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ENERGY IMPLICATIONS OF BLOWER OVERRUN STRATEGIES FOR A ZONED RESIDENTIAL FORCED-AIR SYSTEM

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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 ft^2 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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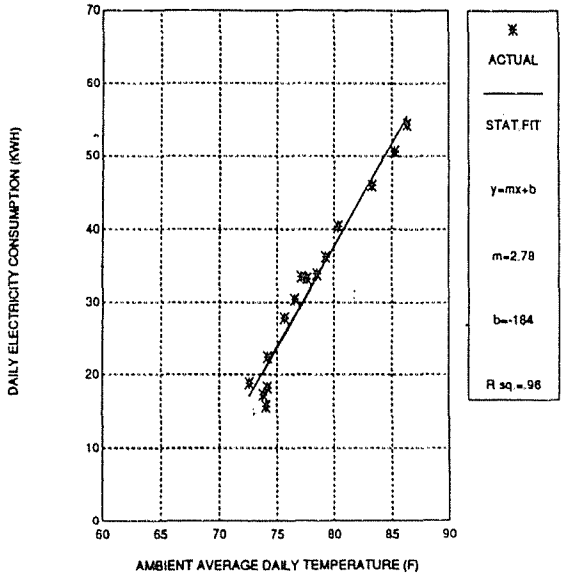

Figure 1 *Daily average condensing unit power consumption for test 1. Test 1 was conventional operation with no zoning, thermostat setup, or blower modulation.*

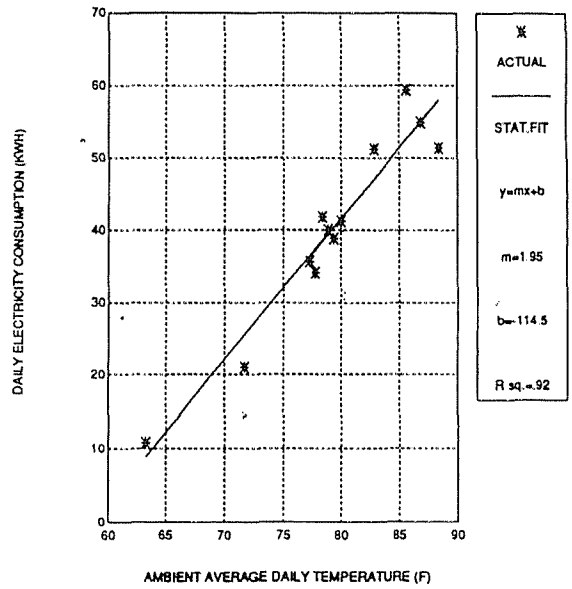

Figure 2 *Daily average condensing unit power consumption for test 2. Test 2 had zoning, blower modulation, and no thermostat setup.*

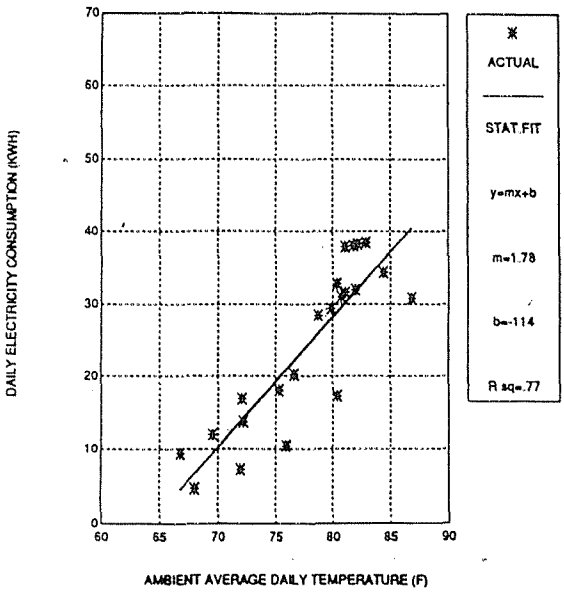

Figure 3 *Daily average condensing unit power consumption for test 3. Test 3 had zoning, blower modulation, and thermostat setup.*

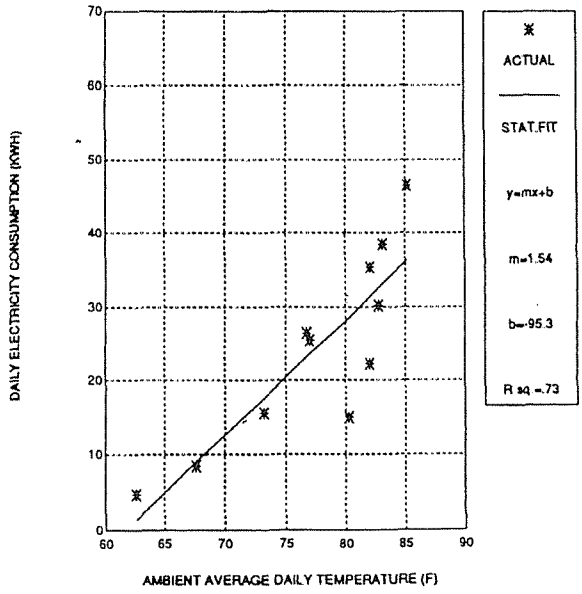

Figure 4 *Daily average condensing unit power consumption for test 4. Test 4 had zoning, blower modulation without overrun, and thermostat setup.*

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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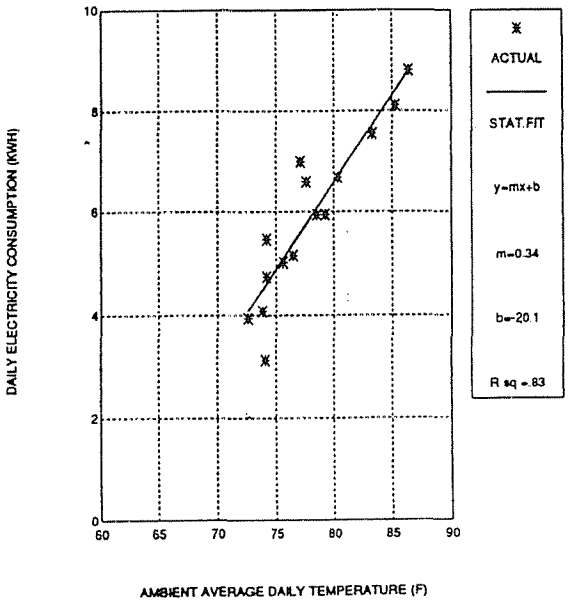

Figure 5 *Daily average blower power consumption for test 1*

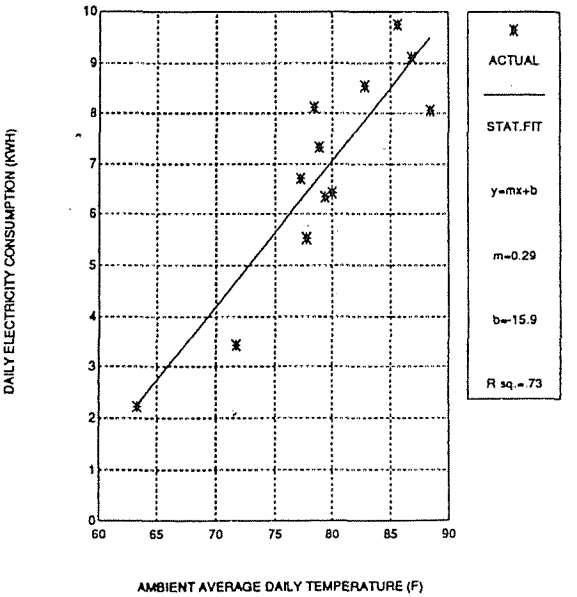

Figure 6 *Daily average blower power consumption for test 2*

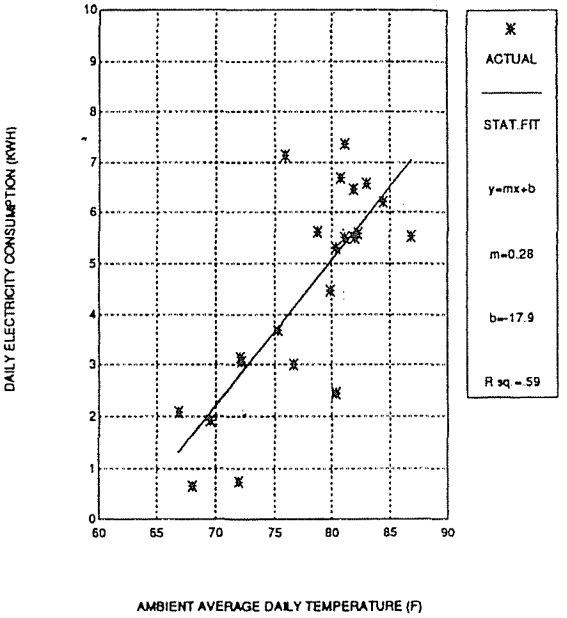

Figure 7 *Daily average blower power consumption for test 3*

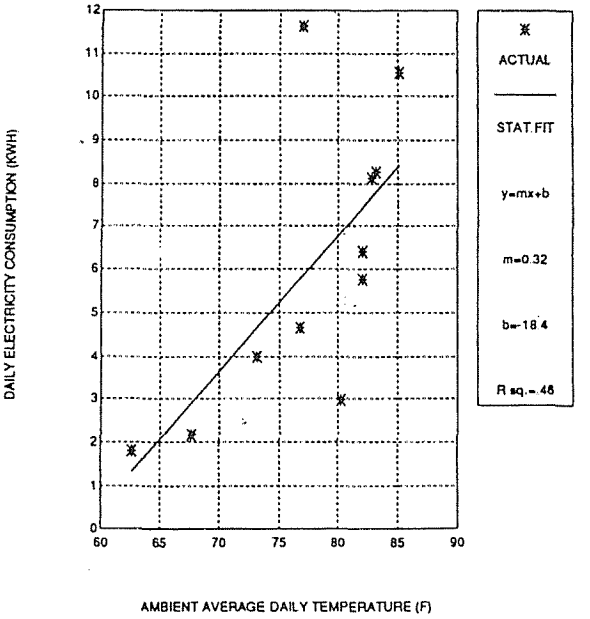

Figure 8 *Daily average blower power consumption for test 4*

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.

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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	

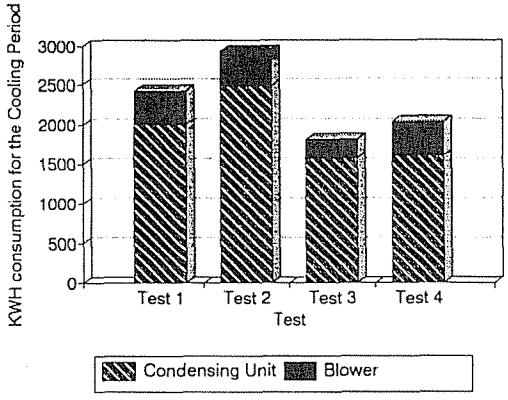

Figure 9 *kWh consumption for cooling season created with historical data and regression lines generated for condensing unit and blower electrical power consumption*

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