroom*	1900-2100
TV -- Living Room*	2000-2200

*Simulated with 150W light bulb in black bin.

Lighting (seven days per week)	
Activity	Time Period
Master Bedroom (150W)	0800-0830
Child's Bedroom (150W)	0800-0830
Kitchen (200W)	0830-1100
Dining Room (150W)	0830-0900
Downstairs (200W)	0900-1100
Kitchen (200W)	1630-1800
Dining Room Living Room	1800-2200
Child's Bedroom	1900-2200
Master Bedroom	2000-2300

Schedule of Occupant Heat and Moisture Simulation

Room	Watts	Time Period
MBR	150	0800-0830
K/DR	350	0830-0900
DN	200	0900-1100
LR	100	1100-1200
K/DR	200	1200-1230
LR	200	1430-1600
K/DR	200	1600-1700
K/DR	350	1700-1800
LR	200	1800-2200
CBR	100	1900-2200
MBR	150	2200-0800
CBR	75	2200-0800
Occupant Moisture Simulation (seven days per week)		
Room	G/H	Time Period
CBR	170	1600-0800
LR	170	0800-1200,
MBR	170	1500-2200
		1600-0840

APPENDIX C

INSTRUMENTATION

Quality Assurance Objectives

Measurement	Reporting Units	Analytical Technique	Detection Limit	Accuracy ± Percentage	Performance Level
<u>Outdoor Measurement Parameters</u>					
Wind Speed	MPH	Photo Chopper	0.5 mi/hr	15	5 mi/hr
Wind Direction	Degrees	Potentiometer	1° (resolution)	10	0°, 90°, 180°, 270°
Solar Radiation	Btu/hr.ft^2	Photovoltaic	4.43 Btu/hr.ft^2	10	221 Btu/hr.ft^2
Barometric Pressure	MBAR	Piezo-resistance	0.05 mbar (resolution)	0.08	1,000 mbar
Relative Humidity	Percent	Capacitive Thin Film	1% RH (resolution)	10 (absolute)	50% RH
Temperature	°F	Thermistor	0.5°F (resolution)	5	72°F
Precipitation	Inch	Tipping Bucket	0.01 in (resolution)	10	0.1 in
<u>Indoor Measurement Parameters</u>					
Temperature	°F	Thermistor	0.5°F	5	72°F
Relative Humidity	Percent	Capacitive Thin Film	1% RH (resolution)	10 (absolute)	50% RH
<u>HVAC Measurement Parameters</u>					
Temperature	°F	Thermistor	0.5°F (resolution)	5	72°F
Humidity	Percent	Chilled Mirror Dewpoint Sensor	8%	±0.56°C (absolute)	50% (absolute)
Pressure	In. H_2O	Variable Capacitance	0.1 in H_2O	±1	0.5 in H_2O
Air Velocity	ft/min	Hot-Wire Anemometer	20 SFPM	±3% Full Scale	
Flue Gas - CO_2	Percent	NDIR	20 PPM	16/24 hrs Full Scale	11%
<u>Energy</u>					
Electric Power	W	Hall Effect	1 W	10	200W
Appliance Usage	On/Off	Contact Closure	NA	NA	NA
Natural Gas	ft^3/min	Dry Gas Meter	0.125 ft^3	10	0.5 ft^3/min

Instrumentation Specifications

Parameter	Measurement Device	Manufacturer/Model Number
Outdoor Environment		
Wind Speed	Cup Anemometer	Climatronics WM-III
Wind Direction	Vane	Climatronics WM-III
Temperature	Thermistor	Omega OL-705
Relative Humidity	Thin Film Capacitance	Vaisala HMP111A
Precipitation	Tipping Bucket Rain Gauge	Climatronics 6021-A
Barometric Pressure	Piezo-resistive Sensor	Qualimetrics 7105A
Solar Radiation	Silicon Photovoltaic Cell	Qualimetrics 3120
Soil Temperature	Thermistor	Omega OL-703
Indoor/Outdoor		
Air Infiltration (SF_6 Decay Method)	Gas Chromatograph	Shimadzu GC-8A
Indoor Environment		
Temperature - Air	Thermistor	Omega OL-705
Temperature - Wall	Thermistor	Omega OL-709
Temperature - Mean Radiant	Globe and Thermistor	Qualimetrics Z001899 with Omega OL-701
Relative Humidity	Thin Film Capacitance	Vaisala HMP-111A
HVAC System		
Temperature	Thermistor	Omega THX-700-AP
Humidity	Dewpoint Hygrometer	General Eastern Dew-10
Pressure	Variable Capacitance Sensor	Setra 261
Air Velocity	Hot-Wire Anemometer	Kurz Velocity Sensor #435-DC-2
Flue Gas - CO_2	NDIR	Horiba PIR-2000
Boiler Temperatures	Thermistor	Omega OL-710-PP
Energy		
Gas Volume	Dry Gas Meter (with Photodiode Sensor)	Rockwell R-175
Electricity	Watt-Hour Meter	Landis and Gyr, MS-Class 200 TA30
Status		
On/Off Status	Microswitch and Mechanical Relay	Site Configured
Data Acquisition		
	Personal Computer with I/O Boards	IBM Compatible with Metrabyte Corporation-Metrabus System
	Signal Conditioners	Site Configured

Data Acquisition System

- American Research Corporation IBM-compatible computer with Samsung monitor, 640K RAM, 2 floppy-disk drives, 40-megabyte hard-disk drive, serial and parallel ports
- 8 Metrabyte Model MAI-16 analog input boards
- 3 Metrabyte Model MCN-8 counter/time boards
- 1 Metrabyte Model MII-32 logic level input board
- 1 Metrabyte Model MDB-64 driver/board
- 1 Metrabyte Model MEM-32 mercury-wetted relay board
- 1 Metrabyte Model PWR-100 power supply
- 1 15-VDC, 1 12-VDC, and 2 24-VDC power supplies
- Environmental monitoring system software

Wiring Codes and Sensor Location

SA	Thermostat - Air Temperature
SW	Thermostat - Wall Temperature
RA	Room Air Temperature
MR	Mean Radiant Temperature
RH	Relative Humidity
AT	Attic Temperature
GT	Garage Temperature
ST	HVAC Supply Temperature
D	HVAC Supply Dewpoint
SP	HVAC Static Pressure
SV	HVAC Supply Air Velocity
SR	HVAC Supply Register Temperature
XT	HVAC Return Temperature
XD	HVAC Return Dewpoint
XV	HVAC Return Air Velocity
WS	Windspeed
WD	Wind Direction
SR	Solar Radiation
RH	Relative Humidity
BP	Barometric Pressure
PR	Precipitation
ST	Ground Temperatures
*DB	Distribution Boxes

Outdoor Measurement Parameters

Parameter	Code	Site
Windspeed	WS	32 ft. above ground
Wind Direction	WD	32 ft. above ground
Solar Radiation	SR	Roof of house
Relative Humidity	RHA	4 to 6 ft. above ground
Air Temperature	OT	4 to 6 ft. above ground
Barometric Pressure	BP	4 to 6 ft. above ground
Precipitation	PR	Gauge opening at least 12 in. above ground
Ground Temperatures	ST1	Remote site away from houses and trees, 6 to 8 ft. deep
Ground Temperatures	ST2	Adjacent to basement wall, centered vertically between soil surface and plane of the top of the basement floor
Ground Temperatures	ST3	Adjacent to basement wall at base in plane of the top of the basement floor

Indoor Measurement Parameters

Parameter	Code	Site
Air Temperature at Thermostat	SA1	One per conditioned zone
	SA2	
	SA3	
	SA4	
	SA5	
Wall Temperature at Thermostat	SW1	One per conditioned zone
	SW2	
	SW3	
	SW4	
	SW5	
Attic Temperature	AT3	(above BRM#2)
	AT5	(above BRM#1)
	AT4	(above great and dining rooms and kitchen)
Garage Temperature	GT	
Air Temperature at 43 in. Centers of Rooms	RA101	Laundry - Primary Zone #1
	RA102	Furnace
	RA203	Basement - Primary Zone #2
	RA204	Basement near fireplace
	RA305	Brm #2 - Primary Zone #3
	RA306	Bath #2
	RA307	Brm #4
	RA308	Stairwell - top floor
	RA309	Stairwell - 1st floor

Indoor Measurement Parameters
(continued)

Parameter	Code	Site
	RA310	Stairwell - basement
	RA311	Half bath
	RA412	Great room - Primary Zone #4
	RA413	Media room
	RA414	Dining
	RA415	Kitchen
	RA416	Foyer - low
	RA417	Foyer - high
	RA518	Brm #1 - Primary Zone #5
	RA519	Brm #3
	RA520	Sitting room
	RA521	Bath #1
Temperature at 4 in. from Floor	RA122	One per conditioned zone
	RA223	
	RA324	
	RA425	
	RA526	
Air Temperature at 8 ft.	RA327	At primary locations where there is a cathedral ceiling
	RA428	
	RA529	
Air Temperature at 4 in. from Ceiling	RA130	One per conditioned zone
	RA231	
	RA332	
	RA433	
	RA534	
Mean Radiant Temperature at 43 in.	MR101	One per conditioned zone
	MR202	
	MR303	
	MR404	
	MR505	

Indoor Measurement Parameters
(continued)

Parameter	Code	Site
Relative Humidity at 43 in.	RH1	One per conditioned zone
	RH2	
	RH3	
	RH4	
	RH5	
MRT Window 12x43 in.	MR306	Brm #2 - NW corner
MRT Wall 24x43 in.	MR307	Brm #2 - south wall
Room Air Window 12x43 in.	RA335	Brm #2 - NW corner
Room Air Wall 24x43 in.	RA336	Brm #2 - south wall
Main Supply Temperature	STX06	
	STX07	
3-Probe Grid	STX08	
Main Supply Dewpoint	SDX01	
Static Pressure	SPX01	Supply vs. return
	SPX02	Indoor vs. outdoor

Indoor Measurement Parameters
(continued)

Parameter	Code	Site
Supply Air Velocity	SV1	One supply branch per conditioned zone
	SV2	
	SV3	
	SV4	
	SV5	
Main Return Temperature	XT	
Main Return Dewpoint	XD	
Main Return Velocity	XV	
Supply Register Temperature	SR101	South wall
	SR102	Near P1
	SR203	South wall between doors
	SR204	South wall east of doors
	SR205	North wall left of fireplace
	SR306	Brm #2 SW corner
	SR307	Brm #2 SE corner
	SR308	Bath #2
	SR309	Brm #4
	SR310	Hall
	SR311	Half-bath
	SR412	Dining near foyer
	SR413	Dining north wall
	SR414	Great room north wall
	SR415	Media center
	SR416	Great room south wall
	SR417	Kitchen south wall
	SR418	Kitchen east wall

Indoor Measurement Parameters
(continued)

Parameter	Code	Site
	SR519	Brm #3
	SR520	Brm #1
	SR521	Sitting room west
	SR522	Sitting room east
	SR523	Bath #1
Distribution Boxes	DB1	Next to P1 T'stat
	DB2	Next to P2 T'stat
	DB201	Spare
	DB202	Spare
	DB301	Spare - brm #2
	DB302	Spare - brm #2
	DB303	Spare - bath #2
	DB304	Spare - brm #4
	DB305	Spare - brm #4
	DB306	Spare - next to bsmt. stair
	DB3	Next to P3 T'stat
	DB407	Spare - kitchen
	DB408	Spare - kitchen
	DB409	Spare - dining room
	DB410	Spare - dining room
	DB411	Spare - media center
	DB412	Spare - media center
	DB413	Spare - great room
	DB414	Spare - great room
	DB4	Next to P4 T'stat
	DB5	Next to P5 T'stat
	DB515	Spare - bath #1
	DB516	Spare - brm #3
	DB517	Spare - brm #3
	DB518	Spare - sitting room
	DB519	Spare - sitting room
	DB320	Spare - half bath
	DB221	Spare - basement south wall
	DB222	Spare - basement south wall