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AHRI Air-Conditioning, Heating, and Refrigeration Institute

DOCKET
12-BSTD-1
DATE APR 10 2012
RECD. APR 10 2012

April 10, 2012

California Energy Commission (CEC)

Re: March 13, 2012 45-day Language Hearing for Residential Buildings - 2013 Building Energy Efficiency Standards (AHRI Comments on Residential Zoned Air Conditioning; Docket # 12-BSTD-1)

Dear CEC Staff:

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.

On several occasions last year, we voiced our concerns with respect to the code change proposals outlined in §150.0(m)14 of the *2013 Building Energy Efficiency Standards* (California Code of Regulations, Title 24, Parts 1 and 6.) Although we appreciate CEC's subsequent response to our comments on the proposed code language, we feel that there are serious flaws with the technical arguments supporting these proposed changes. Our previous comments outlining our concerns on the studies that justify the code change proposal are attached (refer to Attachment 1).

We recently reviewed a summary of Mr. Rick Chitwood's field visit to a Sacramento residence with a new zoned system (refer to Attachment 2). The field visit was conducted in September 2011, long after the inclusion of the proposed code language in the standard. Upon reviewing this summary, we feel that the same mistakes are being repeated in the field, thereby leading to findings that are either inconclusive or questionable. The proposed code language in §150.0(m)14 takes these findings into account and unfairly penalizes bypass ducts. We have the following comments on the Sacramento field test report:

- The report does not confirm that 2nd stage cooling demand was present by taking a voltage reading.
- A condensing unit will not energize the 2nd stage cooling mode if the 1st stage control wire is disconnected.
- Unknown cause of the high static pressure and poor airflow must be from poorly installed high friction flex-duct and undersized return or lack of returns.
- High friction flex-duct can be the cause of the high motor watt draw.

- The furnace tripped the factory limit with both zones calling and the bypass closed (this may be an indicator of restricted duct-work.) Technically, with both zones calling and the bypass closed, the system is not zoned. Undersized duct work must be the reason for low airflow when the bypass is closed.
- The report points out that no check was performed for super heat and external static pressure. We feel that such checks should have been conducted.
- The zoned residential system described in the report does not represent a complex system. The system described in the report is simple and should have led to conclusive results during the field visit.
- Survey zoned home may not have been a good installation with low airflow over the heat exchanger, low velocity at the supply grilles, etc.
- The proposed code language is eliminating bypass, when the scrutiny should be on flex duct installations and proper returns. We have not yet seen a study which completely isolates these factors and solely analyzes the effects of bypass ducts on system performance. We believe that these factors contributed heavily towards lowering the overall system performance.

In summary, we feel that the conclusions stated in the Sacramento field test report are flawed due to the repetition of the same mistakes and assumptions that were made in the original Public Interest Energy Research (PIER) study. We want to emphasize that in the case of this field study, the flex duct is the reason for the loss in overall system efficiency, not the bypass air into the return. We urge CEC that reconsider the proposed code language in §150.0(m)14 during the next rulemaking cycle rather than hastily including it in the current rulemaking cycle. It would enable our industry, CEC's consultants and other stakeholders to work together on determining a feasible solution. We strongly feel that all the studies that have been used to justify the proposed code language in §150.0(m)14 should be revisited. There is a need to isolate certain factors that are unrelated to bypass ducts but contribute towards the lowering of the overall system performance.

We appreciate this opportunity to provide comments on residential zoned air conditioning. 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
2111 Wilson Boulevard, Suite 500
Arlington, VA 22201-3001, USA
703-600-0383 Phone
703-562-1942 Fax
aroy@ahrinet.org

Attachments:

1. 2011-10-31 AHRI Comments – Res Zoned AC
2. Sacramento Zoned HVAC System Performance Test

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AHRI Air-Conditioning, Heating, and Refrigeration Institute

October 31, 2011

California Energy Commission (CEC)

Re: October 14, 2011 Residential Staff Workshop - 2013 Building Energy Efficiency Standards (AHRI Comments on Residential Zoned Air-Conditioning)

Dear CEC Staff:

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.

We have several concerns with respect to the measures discussed at the October 14, 2011 CEC staff workshop. Firstly, we feel that stakeholders who are impacted by the code change proposal were not given sufficient time to prepare for this workshop because the notice was sent about two days prior to the workshop, and the agenda and other supporting documents were uploaded on CEC's website a few hours before the workshop. We are greatly disappointed to see that CEC did not make any changes to the original zoned air-conditioning proposal despite receiving several comments from the industry raising serious issues with the proposal discussed at the Codes and Standards Enhancement (CASE) stakeholder meetings and the CEC staff workshops. On August 18, 2011 we facilitated a face-to-face meeting between our members and the CEC staff, and pointed out flaws within the Public Interest Energy Research (PIER) that drove the code change proposal. Following this meeting, some of our member companies also shared data with CEC staff that substantiated the arguments against the code change proposals. We request that CEC staff acknowledge the receipt of AHRI and its members' comments and respond as to why these comments were not substantive enough to warrant any changes to the code change proposal. As a state regulatory agency, CEC must ensure a fair and transparent code development process and should explain why it rejected our comments.

The 350 CFM/ton proposed prescriptive requirement will adversely impact the use of zoning. There are several variable speed systems that are available in the marketplace today. Since variable speed systems can operate at airflow below 350 CFM/ton, the prescriptive airflow requirement in the code language will prevent the use of zoning with variable speed systems. Additionally, the prescriptive requirement will also prevent the use of zoning with single-speed systems that can operate below 350 CFM/ton.

Attached are the following comments that AHRI submitted to CEC and its consultants on residential zoned air-conditioning earlier this year:

1. Clarification on the Manual Zr discussion relating to 10% to 90% bypass air.
2. Canadian study showing that zoning saves energy.
3. Public Service Electric and Gas Company (PSE&G) advertisement recommending customers to close off vents in unused rooms without considering the impact on equipment efficiency. The news release can be accessed via the following link: http://www.pseg.com/info/media/newsreleases/2011/2011-06-07.jsp
4. AHRI comments on July 15, 2011 CEC staff workshop
5. June 6, 2011 AHRI comments on April 12, 2011 residential zoned air-conditioning stakeholder meeting.
6. AHRI comments to CEC consultant on May 17, 2011

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
2111 Wilson Boulevard, Suite 500
Arlington, VA 22201-3001, USA
703-600-0383 Phone
703-562-1942 Fax
aroy@ahrinet.org

Manual Zr Text Relating to 10% to 90% Bypass Air

Section 7-1 Bypass Air Limit

The bypass factor limit value depends on the operating scenario, so there are sets of BPF values for each zone damper system. The smallest BPF in a set may be less than 0.10. The largest BPF in the set may be about 0.90. The range of the set depends on the attributes of the equipment, the external conditions, and the controls.

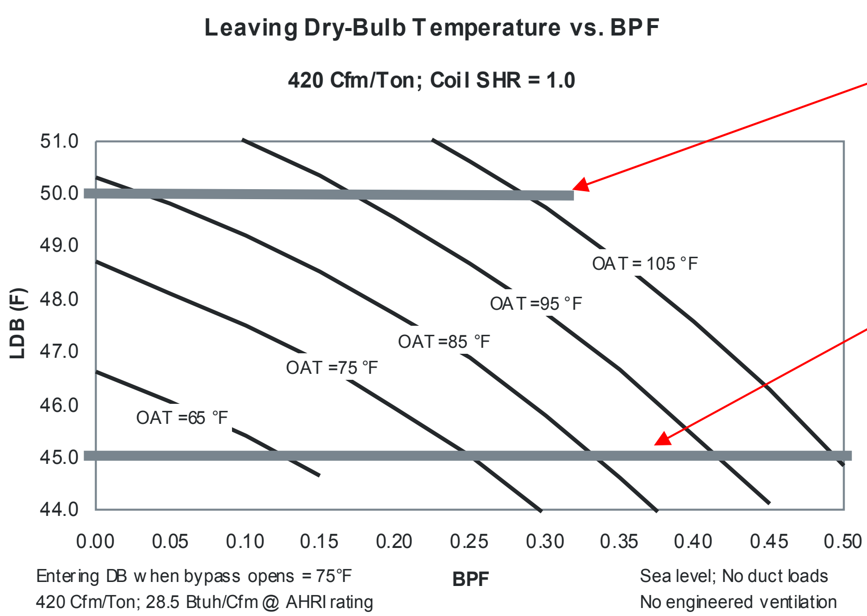
This sentence summarizes the limits of all theoretical scenarios (i.e., not a design guide). There is much more to the story, as explained below.

7-3 Bypass Air Cfm for Cooling

Equipment manufacturers specify a minimum value for leaving air temperature. This value has a significant affect on the amount of bypass air Cfm that will not cause a low temperature problem. For example, the conditional bypass Cfm for a 45°F limit might be 25% of the blower Cfm, or less than 10% of the blower Cfm for a 50°F limit.

Final designs should be based on the low limit value that is specified by the OEM. These values may be in the 38°F to 50°F range.

For example, Figure 7-2 shows acceptable bypass factors for a piece of single-stage, air-cooled equipment when it operates with a dry-coil (CSHR = 1.0).



For 420 Cfm/Ton and a 50F limit, bypass ranges from less than 5% at 105F outdoors to more than 30% at 105F outdoors.

For 420 Cfm/Ton and a 45F limit, bypass ranges from less than 15% at 85F outdoors to more than 50% at 105F outdoors.

Figure 7-2

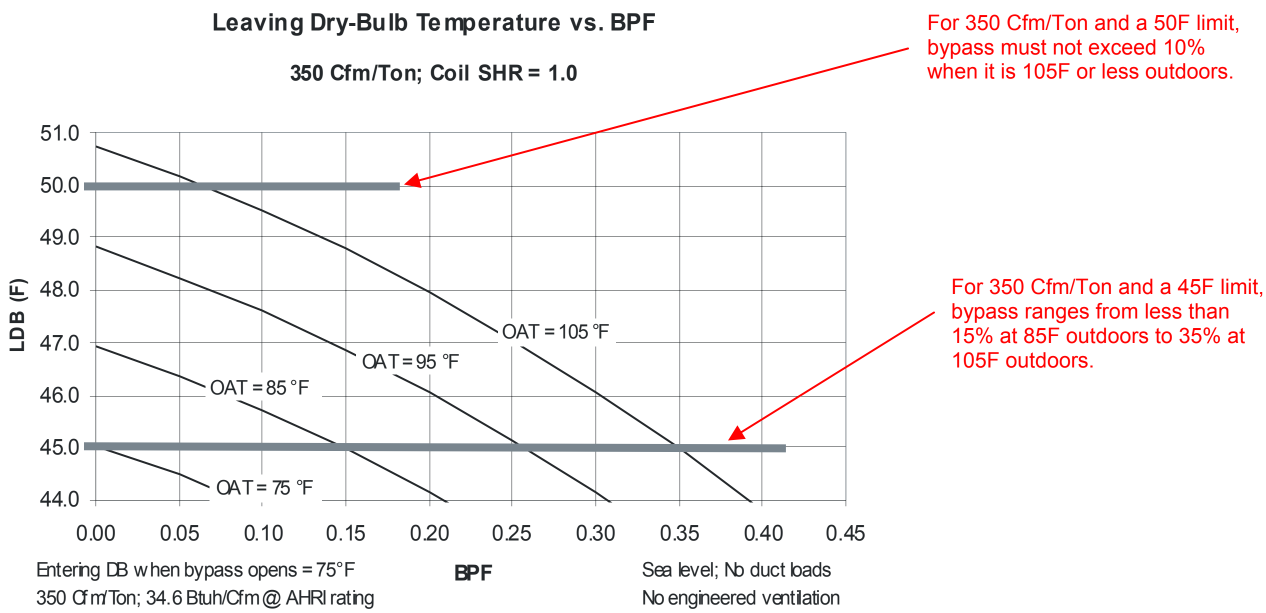


Figure 7-3

7-5 Bypass Air Cfm for a Fossil Fuel Furnace or Electric Heating Coil

The conditional air temperature rise (TR) through a furnace heat exchanger or electric heating coil depends on the heat added to the airflow (Hbtuh), …
The conditional bypass factor (BPF) depends on the conditional temperature rise, the OEM's high limit for leaving air temperature (HLT), …

Conditional Bypass Factors for Heating[1]
Fossil Fuel, Electric Coil, or Refrigerant Coil

B/C	OEM Limit for Leaving Air Dry-bulb (°F)								
	100	110	120	130	140	150	160	170	180
10	0.69	0.77	0.81	0.84	0.86	0.87	0.88	0.91	0.92
20	0.39	0.54	0.63	0.69	0.72	0.77	0.79	0.82	0.83
30	0.09	0.32	0.45	0.54	0.61	0.65	0.69	0.73	0.75
40		0.09	0.27	0.39	0.48	0.54	0.59	0.64	0.67
50			0.09	0.24	0.35	0.43	0.49	0.54	0.59
60				0.09	0.21	0.31	0.39	0.45	0.50
70					0.09	0.20	0.29	0.36	0.42
80						0.09	0.19	0.27	0.34
90							0.09	0.18	0.26

1) The entering dry-bulb temperature is 70°F for the first temperature rise calculation, then it increases to a constant value after some number of air mixing cycles.
2) The initial entering dry-bulb temperature conditionally depends on the space temperature, the return duct load, and on the use of outdoor air for engineered ventilation.
3) B / C = Btuh of heat to blower air / Blower Cfm.
4) Sea level values. (BPF values depend on altitude, which affects burner capacity and the sensible heat equation for air.)

A B/C from 10 to 90, and the 100F to 180F leaving DB limit cover all possible scenarios. Most of these are unlikely.

So, acceptable bypass values can be unusually large if B/C is unusually small, and the limit temperature is unusually high. **This is where the 90% value in Section 7-1 comes from.**

Heating equipment typically operates in this range. Maximum bypass is much less than 90%.

7-6 Bypass Air Cfm for Heat Pump Heating

The conditional capacity of the bypass duct depends on the OEM's high-limit value for discharge air temperature at the indoor refrigerant coil, and the discharge air temperature at the resistance heating coil. It also depends on heating B/C ratio at the indoor coil, and the B/C ratio at the electrical resistance heating coil.

Conditional Bypass Factors for a Heat Pump

KW on Line	Performance	OAT						
		10	20	30	40	50	60	70
0 KW 110°F Limit	Btuh	15.3	18.5	22.4	27.1	32.8	39.3	46.9
	TR	11.1	13.5	16.3	19.7	23.9	28.6	34.1
	BPF	0.72	0.66	0.59	0.51	0.40	0.29	0.15
20.0 KW 155°F Limit	Btuh	83.6	86.8	90.7	95.4	101.1	107.6	115.2
	TR	60.8	63.1	65.9	69.4	73.5	78.2	83.8
	BPF	0.29	0.26	0.22	0.18	Electric coil off		
15.0 KW 155°F Limit	Btuh	66.5	69.7	73.6	78.3	84.0	90.5	98.1
	TR	48.4	50.7	53.5	56.9	61.1	65.8	71.3
	BPF	0.43	0.40	0.37	0.33	Electric coil off		
10.0 KW 155°F Limit	Btuh	49.4	52.6	56.5	61.2	66.9	73.4	81.0
	TR	35.9	38.3	41.1	44.5	48.7	53.4	58.9
	BPF	0.58	0.55	0.52	0.48	Electric coil off		
5.0 KW 155°F Limit	Btuh	32.4	35.6	39.5	44.2	49.9	56.4	64.0
	TR	23.5	25.9	28.7	32.1	36.3	41.0	46.5
	BPF	0.72	0.70	0.66	0.62	Electric coil off		

1) BPF values show the consequences of activating supplemental heat in each temperature bin.
2) Controls should lock out supplemental heat when it is not needed. Supplemental heat should be staged when it is needed.
3) Sea level values (adjust for altitude above 2,500 Feet).

If the electric heating coil is off and if the refrigerant coil limit temperature is 110F, bypass ranges from about 15% at 70F outdoors to about 60% at 30F outdoors. The active electric heating coil limit usually supersedes the refrigeration coil limit when supplemental heat is active.

If the electric heating coil is active, bypass ranges from about 20% at 40F outdoors to about 70% at 10F outdoors. The electric heating coil limit usually supersedes the refrigeration coil limit when supplemental heat is active.

The worst case Cfm (as far as air relief is concerned) is the smaller of the smallest bypass Cfm for heating and the smallest bypass Cfm for cooling.

7-9 Design Value for Bypass Cfm

If the bypass damper is controlled by signals from a static pressure sensor (or flow station), and one or two discharge air temperature sensors; the design Cfm for bypass airway sizing may be considerably larger than the worst case Cfm.

If the bypass damper is an open-close damper or a counterweight damper, the design Cfm for bypass airway sizing is the worst case Cfm.

The design value for bypass Cfm is obtained from the bypass airway sizing worksheet (see Section 8-11).

8-10 Excess Air Worksheet

Use the excess air worksheet (Figure 8-2) to validate a relief strategy. The goal is to drive the residual Cfm value to zero. In this regard, values that are 5% of the blower Cfm, or less, are acceptable.

Bypass Cfm for Cooling (Example problem, page 83)
For a 75°F outdoor temperature and a 45°F limit temperature, the BPF for 400 Cfm per AHRI Ton (Figure 7-2) is 0.25; and for 65°F outdoors, the BPF drops to 0.13. For this example, the zone damper controls are designed to stop the equipment in a normal manner before an OEM limit control takes action. Use the 0.25 BPF for 75°F outdoors. For Figure 8-2, there is 313 (0.25 x 1,250) bypass Cfm for cooling.

Bypass Cfm for Heating (Example problem, page 83)
Figure 7-8 shows that the smallest bypass factor for just refrigerant coil heat (see the 0 KW set) is 0.15 when the outdoor temperature is 70°F. This increases to 0.29 when the outdoor temperature is 60°F; and this increases to 0.51 when it is cold enough (40°F) to activate electric coil heat. The bypass factor for 65°F outdoors is about 0.22. For Figure 8-2, there is 275 (0.22 x 1,250) bypass Cfm for heating.

Excess Air Worksheet (Revised Strategy)		
Excess Air Cfm	**Cooling**	**Heating**
Design value for blower Cfm	1,250	1,250
Smallest design Cfm for zone dampers	375	375
Cfm to spaces that are not zoned	100	100
Excess air Cfm (line 1 - line 2 - line 3)	775	775
Relief Measures and Residual Cfm	**Cooling**	**Heating**
Bypass duct Cfm for LAT limit	313	275
Damper stop Cfm	180	180
Dump zone Cfm		
Selective throttling Cfm		
Over Blow Cfm (0.30 x 375 = 112)	113	113
Residual Cfm (Limit = 0.05 x 1,250 = 63)	169	207

There is a lot of excess air for this example because the Cfm for the smallest zone load is much less than the blower Cfm.

The design value for bypass Cfm is about 25% of the blower Cfm.

Bypass, damper relief plus over blow have not resolved the excess air issue. So, as explained in the rest of the example:

a) Consolidate zones to increases the Cfm for the smallest zone load.
b) Use two-stage equipment.
c) Or consolidate zones and use two-stage equipment

Figure 8-4

The residual Cfm issue would be a much bigger problem if bypass was not an air relief option.

Rationale for Smart Zoned HVAC Technology

- Briefing Presentation

Terry Strack
Strack and Associates

Based upon work carried out with:

- CanmetENERGY,
- McMaster University,
- Ontario Power Authority,
- EnerQuality & builders in 4 Ontario cities,
- *Elect. Dist'n Co:* Chatham-Kent Hydro, Kitchener-Wilmot Hydro,
- *Gas Dist'n Co:* Enbridge Gas, Kitchener Utilities, Union Gas.

Focus of Today's Discussion

1. Context
   - *Briefing on Zoning & Smart-Grid Zoning field trial*
   - *Builder feedback on Zoned HVAC*
   - *Ontario's Smart Grid and related Smart Grid Fund*
   - *Future market vision & breadth of technologies*

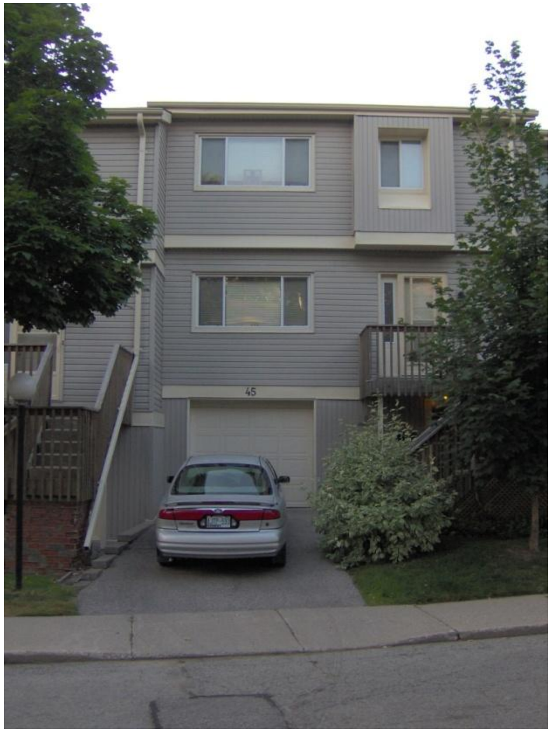
2. "Smart Zoned HVAC" Project Concept
   - *Project Outline*
   - *Comfort check: interest & participation*
   - *Next steps*

Overview of the Zoned Field Trial

- **Focus:** Impact on energy use, peak energy use & comfort with:
  - Forced air zoning
  - Forced air zoning with utility controlled operation
  - Heating & cooling seasons with emphasis on summer peak demand periods
- **Home Sample**:
  - 17 monitored houses (10 zoned systems & 7 non-zoned systems)
  - Retrofit and new-construction applications (3 retrofit and 7 new const'n)
  - south-western Ontario sites
- **Team**:
  - Monitoring and analysis: *McMaster University*
  - Funders and contributors: *NRCan's CanmetENERGY and the OPA*
  - Participating LDCs: *Kitchener-Wilmot Hydro & Chatham-Kent Hydro*

Homes – examples

New Construction
– 2-storey plus finished basement

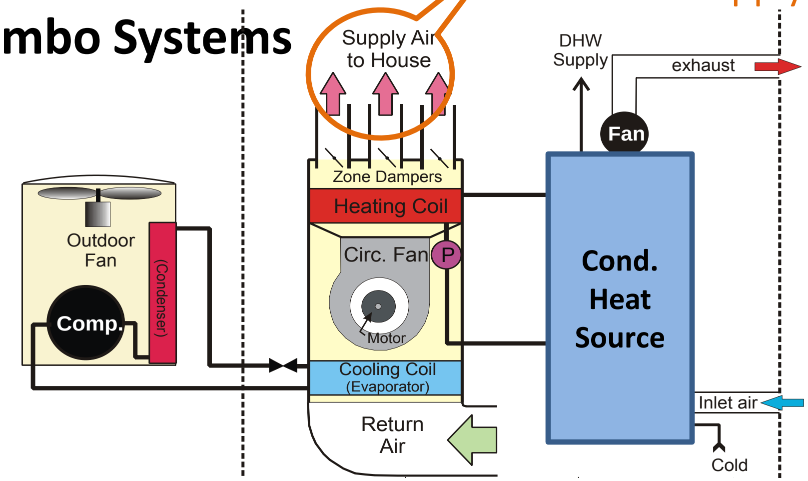


Retrofit
– 3-level townhome

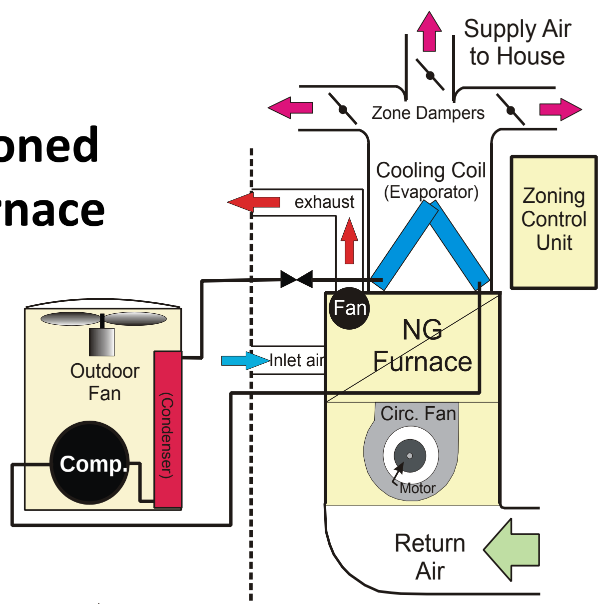
Mechanical Systems - Illustration

With & without "utility control"
of the supply air delivery

Centrally zoned Combo Systems

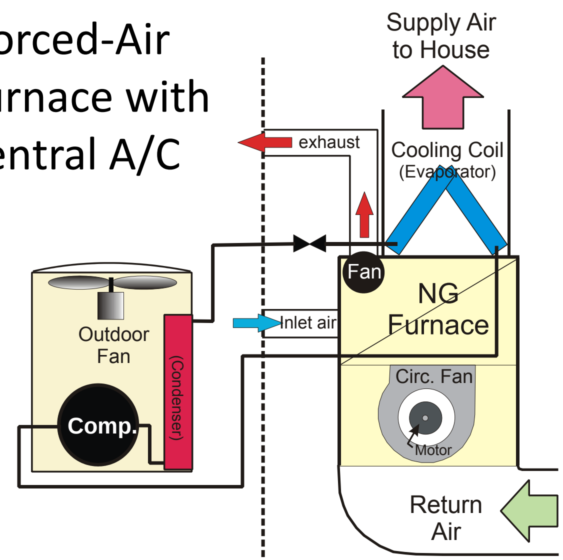


Centrally zoned 2-stage Furnace



Benchmark systems

Forced-Air Furnace with Central A/C



Evaluation Method

Information from Zoned and Non-Zoned Houses in the Field Trial

- Utility Data N.G. & Elect.
- HVAC System Elect. Usage
- Indoor Temp. & Humidity Levels
- Local Weather Conditions
- Homeowner Info. & Feedback

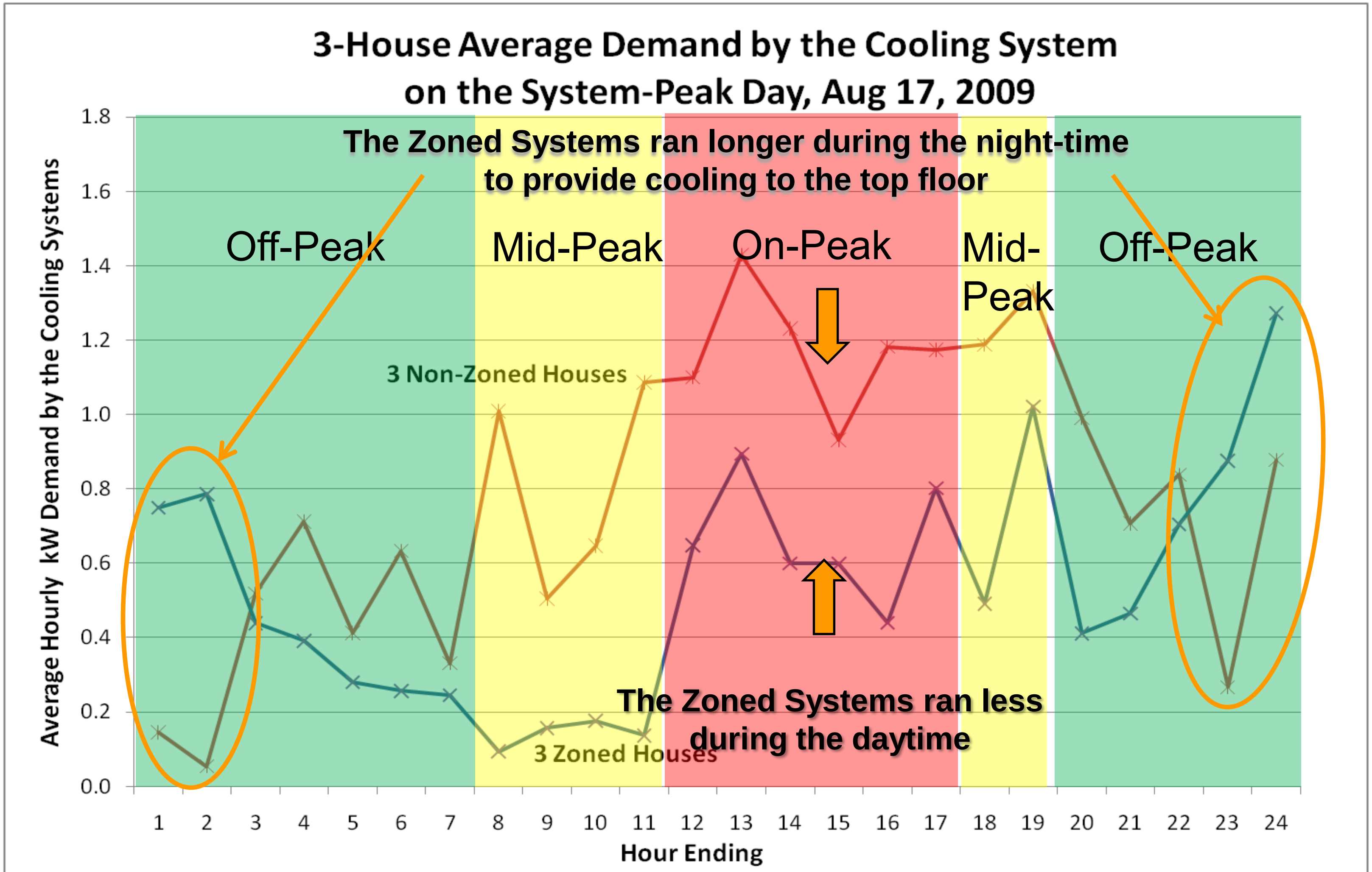
Multi-variable Panel Regression Model of Energy Usage by the HVAC Systems in the test houses

(Completed by the DeGroote School of Business, McMaster University)

Average HVAC Energy Usage Estimates
for Zoned and Non-Zoned Houses, with and without "utility control"
(normalized for house size and occupancy)

Peak Electricity Usage

Avg. Conv.	19.3 kWh
Avg. Zoned	12.9 kWh



Zoned Field Trial

Preliminary Comparison of Peak-Period A/C Loads

HVAC configuration / System control	Avg. A/C Cond.* Demand on a Hot afternoon in 2010	% Savings compared to Standard house
Standard, Non-zoned House	1.03 kW avg	Reference case
Zoned House	0.90 kW avg	13% reduction
Zoned house with “zone-saver” control	0.37 kW avg	64% reduction

* All houses are equipped with identical, 2 ton, SEER 13 A/C condensers

Summary of Field Trial Results

Zoned Systems:

- **improved indoor comfort**, especially on the upper floors of the test houses during occupancy periods
- **reduced average energy** consumption of the heating and cooling systems compared to non-zoned systems in the test houses.
- **reduced cooling loads** and shifted some cooling load to the *off−peak, low−cost period* of the day.
- **High perceived value:** 95% of homeowners rated systems as being "effective" or "very effective" in delivering on claimed performance advantages.
- **Significant Demand Response potential:** can reduce afternoon peak electrical demand during summer peaks
    - without impacting main-floor comfort
    - with ability to rapidly cool the upper floor during off-peak period

Zoned Field Trial

Discussion

Builder Feedback on Zoning

- ***Builders in 4 Ontario cities*** put forward 80 new energy-positive technologies that they potentially wanted to try.
- After systematic examination, ***centralized forced-air zoning*** was found to be in the top 10% of these technologies in all 4 regions
- The 30 participating tract and custom builders are planning their *"discovery homes"*. ***Many plan to use forced-air zoning***.

Note: LEEP/TAP initiative background:

- *Local Energy Efficiency Partnerships (LEEP) is a CanmetENERGY initiative being piloted by EnerQuality in 4 Ontario regions. It enables a group of builders to direct inquiry into new and innovative technologies and practices.*
- *The Technology Adoption Pilot (TAP) Process will lead to the production of a series of demonstration or "discovery" homes that use the technologies selected through LEEP. It is led by EnerQuality.*
- *The two initiatives are inextricably linked and jointly funded by Natural Resources Canada's CanmetENERGY, the Ontario Power Authority, Enbridge Gas Distribution, and Union Gas.*

Zoning Market Feedback

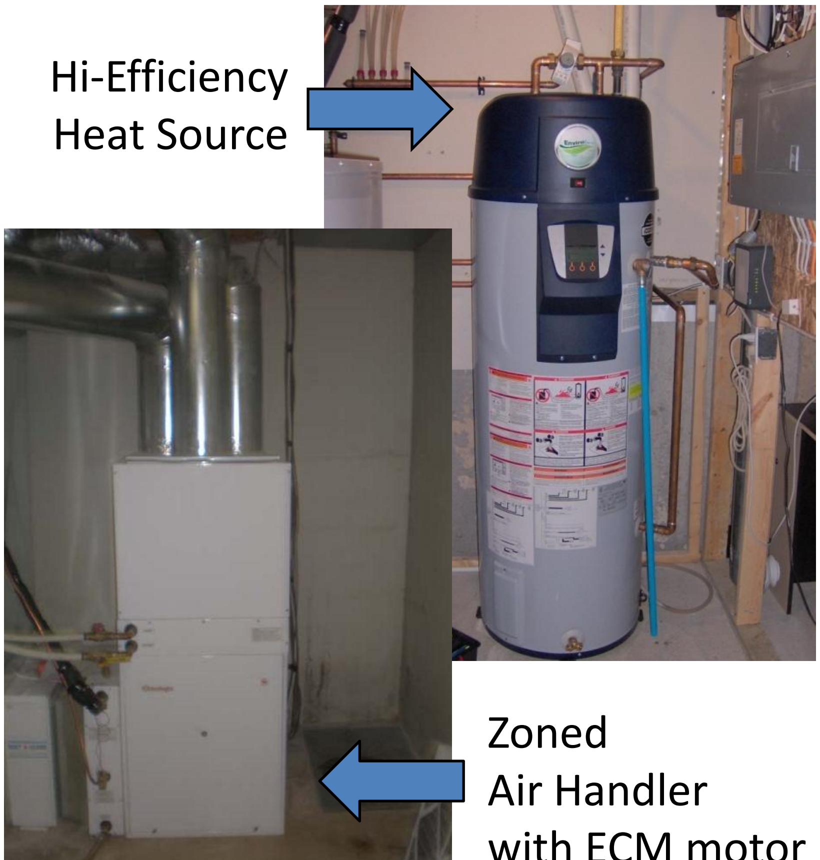
Key Comments from Builders:

- **multi-story houses:** seen as a way to improve homeowner comfort and minimize call-backs related to poor heating and cooling performance on upper levels;
- **Simpler solutions:** centralized forced-air zoning seen as a simpler installation for trades;
- **Suppliers:** multiple suppliers of zoning equipment is essential;
- **Duct Cost:** Installed cost of new zoned duct designs is similar to the cost of conventional duct systems (*this was not expected*);
- **Marketability:** Additional cost for zoning equipment can be sold to customers based on comfort and energy savings. Being considered either as an upgrade option, or as part of the base package.

Ways of Achieving Centralized Zoning

Zoned Combo System

Hi-Efficiency
Heat Source



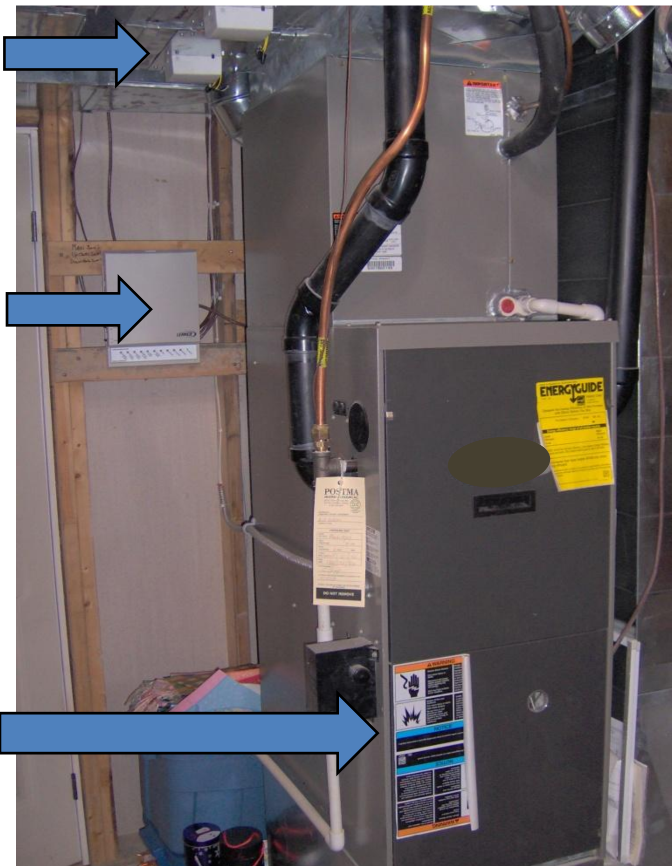
Zoned
Air Handler
with ECM motor

Zoned Furnace System

Zoning
Dampers

Zoning
Controller

Hi-Effic.
2-stage
Furnace
with ECM
motor



Zoned Duct Cost Comparison

RELIANT CLIMATE CONTROL, INC.
WORK QUOTATION

RELIANT CLIMATE CONTROL, INC.
2 - 203 Mohawk Street
Brantford, ON N3S 2X1
(T) 519-758-9277
(C) 519-861-0758
(F) 519-758-8515

Wednesday, February 16, 2011

Quote Expiry:
Wednesday, March 16, 2011

Prepared for:
Bowser Technical, Inc.
Dara Bowser
200 St. George Street
Brantford, ON N3R 1W4
519-756-9116

Quoted by:
Robert Tamilia

Project Identifier :
LEP Example - T9 Unit, 2504
(Standard Duct Work)
Number of Pages: **1**

QUANTITY	DESCRIPTION	UNIT PRICE	TOTAL
	Supply and install:		
	Standard duct system for heating/cooling only.		$ 3,098
	All supply and return air duct work trunks and branch runs as per drawing to basement, main floor and second floor.		
	All return air grills and registers, low voltage thermostat wire to main floor.		
	OMISSIONS: All heating/cooling equipment, bathroom fans, ducting of bath fans, kitchen exhaust duct and HRV system and duct work.		

SUB-TOTAL	$ 3,098.00
H.S.T. 13%	$ 402.74
	$ -
TOTAL	**$ 3,500.74**

This quotation is "as shown".

SPECIAL NOTES:
Thank you for the opportunity to quote!

Authorized by: ____________________

Standard duct system = $3,098

RELIANT CLIMATE CONTROL, INC.
WORK QUOTATION

RELIANT CLIMATE CONTROL, INC.
2 - 203 Mohawk Street
Brantford, ON N3S 2X1
(T) 519-758-9277
(C) 519-861-0758
(F) 519-758-8515

Wednesday, February 16, 2011

Quote Expiry:
Wednesday, March 16, 2011

Prepared for:
Bowser Technical, Inc.
Dara Bowser
200 St. George Street
Brantford, ON N3R 1W4
519-756-9116

Quoted by:
Robert Tamilia

Project Identifier :
LEP Example - T9 Unit, 2504
(Zoned Duct Work)
Number of Pages: **1**

QUANTITY	DESCRIPTION	UNIT PRICE	TOTAL
	Supply and install:		
	3 Zone duct system for heating/cooling only.		$ 3,372
	3 supply air trunks with common return air duct as per drawing to basement, main floor and second floor.		
	All return air grills and registers, low voltage thermostat wire rough in to basement, main floor and second floor.		
	Duct seal all joints to supply and return air.		
	OMISSIONS: All heating/cooling equipment, bathroom fans, ducting of bath fans, kitchen exhaust duct and HRV system and duct work.		

SUB-TOTAL	$ 3,372.00
H.S.T. 13%	$ 438.36
	$ -
TOTAL	**$ 3,810.36**

This quotation is "as shown".

SPECIAL NOTES:
Thank you for the opportunity to quote!

Authorized by: ____________________

Zoned duct system = $3,372

Zoned Duct Cost Comparison

- Standard System: $3,098 (before taxes)
- Zoned System: $3,372 (before taxes)
  - Extra over standard: $ 274 (incl'g duct seal @ $250)
  - Extra if no duct seal: $ 24

  - Labour costs similar/identical

Builder Feedback on Zoning

Discussion

Smart Grid in Ontario

- Ontario has deployed smart meters to all residential houses and small businesses
- Time of Use Pricing will be the new norm for residential customers

	5.9 ¢ Off-peak	8.9 ¢ Mid-peak	10.7 ¢ On-peak
All-in Price:	11.0 ¢	14.5 ¢	16.6¢

Summer - Weekdays
(May 1 - October 31)

The Summer Weekdays
The highest energy prices occur over the afternoon largely due to greater air-conditioning use. That's why the on-peak rated is from 11 a.m. to 5 p.m.

Smart Grid Fund (1 of 4)

Ontario Ministry of Energy:

- $50 million fund announced April 27, 2011
- Two project categories:
  - *Capacity building SGF projects*
  - *Demonstration SGF projects*
- Submission of proposals:
  - *2-stage proposal process (online):*
    - Project Overview Submission due by May 27, 2011
    - Business Case Submission due by July 6, 2011

Smart Grid Fund (2 of 4)

- **SGF Objectives:**
  - *Developing and advancing the smart grid in Ontario*
  - *Creating economic development opportunities, including jobs in Ontario*
  - *Reduce risk and uncertainty of electricity sector investments by enabling utilities and stakeholders to develop, test and evaluate smart-grid technologies and business models*

Smart Grid Fund (3 of 4)

- **SG Fund Outcomes:**
  - *Accelerate Commercialization of SG technologies and build market competitiveness*
  - *Establish Ontario as a leader and location in which to develop and manufacture SG products & services*
  - *Ensure a coordinated approach which will focus on provincial benefits/policy and deliver scalable solutions*
  - *Identify potential solutions to the development and deployment SG technologies in Ontario*

Smart Grid Fund (4 of 4)

- **Capacity Building Category** - overview:
  - *Project timeframe: maximum of 4 years*
  - *Minimum project total: $1.5 M*
  - *SGF contributions up to 30% of eligible project cost, to a maximum of $5 M per project.*
  - *50% of total project costs must be from non-government sources*
  - *Organizations must have 3 yrs of active operations*

Smart Grid & Smart-Grid Fund

Discussion

Vision for Residential HVAC

- Ontario deployment
  - New housing sector will increasingly deploy zoned HVAC systems
  - Existing housing sector will employ a wide variety of zoned HVAC options
- Homeowners
  - Zoned systems will provide increased comfort and flexibility in managing rising energy costs (both electricity and natural gas)
- Utilities
  - Zoned cooling systems will provide new demand response (DR) capabilities
    - help manage peak electricity demands and improve system reliability
    - respond to information and signals conveyed by the Smart Grid’s 2-way communication capabilities
- We are calling this type of system:

“Smart Zoned HVAC”

Smart Zoned HVAC Technology

Smart Grid + Smart Controls + Zoned HVAC

= Reduced Peak-period power demand

+ lower electricity costs for cooling

+ superior homeowner comfort

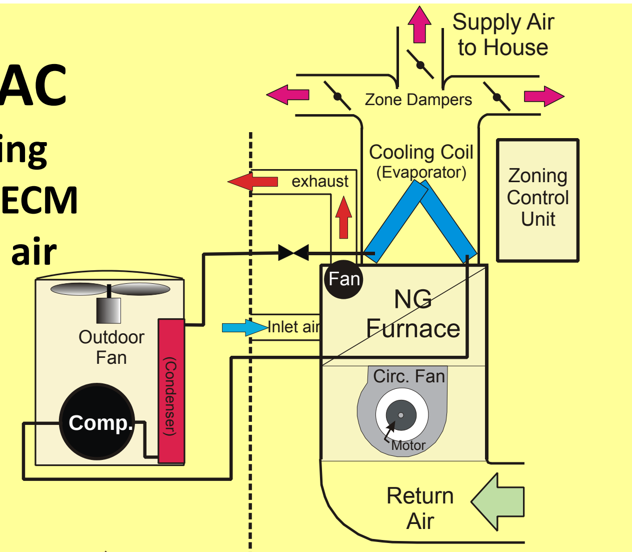
Smart Zoned HVAC Technology

In-house Smart Controls

Smart Thermostats / In-home Displays /Home Energy Managements Systems / Demand Response Switches

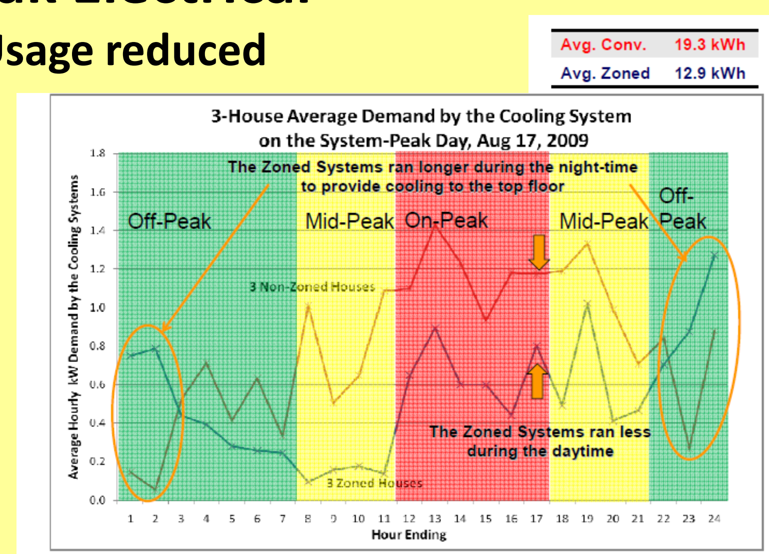
Advanced HVAC

<u>Example</u>: modulating heating equipment, ECM fans, and two-stage air conditioners



Peak Electrical

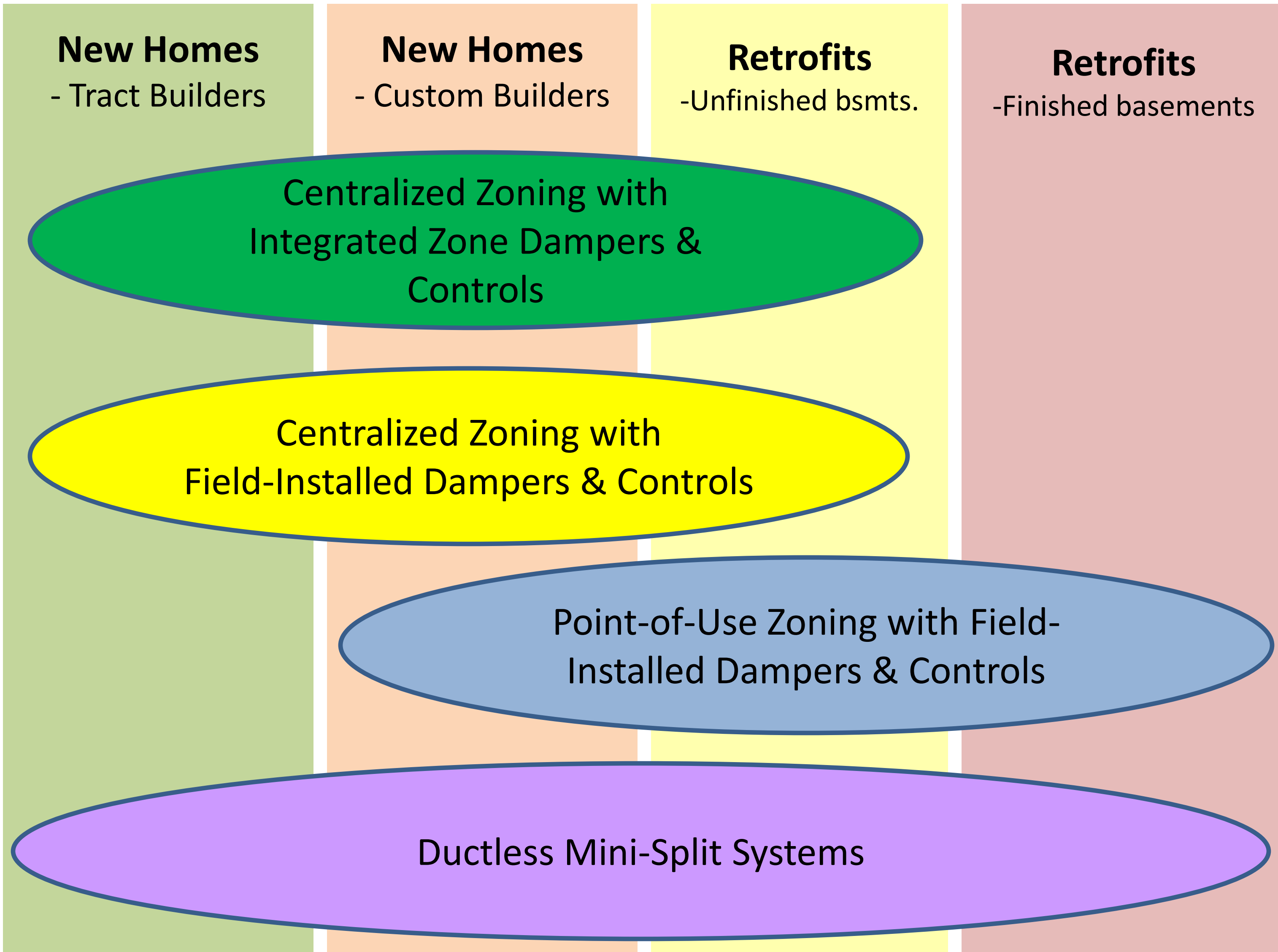
Usage reduced



Smart Grid

2-way communication, Smart Meters & Time-of-Use Electricity Rates

Smart Zoned HVAC Tech & Appl’ns



Smart Zoned HVAC Technology

- Multiple vendors capable of delivering *Smart Zoned Technology*
- Various equipment configurations are possible
- Can be applied to a wide range of new-housing and retrofit applications

Smart Zoned HVAC Technology

Discussion

Smart Zoned HVAC Project Concept

- To put in place the necessary infrastructure to support the rapid deployment of ***Smart Zoned HVAC Systems*** in Ontario.
- **Potential role for HRAI** in this new initiative
- Funding opportunity from the ***Smart Grid Fund***
  - *Capacity Building Category*

What's in it for HRAI Members

- **HRAI's membership** includes manufacturers, wholesalers and HVAC contractors.

  … all are needed to deliver Smart Zoned HVAC solutions

- **New business** opportunities for HRAI members:
  - ***Manufacturers & wholesalers:*** *new market-drivers to support the sale of higher-efficiency & higher-end equipment* (e.g. furnaces, A/C condensers, thermostats, zoning dampers and controls)
  - ***HVAC Contractors:*** *new retrofit solutions to deliver energy management & comfort benefits to homeowners* (e.g. adds zoning and two-stage condensers to their sales tool kit)

What’s in it for HRAI

- Expands **HRAI’s** leadership role
- Expands **HRAI’s** training & software business
- Fits well, and would likely expand **HRAI’s** delivery of programs such as ***“saveONenergy”***
- Provides new membership opportunities

  e.g. build up “energy efficiency contractors network”

Major Project Tasks

- ***6 Major Tasks:***
  1. Overall Project Management
  2. Smart Zoned HVAC System Specification
  3. Hardware Solutions to deliver SZ HVAC to market
  4. Energy and Demand Response Performance Modeling and Rating guidelines
  5. Builder and contractor engagement
  6. Show-case demonstrations of Smart Zoned HVAC technologies in different regions of Ontario

Task 1: Smart Zoned HVAC System Project Management

- Overall project coordination provided by HRAI
- Provide a centralized focal point for the *Smart-Zoned HVAC Technology* concept
  - Specific hardware implementations will be delivered by individual manufacturers & suppliers
- Coordinate and disseminate new knowledge and know-how used to support and promote *Smart-Zoned HVAC Technologies* in the market

Task 2: Smart Zoned HVAC System Specifications

Better practice functional specifications for:

- Grid interactive HVAC interfaces
- Zoned HVAC technology
  - *Centralized Zoned Technology*
    - Centrally zoned forced air systems
    - Centrally zoned duct designs
  - *Distributed zoned duct systems (branch duct installations)*
  - *Ductless mini-split systems*

... satisfied by *a number of manufacturers* in each case

Example Equipment Specification

An Effective Centrally-Zoned System needs:

- Mechanically dampers on the main zoning supply trunks, close to or within the heating/cooling device;
- 3 or more zones of supply air;
- A condensing heat generator for space heating;
- An ECM, variable speed motor on the circulation blower;
- Programmable thermostats for each zone; and,
- A master system operating-mode switch to set the zoned system fully in either heating or cooling mode, *to prevent alternate heating and cooling cycles from occurring in "shoulder seasons";*

Plus <u>all</u> Smart Zoned HVAC systems need:

- An in-home display (IHD) and/or automated controls which receive information from the smart grid to manage the peak-period cooling load in a portion of the home.

Task 3: Development of Smart Zoned Hardware Solutions

- Development of a range of **Smart Zoned HVAC technology hardware packages** designed to deliver the performance and functionality described by the Smart Zoned HVAC Specification
  - *The technology specification will likely define a range of displayed information options and control options with increased functionality and performance expectations.*

Task 4: Modeling & Rating of Smart Zoned HVAC Technology

- Develop **Energy Modeling** and propose **Rating Methods** for Smart Zoned HVAC Systems
  - *HRAI to advise on equipment configurations and application scenarios of SZ HVAC systems;*
  - *Develop new modeling modules which will support the application of SZ HVAC Systems in a range of hardware implementations & housing types;*
  - *Propose new mechanisms to give credit for the energy and demand-response benefits that accrue from the operation of SZ HVAC Systems*.

Task 5: Industry Outreach

- **Builder and contractor outreach** activities to build SZ HVAC system delivery capacity in the new housing and retrofit market segments
  - *new training initiatives by HRAI on Smart Zoned technology directed at manufacturers, wholesalers and HVAC contractors*
  - *EnerQuality's activities with home builders and renovators to include Smart Zoned technology*

Task 6: Regional Showcase of Smart Zoned HVAC Technologies

- Showcasing of **Smart Zoned HVAC systems** in five regions of Ontario, each with targets in the new-housing and retrofit market segments.
  - *Set up regional sites to promote the benefits of Smart–Zoned HVAC Systems and technologies in different regions of the province.*
  - *Showcase different hardware solution from different manufacturers*
  - *Used to engage local stakeholders, homebuyers and homeowners*

Project stakeholders

- Ontario Ministry of Energy
- HRAI (project lead)
- EnerQuality
- NRCan – CanmetENERGY
- Equipment manufacturers
  - *HVAC equipment*
  - *Controls / display technologies*
- Contractors and home builders
- Ontario Power Authority
- Ontario electricity and natural gas distribution companies
- Municipal and Regional governments

Project Budget

Task	Funding ($)	In-Kind ($)	Total ($)
1. Project Management	$ 300K	$ 0	$ 300K
2. Develop Smart Zoned HVAC Functional Specification	$ 300K	$ 600K	$ 900K
3. Develop SZ Hardware Solutions	$ 100K	$ 300K	$ 400K
4. Performance Models & Rating Methods for SZ HVAC Systems	$ 100K	$ 300K	$ 400K
5. Engage Builders & Contractors; Develop SZ training modules	$ 100K	$ 600K	$ 700K
6. Regional Showcase of SZ HVAC	$ 400K	$1,300K	$1,700K
Totals	**$1,300K**	**$3,100K**	**$4,400K**

Summary and Discussion Points

- ***The Smart Zoned HVAC concept*** has been evaluated and field tested by CanmetENERGY and its energy- and building-sector partners
  - Now have demonstrated proof of concept for peak shaving benefits in Ontario

- **Looking for feedback** on the proposed capacity building project:
  - Interest
    - Who would like to participate?
    - Is there a will to proceed?
    - Potential contributions?
  - If want to proceed then ... logistics
    - Is there agreement on scope?
    - How can HRAI best lead?
    - Does the nominal budget seem reasonable?
    - Need to develop a stage 1 submission to the Smart-Grid Fund (due May 27th)
  - Next steps

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AHRI Air-Conditioning, Heating, and Refrigeration Institute

August 8, 2011

Mr. Mazi Shirakh
mshirakh@energ.state.ca.us
California Energy Commission (CEC)
1516 Ninth St. Mail Stop 37
Sacramento, CA 95814

RE: AHRI Comments on 2013 Building Efficiency Standards Staff Workshop on Residential Zoned A/C – July 15, 2011 (Docket Number 10-BSTD-01)

Dear Mr. Shirakh,

AHRI would like to thank you for giving us the opportunity to participate in your 2013 building energy efficiency standards workshops and to submit comments on the data/findings and code change proposals with respect to residential zoned air-conditioning. On June 2, 2011 and June, 2011, we submitted comments to you and Mr. Wilcox outlining the industry's concerns about the studies that were conducted to justify that the performance path zoning credit should be eliminated. We also provided you with two studies that confirm substantial energy savings through zoning when the system is properly designed and installed. We feel that the July 15, 2011 CEC workshop did not address any of our written comments. Although we are submitting additional comments on the issues discussed at the July 15, 2011 CEC staff workshop, we are not sure whether these comments will be addressed in the standards process. We recommend that a meeting involving our member manufacturers, CEC staff and CEC technical contractors be scheduled to discuss the technical studies, code change proposals and the industry's concerns.

AHRI and the members of the AHRI Zone Control Systems Technology Section believe the reports being submitted are biased against air-zoning. We would like to point out what we feel was information omitted or not present in the report.

1. The report of this workshop stated that previous studies from 1991 (Oppenhiem) and 1994 (NAHB/Carrier) stated that zoning can cause an increase in energy costs, as much as 35%* more, when all thermostats are kept at the same temperature. The report to the CEC does not mention that both of these studies also clearly stated that zoning can save as much as 25% when the zone systems use setback. As you are already aware, Title 24 clearly mandates the use of setback thermostats. If a homeowner has the ability to shut off the air conditioning in unused rooms, the homeowner will do it, either with a thermostat or by closing the supply vents. Why was this clear energy savings result not included? This in itself should convince the CEC that zoning not only should remain as part of the program, but should be further be promoted as a low cost method of providing substantial energy savings.

**The 35% more number, quoted on the 1994 report, is believed to be taken from the 1991 report which was reported as only 20% more energy, and an error in transcription must have taken place as the 1991 report has no such number in it.*

2. We have to express our objections to the report stating that bypass ducts adversely affect the efficiency of the air conditioning systems. Studies of performance and efficiency of systems should be performed under controlled conditions, not based on four year old systems where the quality of installation may be questionable. Mileage ratings for cars and energy efficiencies for all appliances are based on controlled laboratory conditions, not on a four year old product in the field whose workmanship was below accepted industry standards.

3. The alternatives to zone damper systems, such as multiple HVAC Systems or mini-splits, as recommended in this report, are not the answer. Adding more units only adds to the overall installation cost. The initial cost of installing two air conditioners is a lot more than the installation costs associated with zone damper systems. The utilities in California and for that matter the whole country are looking for ways to reduce their loads, not increase them.

CEC's best answer for a simple, low cost energy saving HVAC solution is zoning. Installing high efficiency equipment into an old, leaky and/or poorly designed duct system is the real problem. Providing an incentive to install zone damper systems will compel HVAC installers to fix leaky ducts and correct poorly designed duct systems.

The majority of homeowners are not always comfortable throughout their home. The more uncomfortable they are, the more often they adjust their thermostats. In many cases, the homeowner is over compensating in one area to get another area comfortable, i.e. the homeowner is wasting energy. If zoning can make the occupants more comfortable, in every zone of the home, they will be adjusting their thermostats less often and saving more energy.

Federal energy legislation on regional standards for HVAC that will be in place soon will mostly be met by using two stage equipment and variable speed fans. HVAC units are sized for outdoor design conditions. At design conditions these units can often still heat or cool a single zone just on first stage capacity. This in itself speaks for the savings zoning can provide and utilities are looking for during peak loads. See the attached chart which was a zoned HVAC technology study for Canadian utilities showing 30% less kWh consumed with zoned systems vs. non-zoned systems. Instead of shutting off the entire system at peak times, the unit can run on first stage and still cool just the family room/kitchen area. This would solve the problem for both the utility and the consumer. The utility will not have to shut down the unit and the homeowner gets cooling at half load for the zone which is occupied by the family.

We strongly urge the CEC to continue the inclusion of zoning in all future energy programs. AHRI and its Zoning Section members would appreciate a face-to-face meeting or teleconference with the CEC staff on this issue, so that the CEC can be fully made aware of the benefits of zone damper systems in both residential and light commercial applications.

Sincerely,

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Members of AHRI's Zone Control Systems Technology Section:

Arzel Zoning Technology, Inc. – Dennis Laughlin
Carrier Corporation – Bob Swilik
Duro Dyne Corp. – Steve Martin
EWC Controls – Mike Reilly
Honeywell International, Inc. – David Arneson
Jackson Systems, LLC – Thomas Jackson
Lennox International, Inc. – Thomas Kerber
Research Products Corporation – Eric Brodsky
Trane – Tim Storm
Zonefirst – Dick Foster

Attachments:

1. Energy Implications of Blower Overrun Strategies for a Zoned Residential Forced-Air System
2. Field Investigation of Carrier Residential Zoning System
3. Peak Electricity Usage Chart – Zoning Energy Savings in Canada
4. AHRI Comments on July 15, 2011 Residential Zoned AC Presentation

3529

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

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.

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

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

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

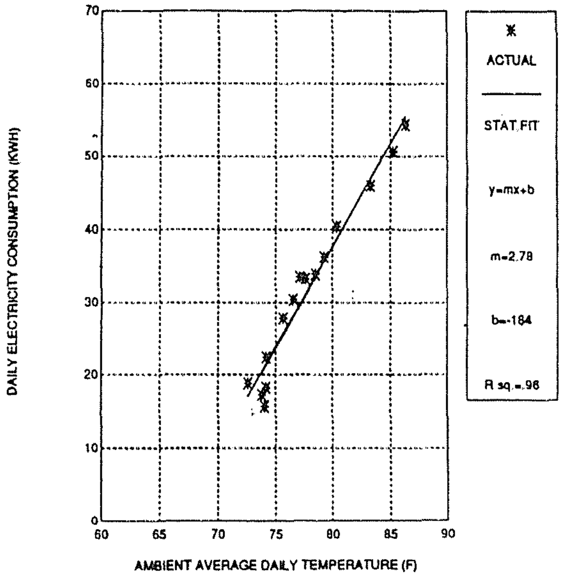

Figure 1 *Daily average condensing unit power consumption for test 1. Test 1 was conventional operation with no zoning, thermostat setup, or blower modulation.*

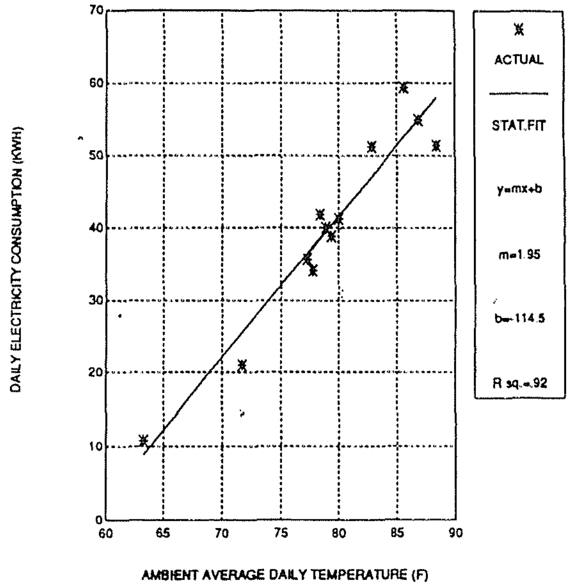

Figure 2 *Daily average condensing unit power consumption for test 2. Test 2 had zoning, blower modulation, and no thermostat setup.*

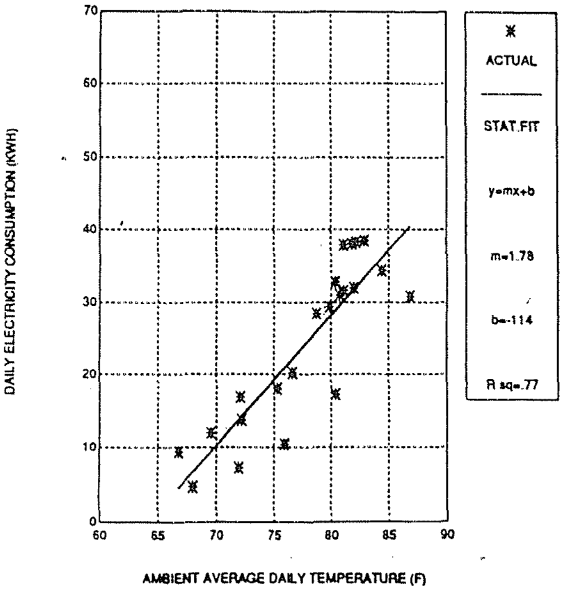

Figure 3 *Daily average condensing unit power consumption for test 3. Test 3 had zoning, blower modulation, and thermostat setup.*

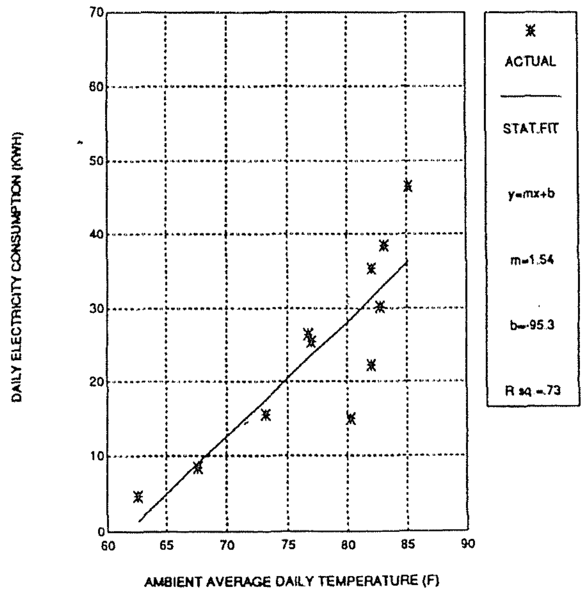

Figure 4 *Daily average condensing unit power consumption for test 4. Test 4 had zoning, blower modulation without overrun, and thermostat setup.*

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.

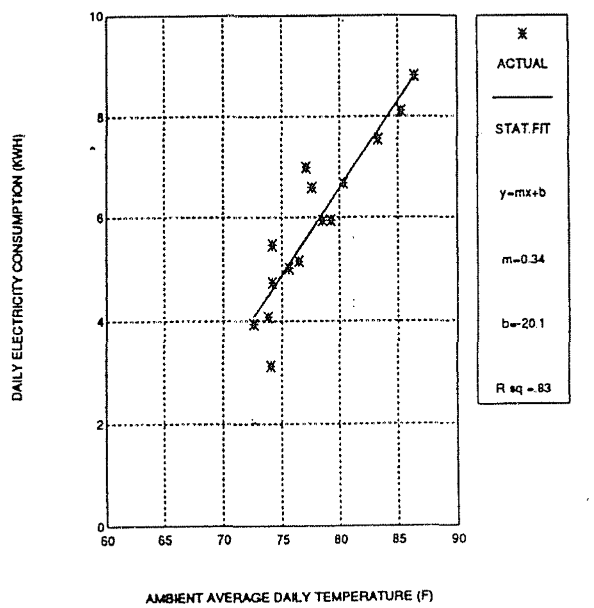

Figure 5 *Daily average blower power consumption for test 1*

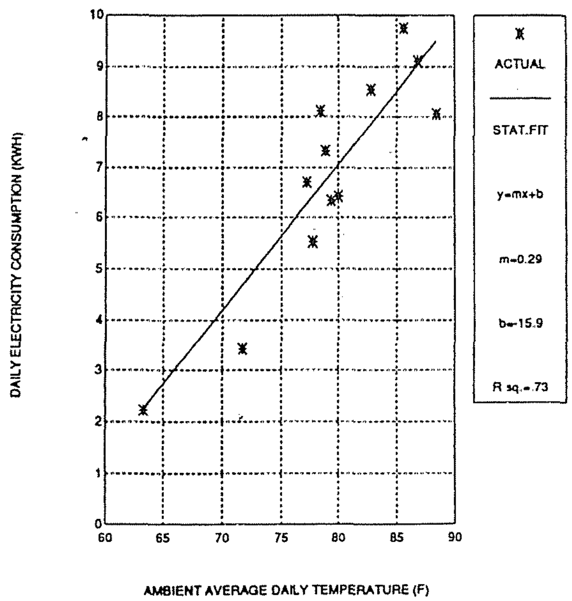

Figure 6 *Daily average blower power consumption for test 2*

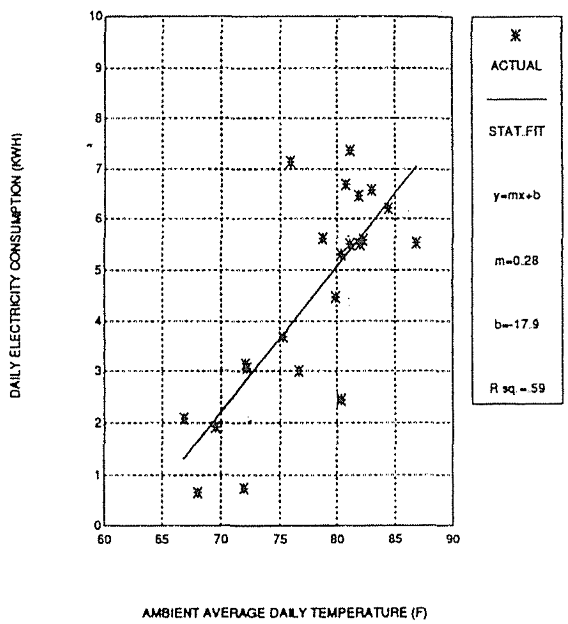

Figure 7 *Daily average blower power consumption for test 3*

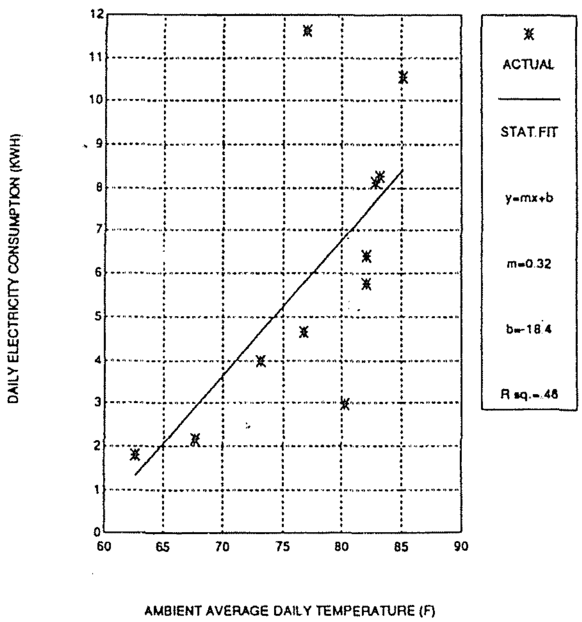

Figure 8 *Daily average blower power consumption for test 4*

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.

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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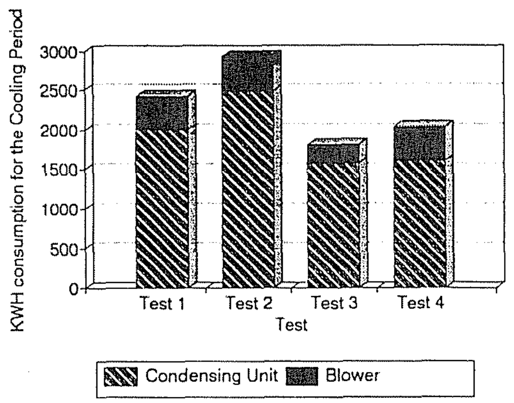

Figure 9 *kWh consumption for cooling season created with historical data and regression lines generated for condensing unit and blower electrical power consumption*

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FINAL REPORT

Field Investigation of Carrier Residential Zoning System

Prepared for

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by

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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

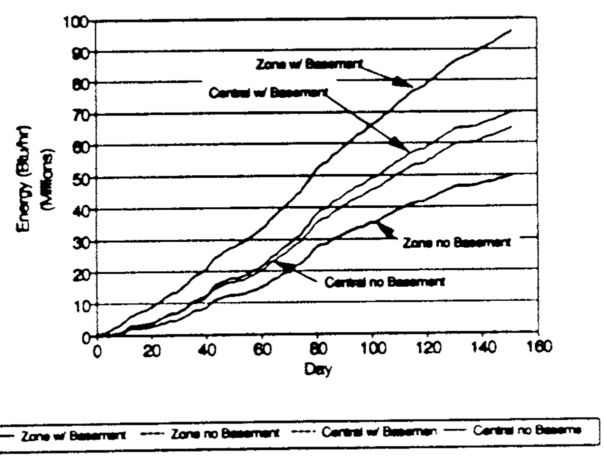
2.1 1993 Heating Season and Record Year Climate

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

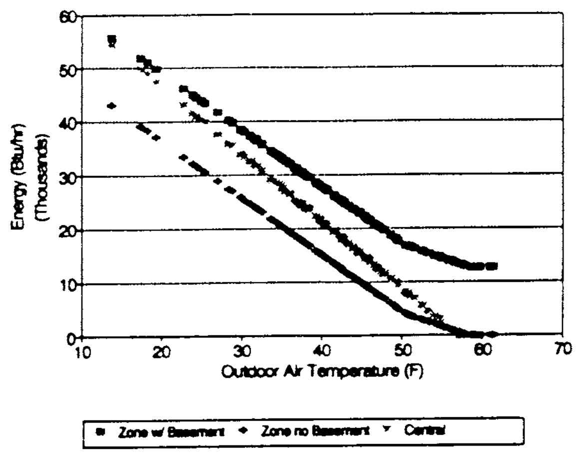


Figure 2.2.3
Zone System
Zone Heating Load

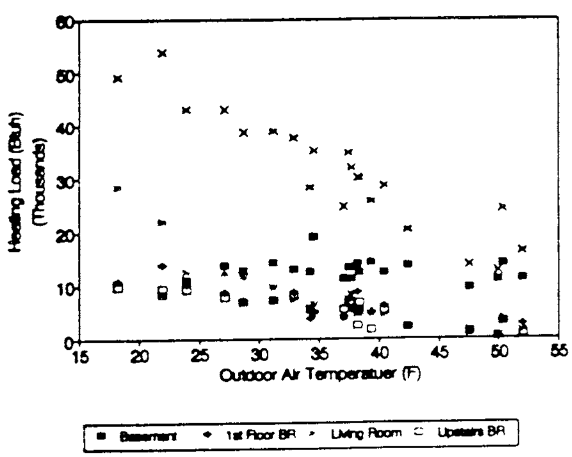
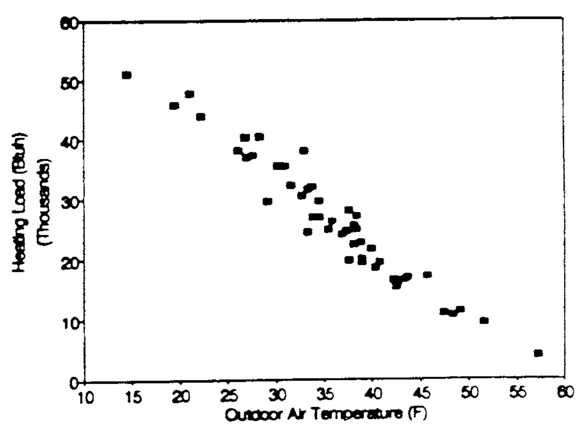


Figure 2.2.4
Central System
Whole House Heating Load



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

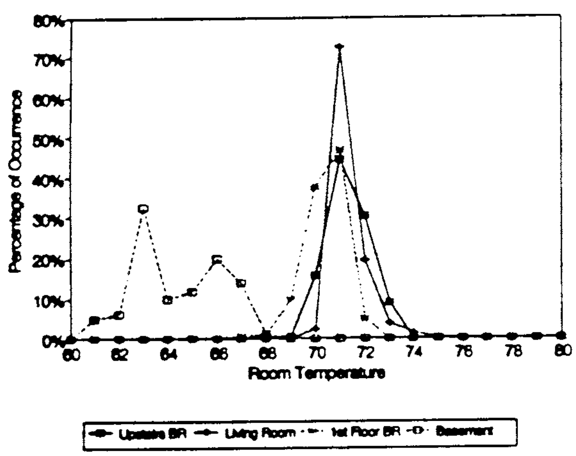
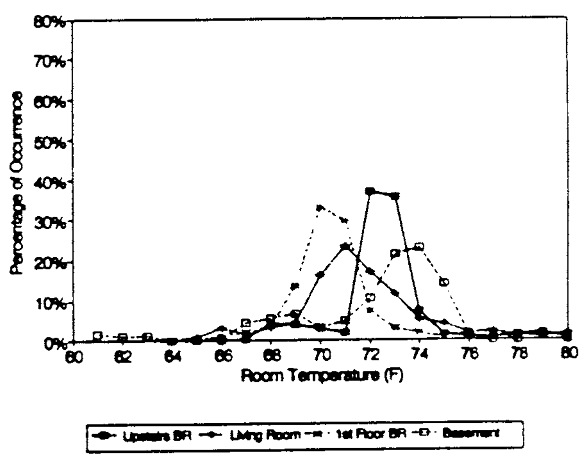


Figure 2.3.2
Zone System
Room Temperature Occurrences



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

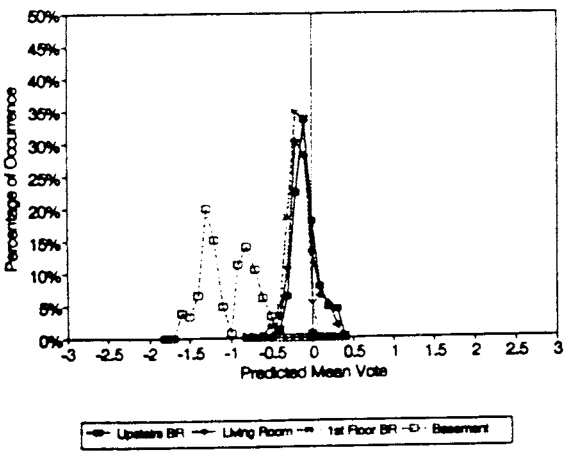
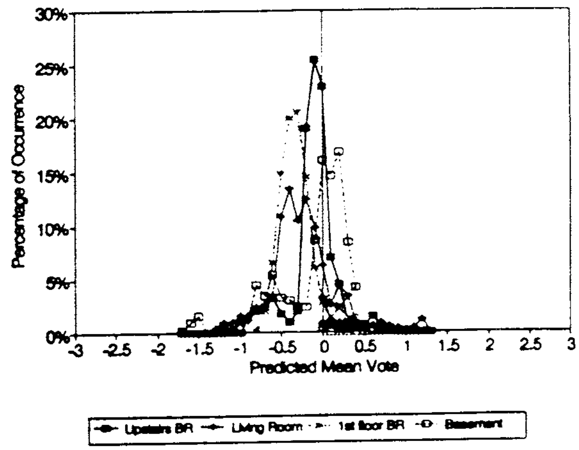


Figure 2.4.2
PMV - Zone System



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

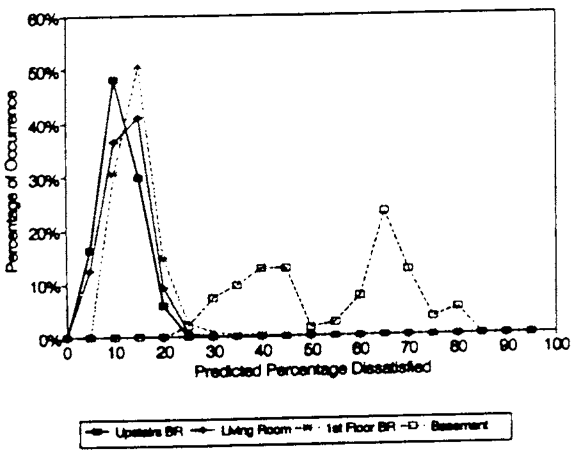
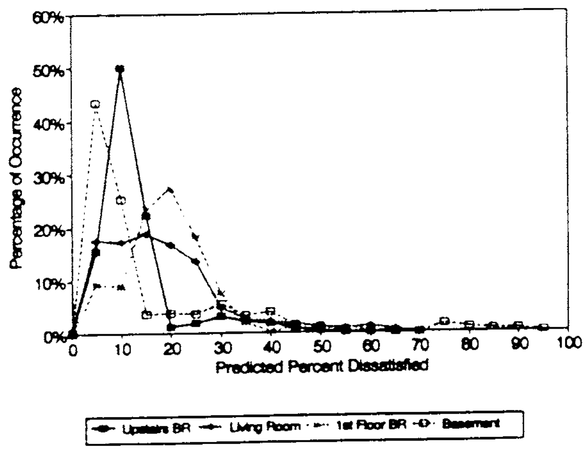


Figure 2.4.4
PPD - Zone System



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

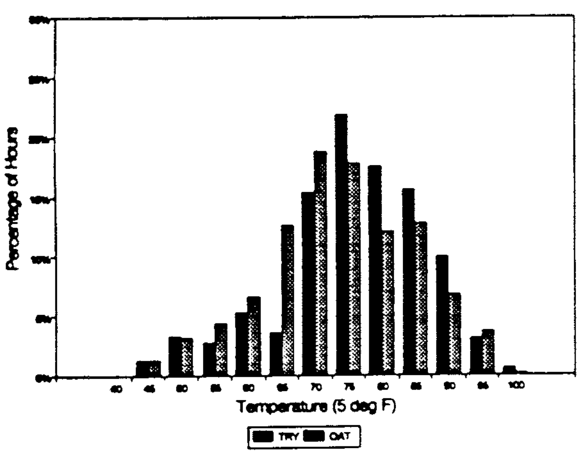


Figure 3.1.2
Outside Air Temperature vs. Typical Record Year
Zone Test

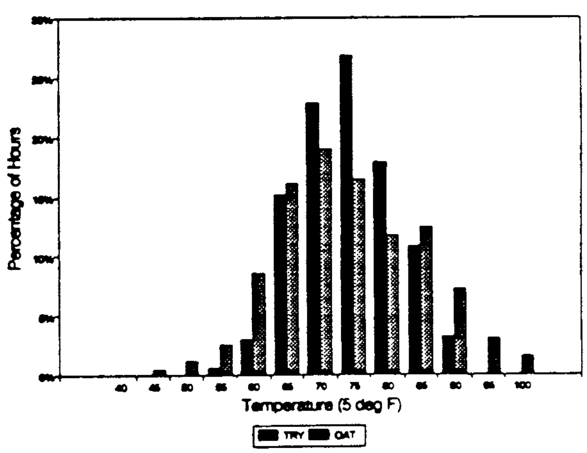
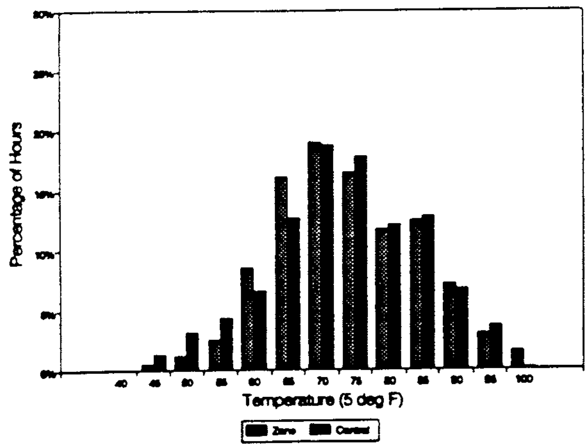


Figure 3.1.3
Outside Air Temperature
Zone vs Central



3.2 Cooling System Operating Performance

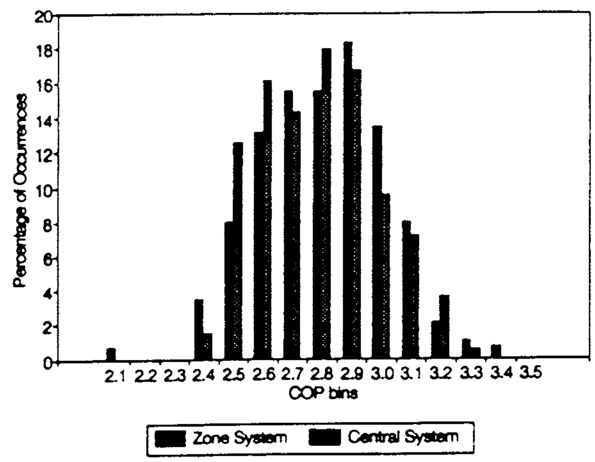
Cooling Equipment Description

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

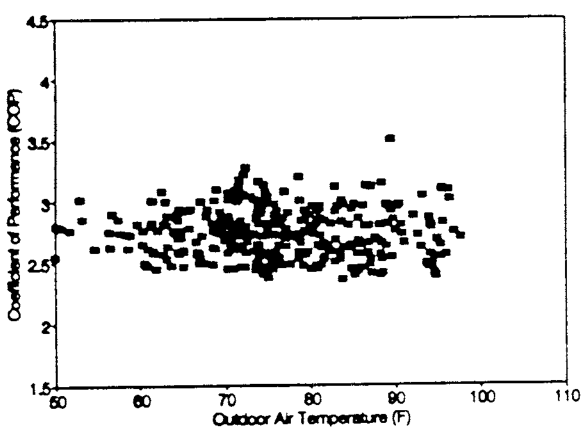
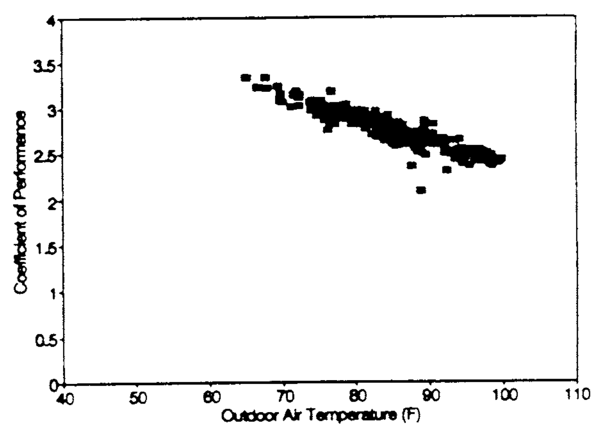


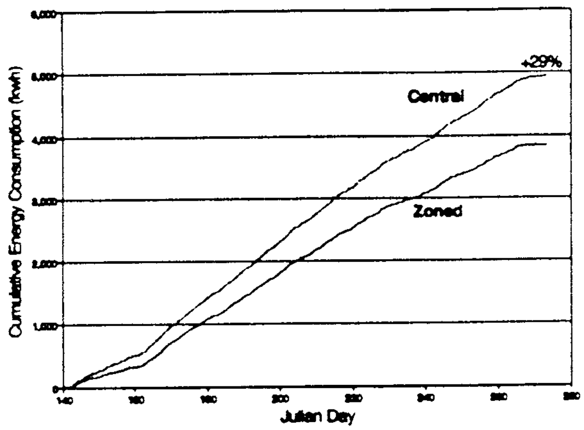
Figure 3.2.3
Zone COP vs. Outside Air Temperature



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.

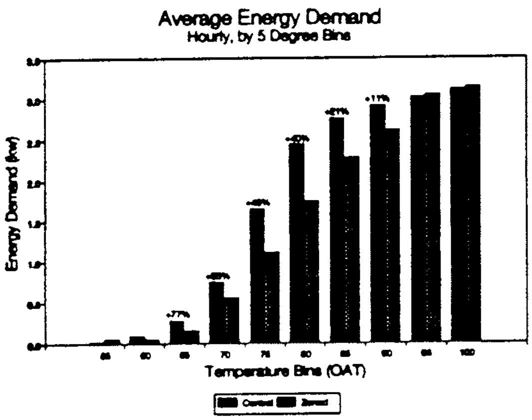
Figure 3.3.1
Cumulative Energy Consumption



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

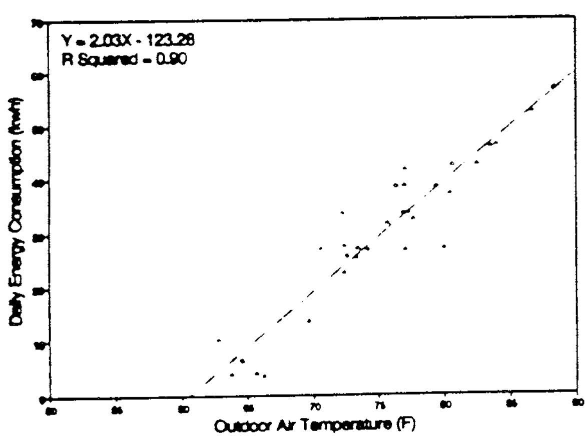
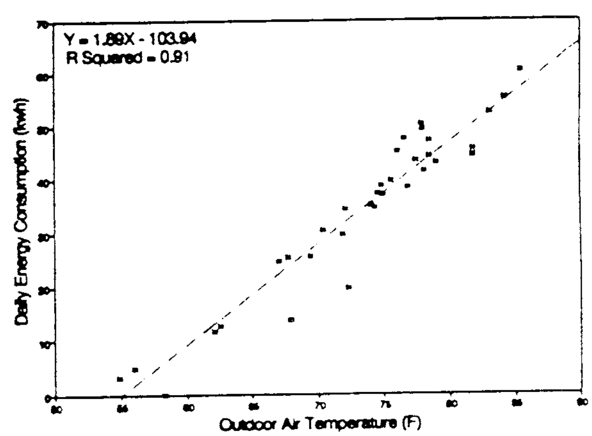


Figure 3.3.4
Central Daily Energy Consumption



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

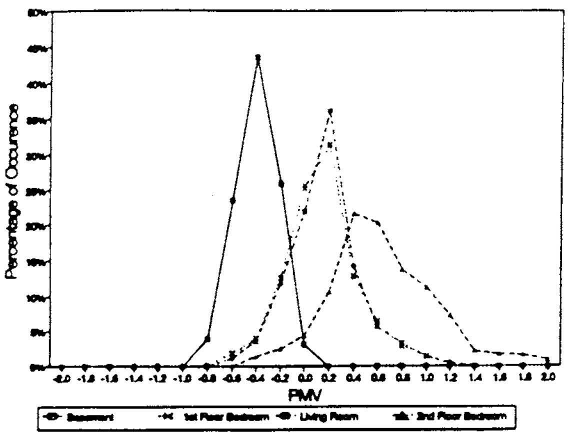
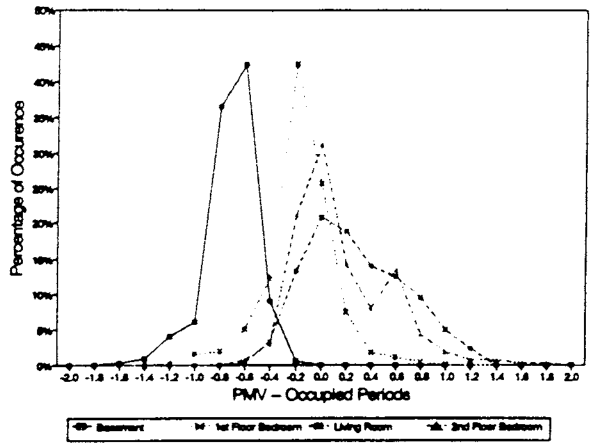


Figure 3.4.2
Zone PMV Distribution



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

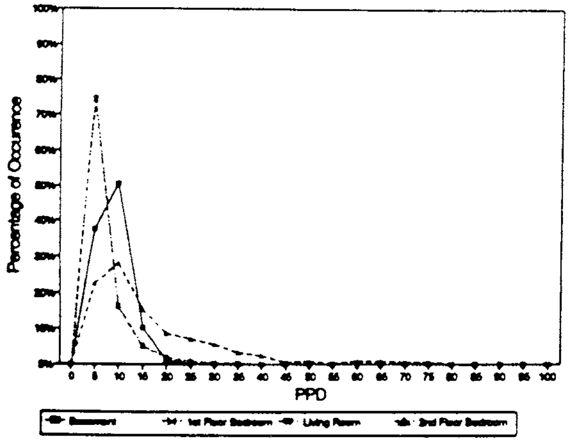
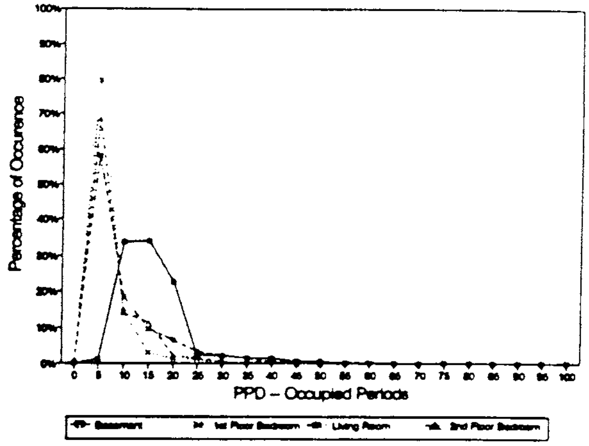


Figure 3.4.4
Zone PPD Distribution



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

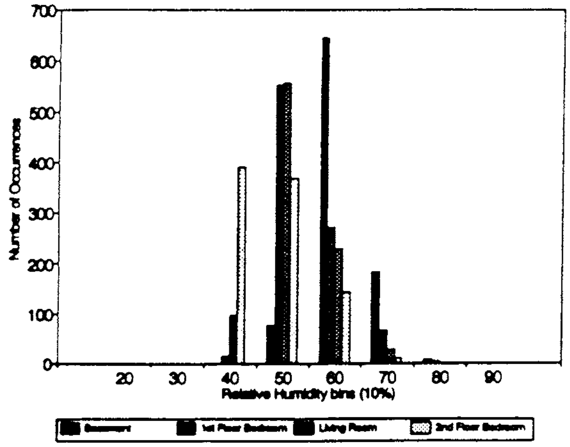
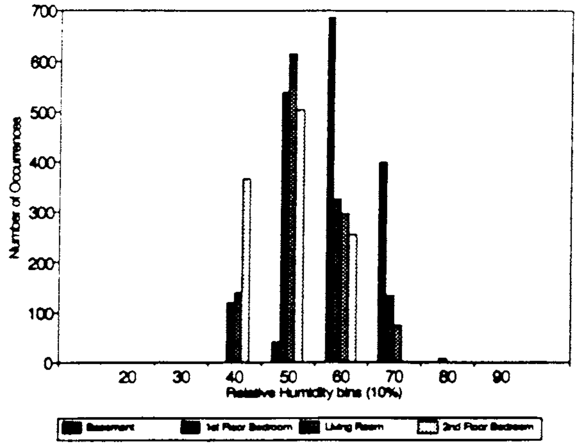


Figure 3.5.2
Zone Indoor Humidity Occurrences



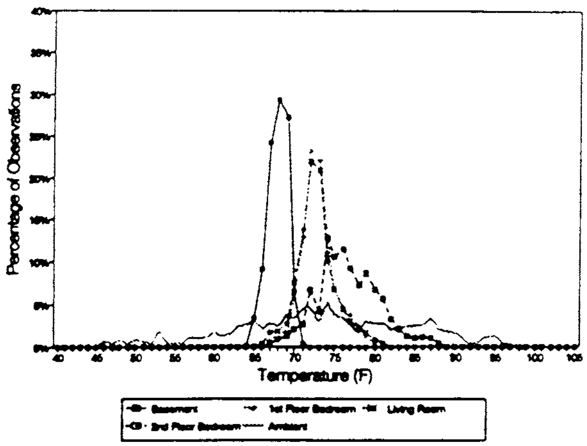
3.6 Frequency of Room Temperature Occurrences

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.

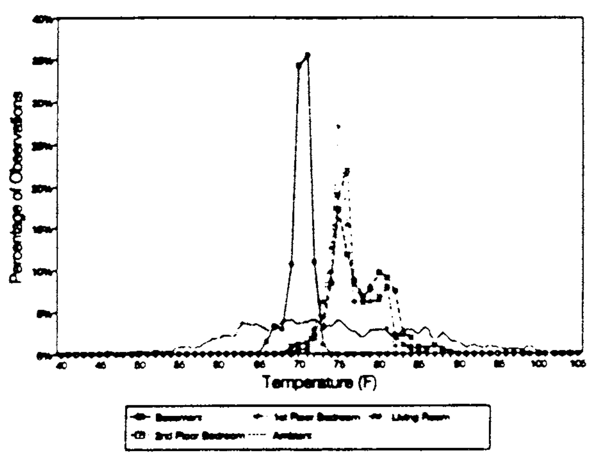
The "stack-effect" contributed excess heat to the second story, and the temperature occurrences for the upstairs bedroom vary widely, with the distribution showing temperature above the setpoint in the living room. The maximum number of observations for the upstairs bedroom occurred at 74°F.

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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Sherman, M.H., *Air Change Rate and Airtightness in Buildings*, American Society for Testing and Materials, Chelsea, Michigan, 1990, pp. 5-20.

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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)

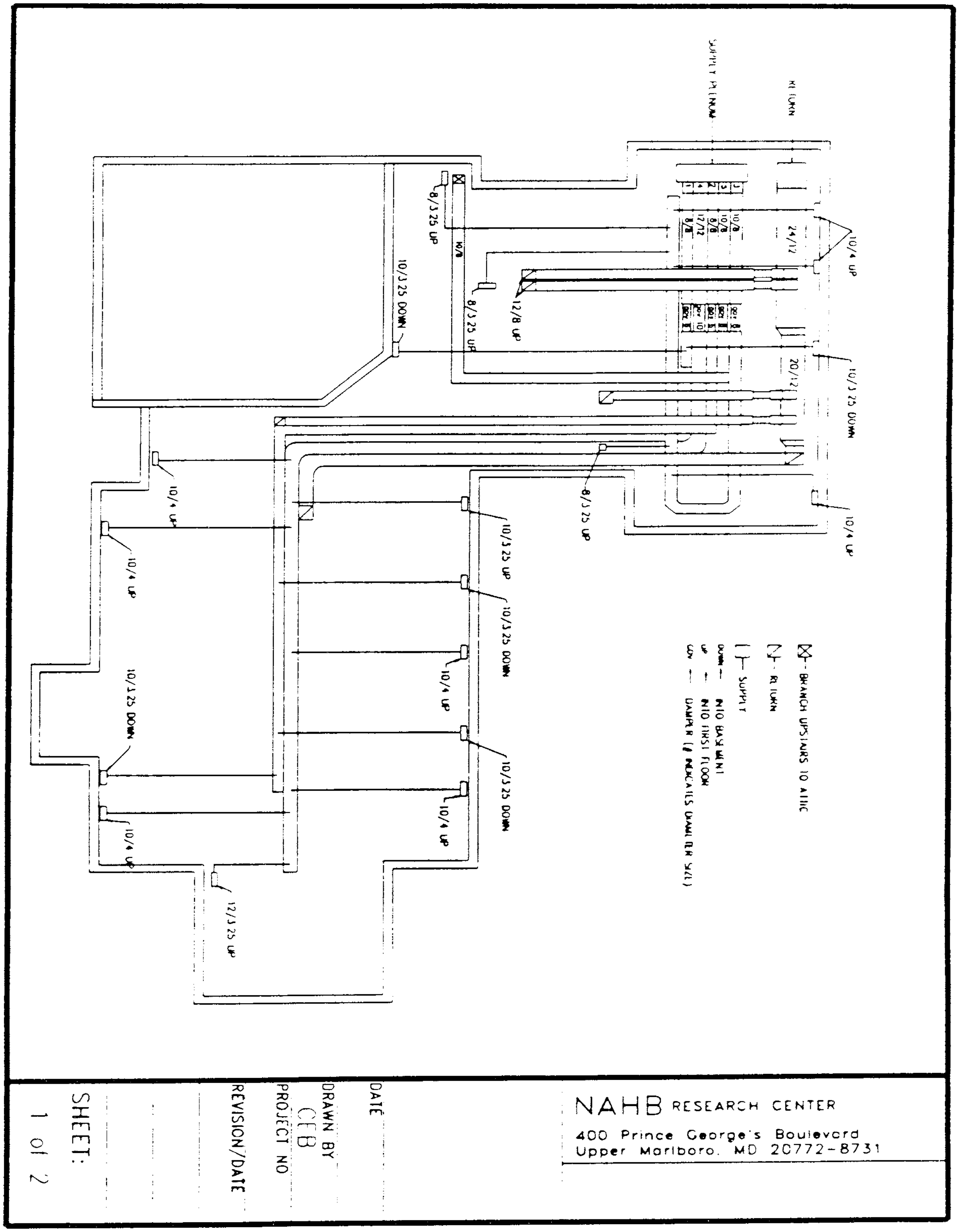
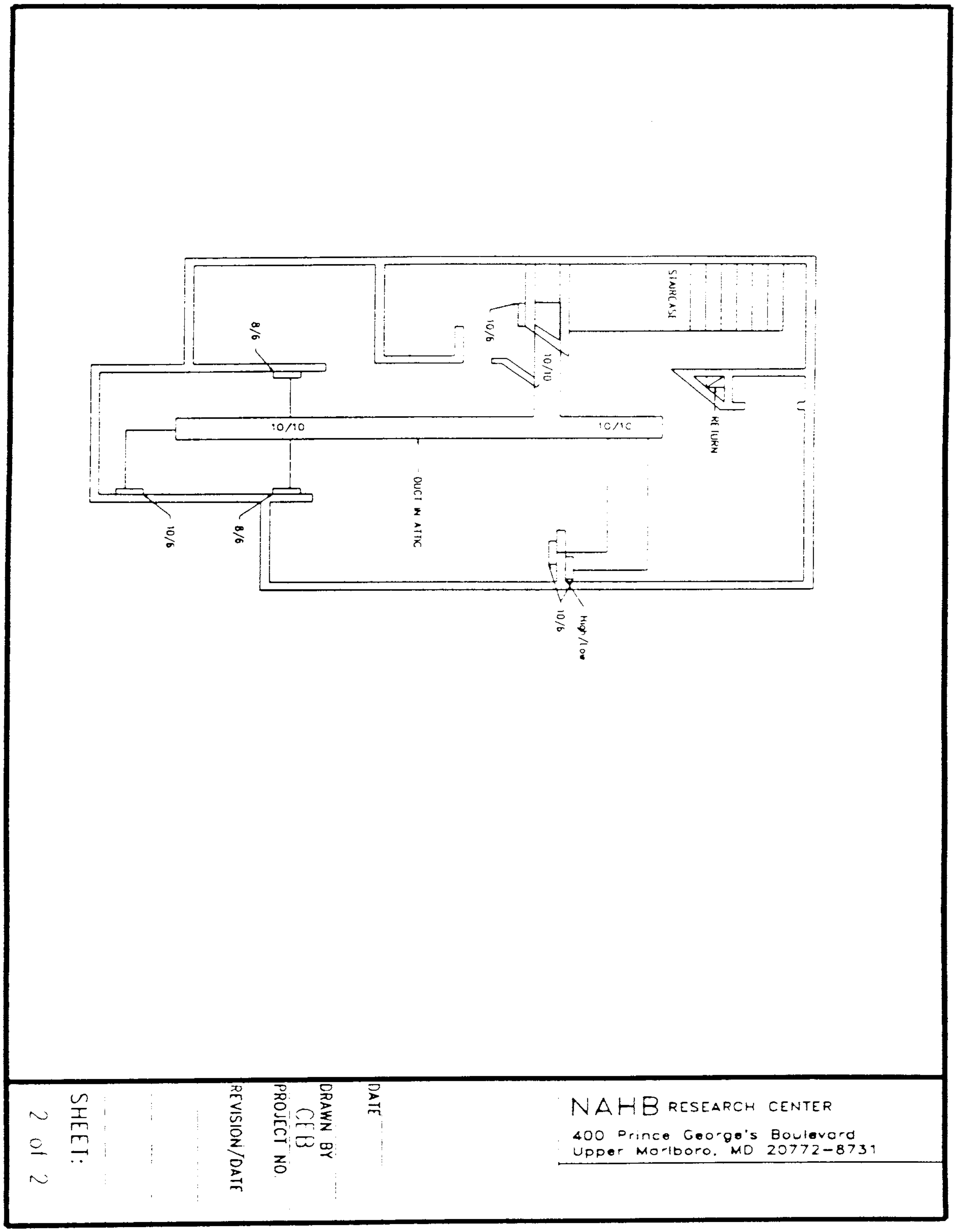


Figure A2
Second-Floor Air Distribution System and Floor Plan
(copy of engineer's drawing)



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 RH2 RH3 RH4 RH5	One per conditioned zone
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 3-Probe Grid	STX06 STX07 STX08	
Main Supply Dewpoint	SDX01	
Static Pressure	SPX01	Supply vs. return
	SPX02	Indoor vs. outdoor

Indoor Measurement Parameters
(continued)

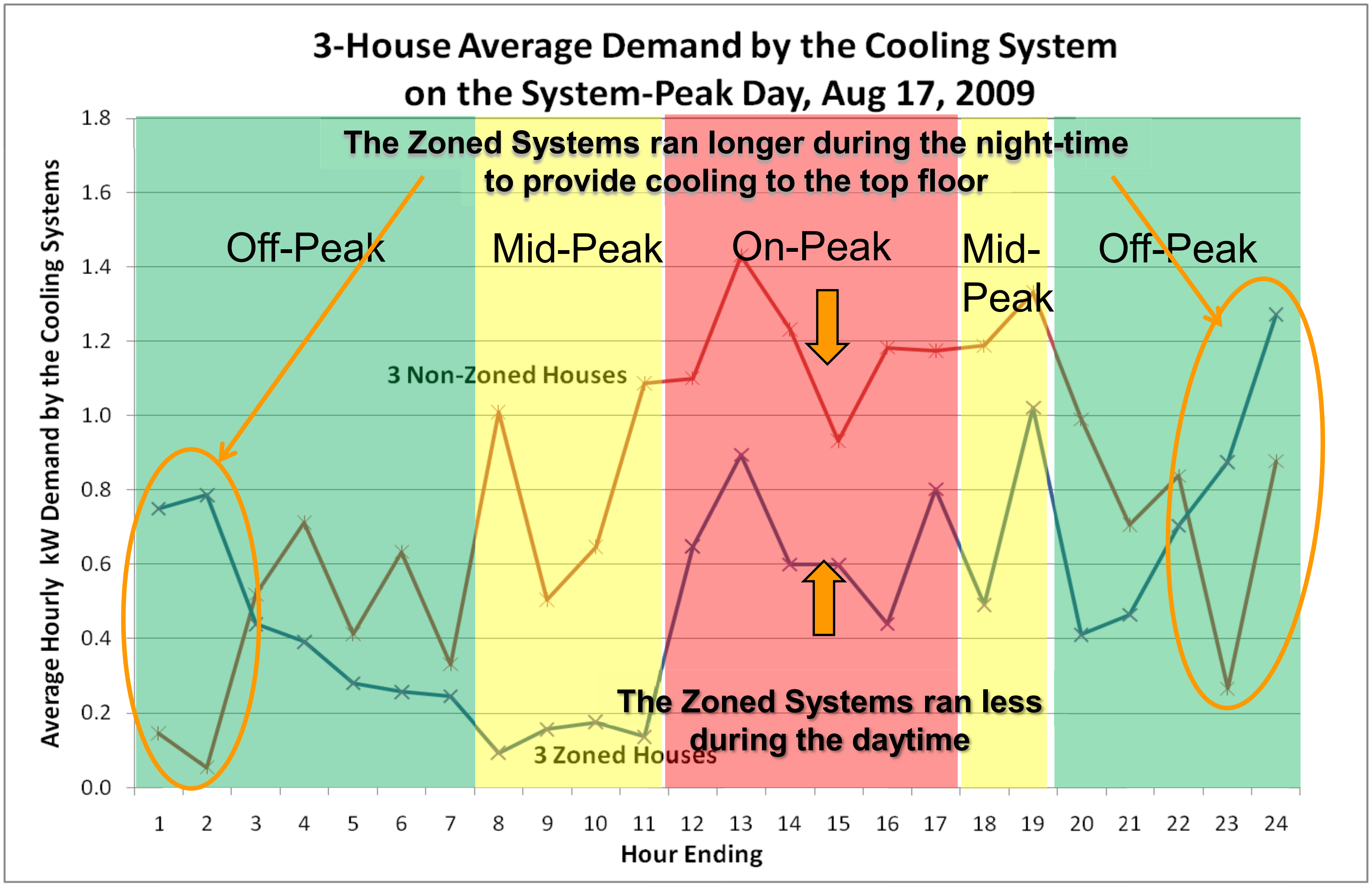
Parameter	Code	Site
Supply Air Velocity	SV1	One supply branch per
	SV2	conditioned zone
	SV3	
	SV4	
	SV5	
Main Return Temperature	XT	
Main Return Dewpoint	XD	
Main Return Velocity	XV	
Supply Register	SR101	South wall
Temperature	SR102	Near P1
	SR203	South wall between doors
	SR204	South wall east of 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

Peak Electricity Usage

Avg. Conv.	19.3 kWh
Avg. Zoned	12.9 kWh



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These comments reflect the views of AHRI and the member companies of the AHRI Zone Control Systems Technology Section. The slide references are to the slides presented at the 2013 Building Efficiency Standards and Residential Zoned A/C Workshop held on July 15, 2011.

Slide 5 – Typical Practice – Two Types of Zonal Systems
Multiple Systems, High Performance as compared to what? The author's previous study showed a number of homes with single systems had lower than expected efficiencies and higher initial cost not only for the equipment but also higher operating costs when both A/C compressors are running. Each furnace, air conditioner and heat pump requires a certain amount of power that must be taken into account in order to calculate the home's electrical load. This increases the load for each home and increases the electric demand on the utility. Homes with multiple systems that can be combined into one unit and zoned with dampers can reduce the utility's demand.

Single speed compressors and fans cannot modulate to track load. Currently with the majority of the installed systems this is true. However, that is why zoning is used to condition the zones inside the home as the load changes in different areas of the home.

Supply air flow is low when all zones are calling. This statement is misleading in that the volume of air (CFM) through the HVAC system is not reduced when all zones are calling. The air velocity and volume delivered to the registers may be slightly lower with all dampers open versus when only one zone is open.

By-Pass ducts are common and are used to control the static pressure and velocity in the duct system as zone dampers open and close, while maintaining a constant volume of air moving through the HVAC Unit.

Slide 6 – Code Change Proposals
Eliminate bypass ducts – The manufacturers of Zone Control Systems who have sold millions of systems for over 50 years cannot all be wrong. By-Pass ducts serve a purpose to maintain air flow and pressure in a duct system.

Delete the current Zonal A/C performance compliance credit – This will result in higher energy costs, resulting in continued poor comfort conditions and homeowners over compensating on thermostats, in order to maintain the comfort level in areas without a thermostat.

Slide 10 – Typical Dampered Multi-Zone A/C System with By-Pass Duct
While this may have been the case in many of the homes in the case study, it is not the recommended method. We believe the case study homes have flaws that affect the operation of the system and contributed to the negative effects of the case studied homes.

Slide 11 – How Zoning with Bypass works
If in actuality these systems were not performing properly, the study results were adversely affected. Since these systems are stated to have low airflow when both zones are calling, may indicate a problem existed before the zone system was installed, such as over-use of high resistance flex-duct and/or excessive duct leakage.

Slide 12 – Bypass Duct Flex from Supply to Return
This slide is indicative of extremely poor workmanship and rampant over-use of high resistance flex-duct. This HVAC system will consume more energy whether or not it is zoned.

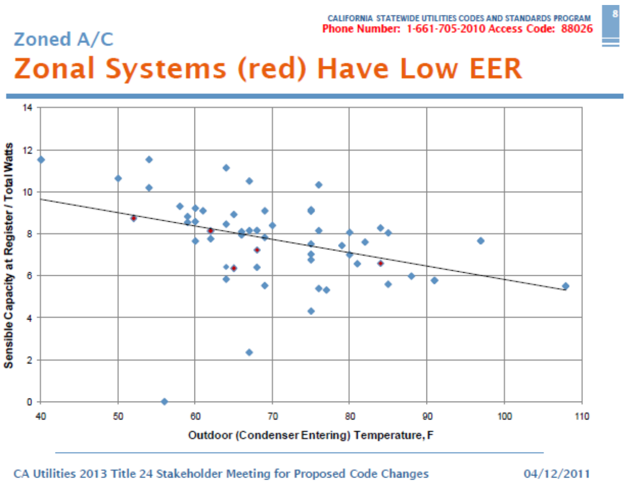
Flex-duct is arguably the single most likely cause of high duct pressures and poor air delivery to the occupied space. The CEC can make a much larger impact on energy savings by limiting flex-duct to the last 6 ft. of branch runs and prohibiting the use of flex-duct on main duct runs and bypass runs.

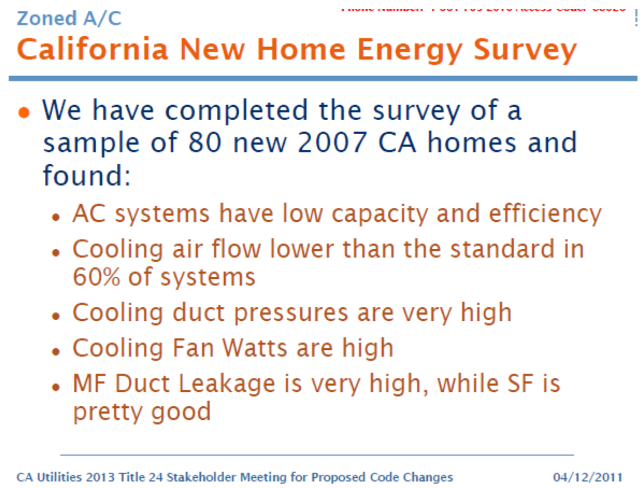
Slide 14 – AHRI – Manufacturers
AHRI will argue that this study's conclusions do not look at the overall energy consumption of the home or how the system is operated. The presenters cite prior studies only to support their positions. The presenters completely ignore the same study's conclusions that zoning can save over 20% when zones are setback. If the goal of the CEC is to provide common sense energy reduction solutions, zoning with setback thermostats provides that ability automatically and not just in some cases, but in all cases.

Slide 15 – Research on Multi-Zoned Systems
These separate research projects both came to the same conclusions, zoning can add 20% (not the 35% as noted on this slide*) to energy costs if no setback is used and can provide 25% savings when setback is used. Attached is another chart from a more recent study on zoning showing a 30% reduction in cooling KWH with zoning. The presenters continue to report only on the increase in energy and not on the savings.

> *The Oppenheim Study from 1991 must have been misquoted in the NAHB/Carrier Study as no place in the 1991 Study does it note a 35% increase. Only a 20% increase is noted. A full copy of both studies is attached.

Slide 23 – Average Energy Impact
The presenter's presentation from April 12, 2011, shows the total number of homes surveyed with lower than acceptable EER ratings. While only two zoning systems were substantially below the acceptable line, 16 non-zoned systems fell at or below the lowest rated zoned systems. Our point is that there are many reasons for systems not to be performing in the field at their rated efficiency levels. Zoning should not be singled out because of poor installations. The efficiency of each of the underperforming zoning systems can be improved by correcting improper installation techniques. We maintain that the presenters are unfairly critical of zoning. Considering that this study also has a substantial percentage of non-zoned systems, 20% whose efficiencies fall below the acceptable line. Slide 6 from the April 12, 2011 presentation states that 60% of the 80 homes surveyed also had lower than standard cooling air flow.





If issues exist with 60% of the systems, and zoning is less than 10% of the systems and only two zoning systems are substantially below the average, common sense dictates that these are not properly performing systems to be used as a standard for gauging performance.

Slide 25 – No Bypass and No Extra Cost – Bonus Supplies
This proposed scheme where the "Bonus Supplies" are damper controlled while the main ducts to the zone have no control at all will result in over-shooting the thermostat. There is minimal temperature control and this will only result in over-shooting thermostat set-points in those zones, causing homeowner discomfort. The presenters should review zoning manufacturers' guidance.

Slide 26 – Damper Stop Relief
This can certainly be a supplement to a by-pass but not a cure all. The damper stop adjustment may be at a point where too much air enters a zone, and will only result in over-shooting thermostat set-points in those zones, causing homeowner discomfort.

Slide 27 – Another Answer
The alternatives to Zone Dampers mentioned are multiple units or mini-splits. This logic makes **no sense** when it comes to energy efficiency. This suggests adding a second or even third unit to a home. Adding units will increase the utilities demand load to provide added electrical capacity to the home by two or three times as using one unit with zoning. Instead of having one 30 Amp circuit and one HVAC Unit, the alternative is to add two or three – 30 Amp circuits. This makes absolutely no sense as utilities are looking to decrease their load requirement. Adding extra air conditioning units only increases generation capacity requirements for utilities.

Slide 28 – Variable Capacity
We concur that variable capacity is a great option but not an alternative to zoning. We believe variable speed systems should be zoned in order to achieve maximum energy efficiency. Zoning will match the capacity of the HVAC system to the zone load. This is where the HVAC Industry is heading. New federal

energy regulations will be in place and manufactured HVAC systems that can meet these new regulations will be the majority of the market by the time these new proposed CEC Regulations take effect. Why not have a regulation in place that anticipates the market?

Slide 29 – Conclusions

Bypass should be eliminated because they intrinsically reduce energy efficiency is not valid as in the NAHB/Carrier study, a by-pass was used and over all energy savings was achieved using setback control.

Multi-Zone Systems are for comfort, not energy savings, is stated only because of the potential for higher energy cost based on misuse of the system or poor workmanship.

This whole study ignores the stated energy savings when zoning is installed with setback control and the ironic part is that setback thermostats are mandated. The CEC should mandate setback thermostats along with zoning and significant energy savings will occur. Why are the CEC presenters ignoring this glaring answer for an extremely viable low cost option to save energy?

Slide 30 - Code Change Proposals

Zoning should remain as part of Energy Code as the occupants have the ability to set back rooms/zones of the home. Just as the CEC presumably would not ban the use of a light switch for each room and only require one light switch for the whole house, the CEC should not ban the use of a thermostat for each zone. Zoning is for comfort and energy savings. The studies have proven so with the use of setback. Also, people who are comfortable are less likely to change the thermostat settings than those who are uncomfortable.

AHRI Comments
on
Proposed Changes
CEC Title 24
Residential Zoned AC

California Statewide Utility Codes and Standards Program
04/12/2011 1 Residential Zoned AC Stakeholder Meeting #2
Bruce Wilcox April 12, 2011

AHRI Version 1.00

June 6, 2011

Response to Proposed Changes

CA Utilities 2013 Title 24 Stakeholder Meeting for Proposed Code Changes

New Mandatory Requirements
No Bypass Ducts, Air Flow and Fan W

SUBCHAPTER 7 LOW-RISE RESIDENTIAL BUILDINGS – MANDATORY FEATURES AND DEVICES

SECTION 150 – MANDATORY FEATURES AND DEVICES
Any new construction in a low-rise residential building shall meet the requirements of this Section.

….

(m) **Air-distribution System Ducts, Plenums, and Fans.**

…

11. Cooling System Bypass Duct. Bypass ducts which deliver supply air to the return duct side of the system shall not be used.

12. Zonally Controlled Central Forced Air Cooling Systems. Central forced air cooling system fans shall simultaneously demonstrate, in every zonal control mode, an airflow greater than 350 CFM/ton of nominal cooling capacity and a fan watt draw less than 0.58 W/CFM as specified in Reference Residential Appendix RA3.

Cooling System Bypass Duct

Bypass ducts should not be banned. They are an effective air management tool, and the net affect on seasonal KWH is not known (for one simple system, or for a state-wide set of systems).

- The ban seems to be based on energy concerns.
- The sensible Btuh of supply air per KW of equipment power ratio does not relate to seasonal energy use.
- The rational for the ban is flawed.
- See Summary Item 4, and supporting detail

Low Limit for Blower Cfm

A low limit of 350 blower Cfm per cooling Ton for any zonal mode is consistent with the performance range for most cooling equipment, and is compatible with air relief strategies. In this regard, cooling Ton needs to be defined.

- There is a Cfm/Ton value for the AHRI rating test, for a given piece of equipment.
- These is an applied Cfm/Ton value for the summer design condition, for a given piece of equipment operating at a particular location.
- These is an applied Cfm/Ton value for any momentary operating condition, for a given piece of equipment operating at a particular location.

High Limit for Blower Watts

It may be that a high limit of 0.58 Watts per Cfm is consistent with the performance range for most cooling equipment and approved duct systems.

- OEM blower tables always provide Cfm vs. IWC values, but may not provide the corresponding Watt values.
- External static pressures may range from 0.10 to about 1.0; and airflow may range from about 600 Cfm to more than 2,000 Cfm.
- External resistance depends on external device and component resistance, and on duct run resistance (must be compatible with blower pressure).

Summary

Comments for Proposed Changes in CEC Code

This is a response to a set of CEC slides that cryptically summerize complex issues and concepts. Comments are made without knowledge of previous discussions and debates, and without knowledge of the documentation and supporting detail that justifies the suggestions and conclusions that appear on the slides.

Item 1 -- System Physics vs. Human Nature

What are the quality control conditions for comparing system performance and efficiency?

- For any installation there is a maximum system efficiency (no design or installation flaws); and a degraded efficiency (some combination of design and installation flaws).
- Quality assurance mandates should be based on observed knowledge of common design flaws and unapproved practices.
- System merit (comfort and efficiency) comparisons, and subsequent rule making, should be based on the *no design or installation flaws* scenario. In this regard, there may be more rules for air zoning vs. single zone.

Item 2 -- Slides Limited to Basic Zoned System

The slides appear to focus on a simple two-zone system that has (roughly) equal design Cfm values for zone supply air; and single speed equipment (one compressor speed, and one blower speed). This minimizes the design value for excess air (blower Cfm minus the smallest design Cfm value for the two zones).

- Two large zones minimum excess air, so this is the best scenario for simple single-speed equipment.
- Two large zones (minimum excess air) require less air relief measures, which may be some combination of bypass air, a dump zone, damper stops, and zone over blow.
- The slides only mention bypass air and over blow, and propose to ban bypass air.
- The Slide 21 solution (Figure 1) shows a duct system that has an undampered supply to each zone, and a dampered supply to each zone. There is no simple way to predict the behavior of this design.

  A zone may not need supply air, but some undetermined amount of supply air will be delivered to the zone (through the undamperd supply) when the zone damper is closed.

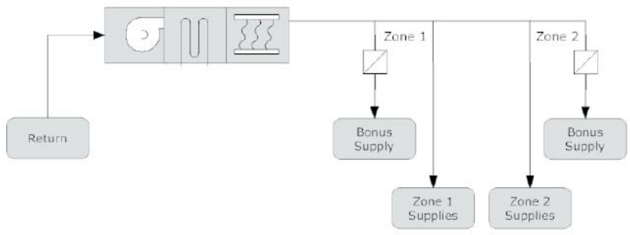


Figure 1

The slide says that 1/.3 of the air will flow to the zone that does not need the air (three ducts open, one duct closed, each duct at 1/3 of the blower Cfm).

How is this going to happen? What are the duct sizing rules (for a particular blower table and set of pressure-dissipating devices, for a particular duct run geometry, and for a particular set of duct fittings); and/or what are the air balancing rules (the slide does not show hand dampers)?

If a zone thermostat is satisfied, and if 1/3 of the air continues to flow to this zone, how long will it take for zone temperature to drop to an unacceptable level; and how does this transient compare to the time it will take ro satisfy the calling zone thermostat?

Air outlet performance (throw and noise) depends on make, model and size. Some guidance is need here.

In general, how can practitioners be sure that the proposed (Figure 1) design will always deliver adequate performance for any operating condition?

Item 3 - Useful Measures and Features Ignored

The slides say nothing about dump zones, damper stops, selective throttling, and variations of a slave zone. The slides mention capacity control, with no subsequent comment or guidance.

- Zone systems are installed for comfort, and comfort depends on providing an adequate number of zones (as determined by architectural and construction attributes, and space use).
- A single-speed design will require some combination of air relief measures. In some cases, the number of zones may be less than desired.
- An up-scale design will use staged equipment (compressor and blower), or variable-speed

equipment (compressor and blower), and some combination of air relief measures. In this regard, capacity control reduces the amount of excess air, and allows more comfort zones.

- Some OEM's provide a complete system (central components and zoning components) that requires bypass air, and may have capacity control.
- Some OEM's provide a complete system (central components and zoning components) that has capacity control with no bypass air (selective throttling systems).
- Zone damper equipment vendors have various methods for adding zoning to a piece of central equipment. This includes all types of air relief measures, including bypass air.

Item 4 - Bypass Duct Banned

What is the rational for prohibiting bypass air? If this is an energy issue, see the Item 5 comments.

Bypass Utility

A properly designed zone damper system may require a set of air relief measures. Eliminating the bypass option will, in general, have an adverse effect on system performance.

- Comfort is the primary reason for zoning. In this regard, the goal is to provide for precise temperature control for all rooms and spaces.

  Bypass air has no adverse effect on zone temperature control (zone Cfm matches zone load).

  Distributed relief (damper stops) and zone over blow conditionally diminish zone temperature control (zone supply air Cfm may not match zone load).

  A dump zone may be conditionally uncomfortable.

- Preferred temperature control may require three or more zones, but the air relief requirement increases with the number of zones.

  The critical zone is the zone that has the smallest design Cfm value.

  The number of zones determines critical zone Cfm.

  Eliminating the bypass air option may force a comfort-compromised zoning plan (zone consolidation).

- Two stage equipment (compressor and blower) reduces the air relief requirement, but does not automatically eliminate the need for a bypass duct.

Bypass Design and Operation

The purpose of the bypass duct is to assure adequate flow though the blower as zone dampers close. Momentary performance (bypass Cfm, cooling coil temperature, and system EER) depend on a set of environmental variables that apply to all system designs; and on a set of variables that apply to a particular system design.

- The low limit for bypass Cfm depends on a lot of issues (primarily on the OEM's blower data, the momentary Btuh of sensible cooling capacity per Cfm of coil air flow, and the OEM's low limit for discharge air temperature).
- The low limit for bypass air may be less than 10% of the blower Cfm, or more than 50% of the blower Cfm, depending on momentary circumstances; and on how the air zoning system is designed and controlled.

  Approved designs maximize momentary bypass effectiveness without causing central equipment problems.

  A bypass duct may be (necessarily) supplemented by some combination of damper stops, bypass duct, dump zone, and zone over blow.

  Bypass effectiveness also depends on appropriate airflow and temperature sensors, controls, and control strategy.

  There still is a lower limit to the size of the critical zone (as far as its design Cfm is concerned).

- A properly designed bypass system, will not, for worst-case conditions (critical zone damper open, all others closed), operate at less than the OEM's lower limit value, which may be 350 Cfm/Ton, or significantly more than 350 Cfm/Ton.

  Zone damper system controls should shut the system down (in a normal manner) before the OEM limit temperature control acts.

  Routine shut downs will not occur if the system is designed and installed correctly.

No Bypass Method

Selective throttling systems tend to be (or are) proprietary OEM packages. This design uses software or firmware code to control compressor capacity, blower Cfm, and zone over blow; and uses a comfort zone as a dump zone.

- These systems do not require a bypass duct, per proprietary design rules.
- The OEM's design rules may limit the number of zones (a comfort compromise).
- There may be proprietary duct layout and airway sizing rules.
- Using a comfort zone as a dump zone conditionally reduces zone temperature control (zone Cfm may not match zone load).

- Zone overblow should be reconciled with supply air outlet performance (re: air mixing and noise).

Item 5 -- Net Sensible EER

Isn't EER defined as momentary cooling equipment output for total (sensible plus latent) cooling Btuh divided by momentary power input in watts? Showing EER as supply air outlet output for sensible Btuh divided by the total electric energy input watts is technical sophistry.

- The reason for air zoning is improved temperature control for rooms and spaces. This is accomplished by reducing supply air Cfm to the zoned space.

  Sensible supply air Btuh per system input KW must decrease when zone dampers fulfill their mission.

  What else might we expect?
- What about latent capacity (some California climates produce latent loads).
- Using a sensible supply air Btuh to input KW ratio to compare zone damper systems with single zone constant volume systems is an apples and oranges exercise.

Momentary EER for Simple Air-Cooled Systems

Momentary EER equals the momentary equipment output capacity at the equipment (Btuh), divided by momentary power input to the equipment (KW).

- Momentary capacity equals the sum of the sensible and latent capacity, which depends on blower Cfm and cooling coil temperature.
- Momentary input power equals the sum of compressor power, outdoor fan power, and indoor blower power.

  Compressor power is affected by blower Cfm and cooling coil temperature.

  Outdoor fan power may be constant (not be affected by blower Cfm and cooling coil temperature).

  Indoor blower power depends on the type of blower motor (PSC or ECM), motor performance setting (speed tap or Cfm tap), and the amount of duct system resistance (produced by components, devices, straight runs, and fittings).

Startup Transient

System performance (and EER) continuously changes for some minutes after startup, then settles to a steady state condition. This affects single zone system and air zoned system efficiency a similar way (the consequences of possible differences in start-up Cfm are not argued here).

Momentary EER for a Simple Single Zone System

For one large zone, the thermostat set-point is maintained by on-off control. After startup, blower Cfm and blower power are constant (because duct system resistance is not a controlled variable), and cooling coil temperature is not affected by the action of a space thermostat.

- Cooling coil temperature and compressor power depend on outdoor temperature, and the condition of the entering air.
- Cooling coil temperature and compressor power depend on the momentary values for sensible and latent load.

Momentary EER for a Simple Zoned System

Thermostat set-points are maintained by adjusting zone damper position. After startup, blower Cfm and blower power depend on zone damper position, and bypass damper position (if applicable); and also depend on other air relief measures (see Item 3). For this design, cooling coil temperature is affected by the action of the zone thermostats, and bypass damper position (and other air relief measures).

The net affect on momentary EER is a complex issue. There are conditional tendencies, which may work in the same direction, or opposite directions.

- Blower motor power (watts) depends on blower speed, blower Cfm, airflow resistance within the cabinet, duct system resistance, blower efficiency, and blower motor efficiency.

  Blower Cfm tends to decrease, and system resistance tends to increase as zone dampers close.

  Blower Cfm tends to increase, and system resistance tends to decrease as a bypass damper opens.

  Air relief measures, in general, tend to stabilize blower Cfm and blower power.
- Refrigeration cycle efficiency and compressor power depend on cooling coil temperature.

  Cooling coil temperature tends to decrease as zone dampers close. This behavior is similar to a blower speed change (OEM performance data correlates sensible and latent capacity, and equipment KW, with blower Cfm).

  Cooling coil temperature rapidly decreases as the bypass damper opens (OEM performance data does not model this behavior).

  Other air relief measures have a much smaller affect on cooling coil temperature.

  If a zone damper system is properly designed and installed, there is a lower limit for cooling coil temperature, and a corresponding limit to Cfm/Ton (which may *be greater than 350 Cfm/Ton, depending on the OEM's value for low limit temperature, and other issues.)*

- Observing that system KW may change as zone dampers close is one thing. Calculating the aggregate impact of the issue is something else.

  The magnitude of the system KW change depends on the details of the scenario, so an investigation would consider a set of likely design scenarios (there would be a maximum, minimum and average value for the set)

  This gets messy. For example, OEM blower tables show that an increase in duct system resistance may translate to more blower power (say 10 Watts per 0.10 IWC), little change in blower power, or less blower power, depending on the product (this behavior needs to be investigated).

  This gets messy. How do we determine the compressor KW change per degree of cooling coil temperature change? Are we talking about a few watts, a 100 watts, or what? Is all equipment equal?

  OEM correction factors (per published performance data) for more or less blower Cfm show that cooling capacity is somewhat sensitive to Cfm (say a one or two percent per 100 Cfm), and that the input KW effect is less than the capacity effect.

  This gets messy, a calculation tool would have to deal with a large set of variables, and correctly estimate small changes in system performance (assuming input data is available, and accurate).

System Merit Depends on Seasonal KWH

System efficiency is a conditional and complex issue, but at the end of the day, overall system efficiency determines system KWH for a default cooling season.

If we are going to compare single zone efficiency with multi-zone efficiency, wouldn't we want to integrate momentary power draw over cooling season time?

$$Seasonal\ KWH = \Sigma\ KW_i \times HOURS$$

In other words, if both systems provide comfort to the best of their ability (everything sized correctly), compare single zone KWH for the season with multi-zone KWH for the season.

Item 6 - Energy Credit

As noted above, seasonal KWH depends on momentary KW integrated over seasonal time. For no set-up or set back, the net effect may be more KWH, less KWH or parity, compared to a single zone system. With set-up or set back, the likelihood of less KWH increases.

The slides only deal with simple single speed equipment and bypass air. Multi-speed equipment and other air relief strategies affect energy use.

- Generalized conclusions are not possible.
- A sophisticated calculation tool is needed to evaluate merit.

Item 7 - Supporting Detail

An effort was made to investigate and understand the information on the slides. Particularly, the slides that pertain to power an energy issues. The following pages provide comments on specific slides.

Supporting Detail

Comments on Most Slides

See also

Excel Spreadsheet -- OEM Data

Slide 3

Current Code

Current Code Requirements

- The prescriptive air flow requirement for 350 CFM/ton in every zonal mode, can be traded away.
- Performance credit (easier heating and cooling set points) for zonal systems capable of maintaining different set points in living and sleeping zones. Common return OK.
- No restrictions on system design, variable capacity control type, commissioning etc.

3.1 Prescriptive Air Flow Comment

Is 350 Cfm/Ton a minimum default value, or a mandatory operating value? Does this apply to single zone systems and air zoning systems, or just for air zoning systems? What is the justification for the 350 Cfm/Ton value?

- The appropriate Cfm/Ton value depends on latent load. In this regard, a colder coil provides more latent capacity.
- 400 Cfm/Ton, or more, provides adequate latent capacity for all USA cities that do not have an unusually large coincident wet-bulb temperature for the summer design dry-bulb temperature (Charleston, SC, for example).
- 500 Cfm/Ton, or more, may be appropriate for a dry-coil climate.
- A substantial amount of HVAC equipment may not be designed to operate at 350 Cfm/Ton (assuming proper refrigerant charge).
- The OEM provides a minimum Cfm per Ton value. If the OEM value is greater than 350 Cfm/Ton, the OEM value supercedes code (due to the laws of physics).
- If a practitioner is ignorant about this issue, he can violate the OEM's guidance by complying with code.
- What does "350 Cfm/Ton in every zonal mode, can be ***traded away***" mean?

3.2 Energy Credit Comment

There are two issues here, which are, different set points, and a common return.

- It is possible that less than whole house conditioning will reduce energy use. Additional comments are provided for the **Code Change Proposal** slide.
- It may be that the return duct system affects energy use, but this may be hard to model. However, there is a significant performance issue.

  Return air from one zone should not affect the thermostat in another zone.

3.3 System Design Comment

There are many strategies for controlling zone temperatures and maintaining suitable equipment operating conditions. In this regard, the devil is in the details.

- The strategy depends on the zone that has the smallest supply Cfm requirement (which depends on zoning decisions).
- The strategy depends on OEM capacity control, and the high or low limit temperature for each capacity stage.
- The strategy depends on the OEM's blower motor type, and its controls.
- The strategy depends on the type of zone dampers (open-close or modulating).
- The strategy depends on the type of zoning controls, sensors and logic.
- The technical issues are manageable, and there are appropriate design procedures.
- It is reasonable to say that energy use is affected by the attributes of a zoning system's design and controls, but quantifying this for all common applications may be impossible.

Slide 4

Typical Practice

Single System with Dampered Supply Ducts

- Return ducts are not zoned
- Single speed compressors and fans cannot modulate to track load
- Supply air flow is low, particularly with one zone calling
- Bypass ducts (short circuit from supply into return) are common
- Results -- low EER

4.1 Return Duct Comment

Return air from one zone should not affect the thermostat in another zone; provide an adequate number of returns (a system design issue).

4.2 Single Speed Comment

The size of the **critical zone** (smallest design value for supply air Cfm) is limited by single speed equipment. If the design is correct, the blower Cfm will not be less than the OEM's low limit value (which may be as low as 350 Cfm/Ton, or more than 350 Cfm/Ton), and the temperature of the air leaving the equipment will be within the OEM's approved range.

- It is assumed that "single speed" implies a PSC blower (vs. and ECM blower)?
- Zoning controls could change PSC blower speed, but this may not be common.
- PSC blower curves tend to be relatively steep, so equipment Cfm does not change much if the system operating point stays on the approved part of the OEM's fan curve.
- Adequate air relief measures are normally required (damper stops, bypass duct, over blow; for example).
- Single stage equipment tends to be compatible with two zones that have similar design values for supply air Cfm.

4.3 Low Supply Air Flow Comment

What does low supply airflow mean?

- The whole point of air zoning is to reduce zone airflow at reduced zone load (to maintain the desired zone temperature).
- If the PSC blower speed does not change. If the blower curve is steep (typical), and if the operating point stays on the approved part of the OEM's blower curve, the acceptable variation in blower Cfm is relatively small.
- If zone airflow is significantly reduced, and if acceptable blower Cfm change is small, adequate air relief measures keep equipment airflow relatively constant.

4.4 Bypass Duct Comment

The momentary amount of bypass duct relief depends on momentary operating circumstances. There are design procedures for determining worst-case (minimum) Cfm and best case (maximum) Cfm. Approved designs maximize momentary bypass effectiveness without causing central equipment problems.

- A bypass duct may be (necessarily) supplemented by some combination of damper stops, bypass duct, dump zone, and zone over blow.
- Bypass effectiveness also depends on appropriate airflow and temperature sensors, controls, and control strategy.
- There still is a lower limit to the size of the critical zone (as far as its design Cfm is concerned).

4.5 EER Comment

What does EER mean, and what does low mean?

- EER is a momentary value (vs. SEER).
- Is EER equal to the total momentary equipment output capacity (Btuh) divided by total momentary input power (KW)?

  Momentary capacity equals the sum of the sensible and latent capacity?

  Momentary input power equals the sum of compressor power, outdoor fan power, and indoor blower power?
- Throttled zone air, and bypass air, tend to reduce evaporator coil temperature, which lowers refrigeration cycle efficiency. So, compressor efficiency, and system EER, depend on momentary evaporator coil temperature.

  The magnitude of this effect varies with the amount of bypass air, which depends on the momentary operating scenario for a given dwelling at a particular location, *served by a given cooling unit that has a given amount of excess capacity, and a given set of performance data.*

How important is this effect for the complete set of California homes (what percentage of cooling season KHW does it account for)?

- It may be that the outdoor fan KW is not significantly affected by the action of the zone dampers and a bypass duct (no effect on system EER)?
- For a constant PSC speed setting, the indoor blower KW tends to increase somewhat as external airflow resistance increases, so there is a small affect on system EER.

  Blower Cfm decreases to the extent that the operating point stays on the blower curve. This may be something like a 100 Cfm (maybe less, depending on how the air relief measures work).

  A change of 100 Cfm as external resistance increases may translate to something like 25 Watts.

  How important is this effect for the complete set of California homes (what percentage of cooling season KHW does it account for)?
- Is there a computer model that computes momentary EER for a given type of dwelling (zoning scenario), for a given equipment make-model-size, for a given amount of excess capacity, for a given air-relief strategy, for a particular location?

  If so, does it compute the dwelling's SEER?

  Then, is there a matrix of SEER values for a set of common dwellings, and cooling system designs, applied to a set of differentiated locations?

  Then, is there a statistical average for the preceding item?

  Then, how does this compare to the average seasonal SEER for a matrix of single zone scenarios?
- A poorly designed single zone, constant volume, system may operate at a coil temperature that just as cold as a properly designed zoned system with a bypass duct (assuming that both operate with no safety trips).

  A single zone system may spend more hours near the low limit temperature because low airflow is a constant condition.

  For a bypass system, coil temperature gets warmer as bypass Cfm decreases.

  There may be run-time issues and start-up issues to investigate (single-zone vs. multizone)?

Slide 5

Proposed Changes

Zoned A/C Code Change Proposals

- Prohibit bypass ducts
- Eliminate the current zonal AC performance compliance credit.
- Mandatory air flow and fan Watt verification in all zonal cooling modes.

5.1 Bypass Duct Comment

What is the rational for prohibiting bypass ducts? If this is an energy issue, see the 4.4 EER comment.

The purpose of the bypass duct is to assure adequate flow though the blower as zone dampers close. The low limit for bypass Cfm depends on a lot of issues (primarily on the OEM's blower data, the momentary Btuh of cooling capacity per Cfm of coil air flow, and the OEM's low limit for discharge air temperature).

- The low limit for bypass air may be less than 10% of the blower Cfm, or more than 50% of the blower Cfm, depending on momentary circumstances; and on how the air zoning system is designed and controlled.
- In other words, momentary bypass Cfm, cooling coil temperature, and system EER depend on a set of environmental variables that apply to all system designs; and on a set of variables that apply to a particular system design.

A properly designed bypass system, will not, for worst-case conditions (critical zone damper open, all others closed), operate at less than the OEM's lower limit value, which may be 350 Cfm/Ton, or significantly more than 350 Cfm/Ton.

- Zone damper system controls should shut the system down (in a normal manner) before the OEM limit temperature control acts.
- Routine shut downs will not occur if the system is designed and installed correctly.

A properly designed zone damper system may require a set of air relief measures. Eliminating the bypass option will, in general, have an adverse effect on system performance.

- Comfort is the primary reason for zoning. In this regard, the goal is to provide for precise temperature control for all rooms and spaces.

  Bypass air has no adverse effect on zone temperature control (zone Cfm matches zone load).

  Distributed relief (damper stops) and zone over blow conditionally reduce zone temperature control (zone Cfm may not match zone load).

  A dump zone may be conditionally uncomfortable.

- Preferred temperature control may require three or more zones, but the air relief requirement increases with the number of zones.

  The number of zones determines critical zone Cfm.

  Eliminating the bypass air option may force a comfort-compromised zoning plan (zone consolidation).

- Two stage equipment (compressor and blower) reduces the air relief requirement, but does not automatically eliminate the need for a bypass duct.
- Selective throttling systems (reduce compressor capacity and blower Cfm, and use a comfort zone as a dump zone) are proprietary OEM packages.

  They do not require a bypass duct, per proprietary design rules.

  The OEM's design rules may limit the number of zones (a comfort compromise).

  Using a comfort zone as a dump zone conditionally reduces zone temperature control (zone Cfm may not match zone load).

- A bypass duct is an important, effective and common air management tool.

  No adverse affect on zone temperature control.

  Significant method for stabilizing blower Cfm and external static pressure (reduces blower operating point excursions).

  The pressure drop for the bypass circulation path (for full bypass Cfm) is no larger than the pressure drop for the zone circulation path that has the most airflow resistance (all circulation paths are in parallel).

  An OEM zoning product (turnkey system) may be designed for capacity control (blower and compressor), with a bypass duct.

  Many zone damper vendor products utilize bypass air.

5.2 Zonal Credit Comment

It is possible that less than whole house conditioning will reduce energy use. However, there should be a way to identify favorable set-up, set-back scenarios (considering climate, envelop performance, primary equipment performance, and zoning equipment performance); and a way to estimate energy savings and dollar savings.

- One obvious issue (among many) is minimum set-back and set-up duration.

  Thermal mass that has cooled down has to be reheated, and vise versa.

  For humid climates, moisture absorbed during set-up has to be removed during recovery; and, entering wet-bulb affects coil performance during recovery.

- Energy use depends on the type of equipment that is used for set-back recovery (heat pump only; electric coil and heat pump; electric coil only; or furnace only).
- We assume that the home owner will use an effective set-up, set-back schedule.

5.3 Air Flow and Fan Watt Test Comments

What is the rational for fan watt testing? This could get complicated and time consuming. How does this correlate with annual energy use?

There may be two-zones to more than four zones. There are PSC blowers, ECM blowers, and variable speed blowers (and blower speed changes). There are open-close zone dampers and modulating zone dampers. There are various methods of air relief, plus selective throttling. There are various types of air relief controls and control logic, plus OEM proprietary selective throttling strategies. There is a significant range of OEM low limit values for cooling Cfm and discharge air temperature. There is one stage equipment, staged equipment, and variable speed equipment. Etc.

- Blower motor power (watts) depends on blower speed, blower Cfm, airflow resistance within the cabinet, duct system resistance, blower efficiency, and blower motor efficiency.
- Blower power may be relatively constant for a single zone system operating at one speed.

  Duct airways are sized (by duct slide rule) for design day airflow rates, and a design friction rate value.

  See the Duct Design sidebar.

- Blower power varies for a zone damper system.

  Duct airways are sized (or oversized) for design day airflow rates.

  Duct airways may larger to compensate for control damper pressure drop.

  Duct run, duct fitting, and device resistance (coil, filter, etc.) decrease at reduced air flow (zone damper resistance increases).

Duct Design

The design friction rate value depends on the **available static pressure** for straight runs and fittings, and on the **total effective length** of the longest circulation path (straight run lengths plus fitting equivalent lengths).

The **available static pressure** equals the blower table **external static pressure** minus the pressure drop for components and devices **that were not in place** when the blower was tested (supply grille, return grille, hand damper, accessory filter, cooling coil added to a furnace, for example).

If proper duct sizing procedures are not used, airways tend to be too small, and fittings tend to be inefficient, so blower power is more than what it would be for a correct design.

- A PSC blower may be set to a higher speed setting (this may, or may not, provide the desired air flow rate).
- An ECM blower will automatically speed up, perhaps to its static pressure limit, and maximum watts. If the normal speed increase does not fix the problem, the Cfm setting can be increased (if not already at its maximum value).

The concept of one blower watt value for all duct systems, no matter what, is questionable. The goal should be appropriate blower power for a given set of circumstances.

- Fittings have a significant affect on system resistance.(use efficient fittings).
- Cooling coils that have the same cooling capability may have significantly different pressure drop values for the desired blower Cfm (0.05 IWC to more than 0.10 IWC).
- Accessory components have a significant affect on system resistance. In this regard, an small-particle filter may add more resistance than an open zone damper.

- Does the code limit blower Watts? If so, does the code just assume that proper duct design/sizing procedures can produce adequate airflow without exceeding the blower power limit for any set of circumstances?

  Larger duct airways and aerodynamic fittings tend to compensate for limited blower power, but this may not be a viable solution for all scenarios.

- For zone damper systems, there is momentary blower power (KW), and seasonal energy use (KWH).

  Seasonal KWH depends on many variables (all the issues mentioned above and on the preceding pages).

  Is there a computer model for seasonal KWH?

  Is there a way to compare seasonal KWH for a representative set of zone damper systems with the seasonal KWH for a representative set of single zone systems?

Slide 6

Home Survey

California New Home Energy Survey

We have completed the survey of a sample of 80 new 2007 CA homes and found:

- 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
- MF duct leakage is very high, SF is pretty good

6.1 General Comment

This is a response to items on a slide show. This format uses a few words to summerize complex issues, observations and concepts. This is ok, but responses are based on assumptions pertaining to what the presentation is actually trying to say. Responses are not based on historical knowledge of related reports, documents, and hearing/meeting discussions.

6.2 Low Capacity Comment

On average, across the country, the way things are done do not correlate with the way things should be done. Investigations by various persons and organizations support these conclusions:

- Cooling equipment has significant excess capacity (when installed equipment size is compared to an aggressive ***Manual J*** load).
- Cooling equipment delivers less than its full capacity because of incorrect practices (refrigerant charge, excessive duct resistance, and duct efficiency issues).
- Cooling equipment performance is affected by return duct issues (conduction and leakage affects sensible and latent capacity at the equipment cabinet).
- A room, space or zone may have deficient capacity, even if the central equipment has excess capacity (usually a duct design and installation issue; and/or an air balancing issue).
- In general, installed capacity is excessive, and delivered capacity (at the cabinet, and for some collection of rooms and rooms and spaces) is deficient.

6.3 Low Efficiency Comment

On average, across the country, the way things are done do not correlate with the way things should be done.

- Duct system design and installation is the biggest issue when ducts are installed in an unconditioned space (this affects equipment efficiency and distribution efficiency).
- Refrigerant charge is a significant issue.
- Maintenance is an issue (air-side components must be clean, refrigerant must not be contaminated or restricted).
- The preceding items are much more important than a few points in published SEER, which applies to test chamber conditions that may not, and usually do not, simulate the operating condition at a home site.

  Maximum achievable SEER varies with climate and system design conditions.

  The maximum achievable SEER may be less than or greater than the published SEER value

  High efficiency equipment with poor design and installation may be less efficient than average efficiency equipment with good design and installation; regardless of climate.

6.4 High Duct Pressure Comment

The OEM's blower curve is what it is, but the resistance curve for necessary air-side components and duct runs depend on the practitioners's design method and installation practice.

- Aerodynamically inefficient fittings cause unnecessary resistance to airflow.
- Undersized airways cause unnecessary resistance to airflow.
- Proper airflow depends on the climate situation (this may be 400 Cfm per nominal AHRI ton, or less; to 500 Cfm per nominal AHRI ton, or more).
- It is assumed that "duct pressure" means available static pressure for supply and return distribution.

  Inefficient duct fittings and undersized airways unnecessarily increase system operating pressure.

  Required and/or desired air-side components produce a necessary increase system operating pressure.

There is no magic number for maximum system operating pressure, but proper design and installation will minimize this on a case by case basis.

- OEM efforts to compensate for practitioner/owner ignorance and negligence can be counter productive. ECM blowers increase system operating pressure when fittings are inefficient, when duct airways are too small, and when air-side components are dirty.

6.5 Fan Watts Comment

It is true that fan watts tend to be excessive on a case-by-case basis. However, there is no magic number for maximum fan watts. See 6.4 comment.

Fan watts is a complex conditional variable for zone damper systems. There is a peak value (which affects power draw, and an average value (which affects seasonal energy use). Trying to sort this out is a can of worms. See the 5.3 comment.

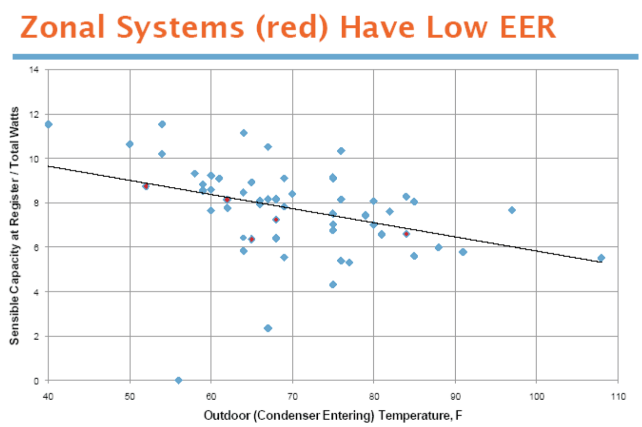
6.6 Duct Efficiency Comment

Not sure what MF duct leakage means and what SF means, but the industry knows that duct system efficiency has a significant affect on comfort, equipment performance, energy use, and operating cost.

- Ducts should be in the conditioned space to the extent possible (considering technical and economic issues).
- Ducts in an unconditioned space must be sealed to the appropriate standard, and insulated to at least R-6, and preferably to R-8. A vapor retarding jacket may be required for humid climates.
- Every effort should be made to minimize the surface area of duct runs in an unconditioned space.
- ***Manual J*** (MJ8) procedures reward efficient duct systems, and severely penalize inefficient duct systems.

Slide 8

Zoned Systems have High Low EER



8.1 EER Comment

Isn't EER defined as momentary cooling equipment output in Btuh divided by momentary energy input in watts? Showing EER as supply air outlet output Btuh divided by the total electric energy input watts is sophistry.

- The reason for air zoning is improved temperature control for rooms and spaces. This is accomplished by reducing supply air Cfm to the zoned space.

  Supply air Btuh per system input KW must decrease when zone dampers fulfill their mission.

  What else might we expect (are tests really necessary to verify this behavior)?

  Using the supply air Btuh to input KW ratio to compare zone damper systems with single zone constant volume systems is an apples and oranges exercise.

- What about latent capacity (some California climates produce latent loads).

- Seasonal system efficiency for a zoned system can be compared to a single-zone system.

  Published equipment SEER is not relevant (it does not model a particular set of circumstances, except by chance; and a given piece of cooling equipment may serve a single-zone system, or a zone damper system.

  System SEER must be scenario specific (depends on local weather data, architectural and structural attributes, comfort system capabilities, equipment performance maps, control strategy, etc.).

  SEER must correlate with the real issue, which is seasonal KWH and peak momentary KW.

- What are the quality control conditions for comparing system SEER?

  For any installation there is a maximum system SEER (no design or instillation flaws); and a degraded SEER (some combination of design and installation flaws).

  System merit should be based on the no design or installation flaws scenario.

  Quality assurance mandates should be based on statistical data for observed design flaws and unapproved practices. In this regard, there may be more mandates for air zoning vs. single zone.

- Momentary EER merit (no design or installation flaws) depends on performance attributes that affect single zone systems and zone damper systems.

  Total capacity, sensible capacity, coil sensible heat ratio, and input KW depend on coil Cfm, outdoor temperature, entering wet-bulb, and entering dry-bulb.

  Input KW equals compressor KW, outdoor fan KW and blower KW (plus some controls power).

  There is a data set for each capacity stage (when applicable).

- Momentary EER merit may depend on issues that are peculiar to zone damper systems.

  For constant blower RPM, blower pressure and blower motor KW tend to increase as zone dampers close; but bypass air, damper stops, dump zone, over blow and selective throttling affect the amount of change in blower pressure, and blower KW.

  There is a significant difference in the way a PSC blower and an ECM blower react to an increase in duct system resistance (KW increases as the PSC operating point moves up the fan curve, or KW increases as the ECM motor speeds up).

  Reduced Cfm across the cooling coil (air relief measures do not completely compensate for throttled zone dampers) lowers coil temperature, and decreases refrigeration cycle efficiency.

  Using a bypass duct to maintain airflow across the cooling coil lowers coil temperature, and decreases refrigeration cycle efficiency.

- See Sections 4.4, 5.3, 6.3 and 6.5 for related comments.

8.2 Comments on the Low EER Graph

There are three markers at average, or close to average. There is one marker that is somewhat below average. There is one marker that is significantly below average. There are a lot of other markers that are much worse than the zoned system markers.

It looks like the tests were conducted for the outdoor condition that existed when the technician arrived at the home site. This has some affect on equipment power draw, and considerable affect on equipment run time. This also has an effect on supply air temperature at the outlets (attic ducts, for example). For zoned systems, this has an affect on supply Cfm at the outlets.

- Momentary power draw (KW) may be zero, or a positive value (which is a conditional variable).
- Annual energy use (KWH) depends on momentary power draw (KW) integrated over seasonal use time (hours).
- Seasonal energy output at the registers depends on momentary values for supply air temperatures and supply air Cfm, integrated over seasonal use time.
- A momentary snapshot does not summarize seasonal performance.

For zone damper systems, momentary outdoor temperature and solar gain have a significant effect on all attributes of system operation. Single zone systems also are affected, but blower Cfm and room supply air Cfm are constant. These tests do not evaluate these issues.

Duct system efficiency should be comparable when a zone damper system is compared to a one-zone system.

- Same location, same floor plan, same construction, same direction for the front door.
- All duct systems correctly designed for their transport load, as far as surface area is concerned.
- All duct systems sealed to the same standard, and insulated to the same standard.

Were all these systems single stage systems, or did some systems have capacity control (for blower and/or compressor performance)? This would have a significant affect on momentary system performance.

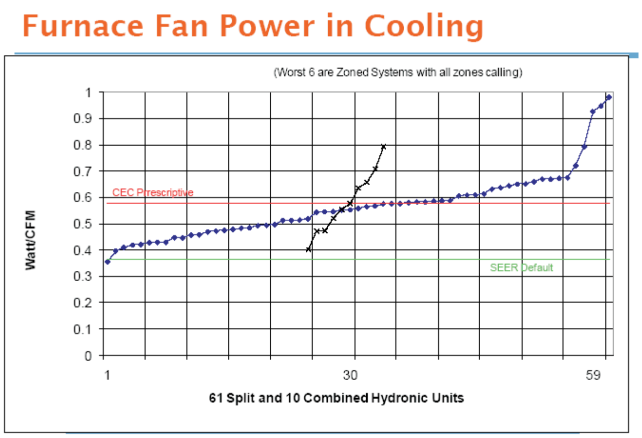
These tests may not be sensitive to equipment installation issues. Was the central equipment correctly sized or oversized? Was Cfm per nominal AHRI ton equivalent for one zone and multzone tests? Is refrigerant charge correct for one zone and multizone tests?

These tests do not seem to be apples-to-apples, as far as the issues that affect momentary system efficiency are concerned. And, the presentation implies that momentary efficiency is equivalent to seasonal efficiency.

This testing effort is affected by too many variables to pass judgement on air zoning. If all the marks on the graph were the same color, which ones should we be most concerned about, and why?

Slides 9 and 10

Zoned Systems have High Watts per Cfm



9.1 and 10.1 Comments

This testing effort is affected by too many variables to pass judgement on air zoning.

Everything else being equal, a zone damper system has some additional flow resistance because of its zone dampers. However, this may not explain the size of the spike in Watts/Cfm for zone damper systems. In this regard, do we know how the installed attributes of the six zone damper duct systems, and their blowers, compare with approved design procedures?

- Is this an unavoidable system attribute issue?
- Is this an avoidable installed performance issue (practitioners must know how to correctly design and install zoned duct systems)?
- It may be (on a case by case basis), that careful design and proper installation can compensate for the pressure drop though open zone dampers.
- Item 6.4 and 6.5 comments are relevant to this slide.
- Did anyone check to see if all the zone dampers were actually wide open?

For air zoning, Watts/Cfm is a conditional variable. A comparison of the seasonal average for single-zone systems and multizone systems may be of more interest.

- For PSC blowers, the operating point moves along the blower curve as zone dampers operate. Blower speed changes are possible (depends on the controls and control strategy).
- For ECM blowers, the motor speeds up as zone dampers close. Blower Cfm set point changes are probable (depends on the controls and control strategy).
- True variable speed blowers tend to minimize fan power because RPM is reduced as zone dampers close.

If system Cfm is measured at the supply air outlets, how much did the supply ducts leak for each test?

- Supply outlet Cfm = Blower discharge Cfm - Supply duct leakage Cfm
- Is the prescriptive Watt/Cfm value for supply outlet Cfm, or for blower Cfm?

Slide 12

Bypass Duct Relief

How Zoning with Bypass Works

In Theory:

- With all zones calling, the bypass damper closes and bypass has no effect. All zones get the design air flow
- When only one zone calls, whatever isn't delivered to that zone is bypassed to the return to maintain coil air flow.

Actually

- Mixing in bypass air lowers the return air temperature entering the cooling coil and this ALWAYS significantly lowers the EER.
- Because of dampers and extra ducts the air flow is typically very low even when all zones are calling.
- Extra dampers and ducts make systems more prone to construction error and failures are common.

12.1 How Bypass Works Comment

Maximum bypass air Cfm is conditional. The momentary bypass Cfm value may vary from less than 10% of the momentary blower Cfm (which may be staged), to more than 50% of the momentary blower Cfm.

- The bypass Cfm demand depends on the smallest design Cfm for the various zones (the critical zone).
- A larger critical zone Cfm translates to less bypass Cfm (for a given floor plan, two large zones are easier to deal with than four zones).
- Total air relief may use some combination of a bypass duct, damper stops, a dump zone (or undamped rooms), and critical zone overblow.
- The air relief strategy must prevent a blower problem, or a discharge air temperature limit problem, when the critical zone is the only open zone (appropriate design procedures are available).
- If the bypass duct has a counter weight damper, it can only react to the worst-case scenario. So, if acceptable bypass Cfm varies from 10% to 50% of blower Cfm, the counterweight is set for 10%.

  Because there is no feed back control.

  If the counterweight is set for more than 10% (for a 10% scenario) there may be controlled shutdowns or nusance temperature trips.

  For controlled shutdowns, zoning controls shut the equipment down (in a normal manner), even if the zone thermostat is calling for conditioned air (this is a shall, as far as proper design is concerned).

  Nusance temperature trips occur when OEM safety controls are forced to act. The system may not restart unless controls are reset (this is a shall-not happen, as far as proper design is concerned).

  Controlled shutdowns depend on momentary operating conditions. A limited number of occurrences may not be noticed by the occupants.

  There is no procedure for predicting the number of controlled shutdowns for a particular system, in a particular home, at a particular location.

- If the bypass damper has feedback control (based on blower static pressure and discharge air temperature), the damper opens to the maximum position that will not cause a pressure or temperature problem. In other words, bypass Cfm is conditionally maximized, and system airflow resistance is conditionally minimized.
- The bypass damper should not be used for air balance.

  A hand damper in the bypass duct reconciles bypass path resistance with zone path resistance (hand dampers also are required for zone paths).

  Some designs use a bypass airway size that causes high bypass air velocity (2,500 to 4,000 Fpm). Noise may be an issue, and design procedures tend to be rule-of-thumb (a rigorous sizing procedure is mathematically challenging for day-to-day work).

12.2 EER Comment

A colder coil does reduce refrigeration cycle efficiency, but this is not the only issue. See Section 8.1.

12.3 Dampers and Extra Ducts Comment

Open control dampers do add an increment of airflow resistance. Efficient fittings and proper duct system design and airway sizing can minimize the effect.

- If the dwelling has zone load diversity (for time of day), *all zones calling* is not a normal operating condition.
- Proper system design procedures and duct sizing procedures provide adequate airflow for all possible load scenarios.

- What does extra ducts mean?

  The design value for duct system resistance depends on the total effective length of the longest circulation path. It makes no difference if there are two shorter paths or ten shorter paths.

  Over sizing duct airways (as recommended by some vendors) reduces system resistance for all operating conditions.

- Air zoning system design and installation is more complex than single zone design.

  Single zone system and multi-zone systems are equally vulnerable to load calculation, equipment sizing, duct airway sizing, and air outlet selection errors.

  Comprehensive air-zoning guidance is available.

  Design work peculiar to air-zoning involves zone selection, excess air management strategy and calculations; and if used, bypass duct design.

  OEM's and zone damper vendors provide comprehensive installation instructions.

- From the code point of view (and from the home owner point of view) quality control is a "how do you control human behavior" issue, not a technical issue. In this regard, home owners are culpable, and/or innocent victims.

  They may want the lowest price.

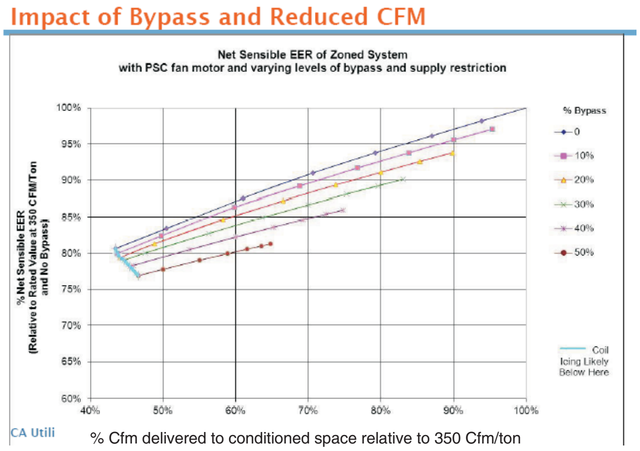
  Even if they do not want the lowest price, they do not know what questions to ask.

12.4 General Comment

The air zoning issue is too complex for the Slide 12 statements. Much more thought and work is required.

Slide 15

Zoned Air Conditioning Model



15.1 General Comment

No information about the model's capabilities, sensitivities, and mechanics provided.

It looks the model applies to a compressor and PSC blower operating at one speed.

- Shouldn't bypass air Cfm (per graph notes) plus Cfm delivered to the conditioned space (per x-axis label) equal 100%?
- Why are the bypass air models lines instead of single dots (for example, 20% bypass would have one EER value for 80% Cfm to the space)?

If this slide is consistent with Slide 8 (Sensible Capacity at the Register), EER is defined as supply air Btuh divided by system KW. **See Section 8.1 for comments on this practice.**

If full flow (100% blower Cfm) is 350 Cfm/Ton, and if there is no bypass air, the graph shows that coil Cfm drops to 245 Cfm/Ton at 70% flow, and to 175 Cfm/Ton at 50% flow. Can cooling equipment tolerate this behavior?

- OEM's have a lower limit for the Cfm per Ton value.
- The OEM's low temperature safety limits Cfm/Ton, depending on the operating condition.

15.2 More 350 Cfm/Ton Comments

Why is 350 Cfm/Ton used (for this slide, and through out the presentation)?

- This is an uncommon result for properly sized equipment (typically 400 Cfm/Ton, or more, when expanded OEM data is used to make sure that sensible and latent capacity is compatible with sensible and latent load for the indoor and outdoor conditions on a summer design day).

  A dry climate favors more than 450 Cfm/Ton. Many California locations are very dry, or relatively dry.

  Locations like Charleston, SC, New Orleans, LA; and Mobile, AL favor 400 Cfm/Ton or less, for a colder coil and more latent capacity (not applicable to California).

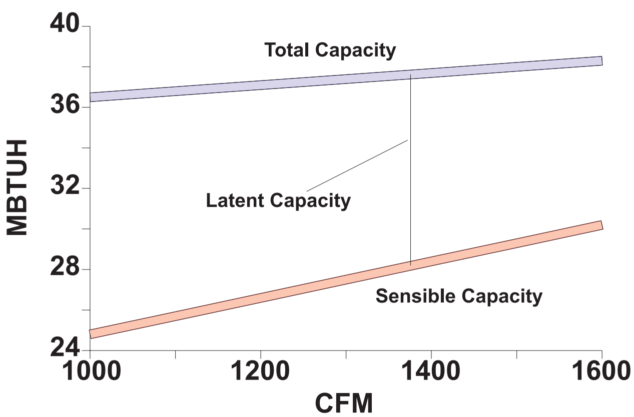
  There may be equipment that cannot operate at 350 Cfm/Ton.

  Low Cfm per Ton translate s to a smaller value for maximum bypass air (explained later).

15.2 Cooling Capacity Vs. Blower Cfm Comment

Cooling performance detail varies somewhat, depending on make and model, but capacity vs. blower Cfm tendencies are similar across product lines, as indicated by the following graph .

- Continuous operation with a wet coil.
- For a dry coil, sensible capacity is approximately
- equal to total capacity.
- It would be useful if there was a line for refrigeration cycle power (compressor KW + outdoor fan KW) .
- Blower KW vs. Cfm depends on what causes the Cfm change (change blower speed, or vary duct system resistance at constant speed).



15.3 EER Comments for 0% Bypass

OEM performance data correlates system KW with variations in coil Cfm. For example, refer to the performance data for a York cooling unit (see the attached Excel file).

- The system tab shows cooling performance values for five PSC blower speed settings (Cfm), and for five sets of outdoor-entering conditions (a sub set of the actual OEM data set).

  *Total Capacity per Watt (TC/Watt) generally increases somewhat as blower **speed** and Cfm decrease.*

  *Sensible Capacity per Watt (SC/Watt) generally decreases (a little more than the change in total capacity Watts) as blower **speed** and Cfm decrease (as demonstrated by the preceding graph).*

Conditional Blower KW

OEM performance data notes say that system KW includes compressor KW, outdoor fan KW, **and indoor blower KW**. In this regard, a system KW change due to a blower speed change may not be equivalent to a system KW change due to a duct system resistance change at constant blower speed.

- Compressor KW and outdoor fan KW may not be affected if a blower Cfm change is caused by a blower motor speed change, or by a throttled zone damper.
- If we had blower KW values for each blower speed-Cfm scenario, they could be subtracted from the system KW values.
- Then for a given duct system resistance scenario, blower KW for a Cfm-IWC set could be read from the OEM's blower table, and added to the compressor KW and the outdoor fan KW.
- Then system KW values could be calculated for Cfm-IWC scenarios caused by zone damper movement.

Blower KW for OEM Data

The cooling performance data for the York unit does not provide values for indoor blower KW, but the corresponding blower table has blower KW values, so a rough blower KW value can be subtracted from the system KW.

- Assume the cooling system KW values are for 0.20 IWC of external static pressure (OEM's tend to choose defaults that minimize KW values).

  For 95/80/67, at 1,050 Cfm, the cooling system data shows 3.1 KW. At 1,025 Cfm and 0.20 IWC, the blower table shows 0.158 KW. So the KW for the compressor and outdoor fan is about 2.942 KW.

  For 95/80/67, at 1,350 Cfm, the cooling system data shows 3.2 KW. At 1,370 Cfm and 0.20 IWC, the blower table shows 0.305 KW. So the KW for the compressor and outdoor fan is about 2.895 KW.

  So for 95/80/67, it looks like 2.92 KW (average 2.942 and 2.895) is the compressor and outdoor fan KW.

- The following KW values apply to the compressor and outdoor fan when the math from the preceding bullet is applied to the (95/80/67); (95/80/57); (95/75/57); (85/80/57); and (85/75/57) scenarios.

  95°F OAT, 67°F EWB = 2.92 KW > 2.9 KW
  95°F OAT, 57°F EWB = 2.87 KW > 2.9 KW
  85°F OAT, 57°F EWB = 2.47 KW > 2.5 KW

- OEM data shows that changes in the blower Cfm setting have a very small affect on system KW, and changes in outdoor temperature have some affect (10% to 15% per 10°F) on system KW.

  For 95/80/67, at 1,050 Cfm, the York cooling system data shows 3.1 KW. At 1,350 Cfm, the data shows 3.2 KW.

  the red markers on the Excel spreadsheet's Data tab show similar behavior.

Air Zoning Affects System KW

The primary difference for system KW for single zone vs. air zoning may be due to blower KW, and to cooling coil temperature.

- As far as compressor KW is concerned, outdoor temperature has a similar affect on single zone systems, and air zoned systems.
- Blower power changes as zone dampers operate. Blower power is constant for single zone, constant volume systems.
- Cooling coil temperature affects refrigeration cycle efficiency. Cooling coil temperature depends on zone damper movement, the type of air relief, and the amount of air relief.

Blower KW for the Design Cooling Load

When a one-zone system is compared to multi-zone system, there may not be much difference in blower Cfm for the summer design condition.

- The block load for the conditioned space is used for equipment sizing. This load is the same for single zone systems and multi-zone systems (it takes credit for time of day diversity, if the dwelling has diversity).
- See the Blower tab on the Excel spread sheet.

  Assume the duct system is designed for 1,400 Cfm (based on matching OEM performance data to the sensible and latent cooling loads for 1,350 Cfm capacity data).

Say that the total effective length of the longest circulation path is 400 feet (a reasonable value for straight runs and fittings).

If a single zone constant Cfm system is designed for 0.40 IWC of external static pressure, the airway sizing friction rate is 0.05 IWC/100Ft , which is too low; and there are 413 blower Watts (see cells B18 to E28).

If a single zone constant Cfm system is designed for 0.60 IWC of external static pressure, the airway sizing friction rate is 0.10 IWC/100Ft, which is ok; and there are 530 blower Watts (see cells H18 to K28).

If zone dampers are added to the system, an open zone damper produces an additional 0.10 IWC of resistance (roughly). If the zone damper system is designed for 0.60 IWC of external static pressure, the airway sizing friction rate is 0.08 IWC/100Ft, which is ok; and there are 530 blower Watts (see cells M18 to Q28).

- So at full air flow on a design day, there may be no difference in blower KW (single zone vs. air zoning), but one-zone airways will be sized for 0.10 IWC /100Ft vs. 0.08 IWC /100Ft for zoning.

  Note that it may not be possible to operate at 500 blower Watts, or less, for a one-zone or multizone system. To get to 400 Watts, the total effective length of the straight runs and fittings must be reduced from 400 feet to 250 feet, which may not be possible if the 400 foot value is based on efficient fitting use.

Blower KW Vs. System Resistance

For no air relief measures, blower Cfm decreases, and external static pressure increases as zone dampers close. When this happens, the air power equation provides a theoretical value for blower watts.

Watts = 745.7 x Cfm x AFR / (6,356 x EFF)

Where:
Cfm = Blower Cfm
AFR = Air flow resistance (IWC)
AFR = External resistance + Internal resistance
Internal resistance is produced by blower cabinet components, and the entrance and exit resistance.
$AFR_2 = AFR_1 \times (CFM_2 / CFM_1)^2$
EFF = The net efficiency for the blower and its motor
EFF is not published with OEM blower data

Actual blower watts may be read from the OEM's blower table. This may not be consistent with the air power equation (assuming the OEM data is correct).

- For example, see the Blower tab on the Excel spreadsheet (for a York blower).

  The table at the top of this page summarizes the York blower data for Watts vs. ESP.

Blower Speed	External Static Pressure (IWC)									
	0.2		0.4		0.6		0.8		1.0	
	W	Δ	W	Δ	W	Δ	W	Δ	W	Δ
L	158	-	175	17	-	-	-	-	-	-
L/M	237	-	260	23	283	23	307	24	-	-
M	305	-	330	25	354	24	377	23	397	20
M/H	-	-	413	-	436	23	454	18	460	6
H	-	-	-	-	530		538	8	521	-17

Note that Watt steps are about 23 KW for 0.20 IWC pressure steps at low to medium speed; but this pattern does not apply to higher speeds.

Note that blower KW changes can be small or negative when the blower is pushed to its aerodynamic limits (presumed reason for erratic performance at high speed).

- Similar, behavior is demonstrated by the blower table for an American Standard blower (see cell O3 on the Excel spreadsheet -- Data tab).
- The blower table for a Lennox multi-speed, direct drive blower (see cell B85 on the Excel spreadsheet data tab) tells a significantly different story. In this case, blower Cfm and watts significantly decrease as external static pressure increases for any blower speed setting.
- So for throttling zone dampers with no bypass air, it looks like the fan power change can be positive (about 0.1 KW per 0.1 IWC for most blower speeds settings), negligible or negative for some higher speed setting), or consistently negative at any blower speed (for the Lennox furnace blower).
- Blower power changes, and the rate of change vs. external static pressure change seems to depend on the product, and the blower speed setting; and the difference in behavior seems to be significant. This requires more investigation.

Compressor KW Vs. Coil Airflow

We are talking about the 0% bypass scenario, so cooling coil temperature tends to decrease as supply air Cfm is throttled; but for no bypass Cfm (or other air relief measures), the effect is similar to reducing blower speed.

- The OEM performance data on the Excel spreadsheet (System tab) shows what happens to system KW as coil Cfm drops, but the values include blower power.
- The discussion at the lower left of the preceding page shows what happens to Compressor power

and outdoor fan power as blower air flow drops from 1,350 Cfm to 1,050 Cfm.

For 95/80/67, at 1,350 Cfm, the cooling system data shows 3.2 KW. At 1,370 Cfm and 0.20 IWC, the blower table shows 0.305 KW. So the KW for the compressor and outdoor fan is about 2.895 KW.

For 95/80/67, at 1,050 Cfm, the cooling system data shows 3.1 KW. At 1,025 Cfm and 0.20 IWC, the blower table shows 0.158 KW. So the KW for the compressor and outdoor fan is about 2.942 KW.

For the York unit, it looks like a 300 cfm drop in coil airflow produces an 0.05 KW increase in compressor and outdoor fan power.

- So for a 22% decrease in York coil Cfm, it looks like Compressor plus outdoor fan power may increase by something like 1%.

  Blower% = (1,350 - 1,050) / 1,350 = 22%
  KW% = (2.924 - 2.895) / 2.924 = 1%

  The red markers on the Excel spreadsheet's Data tab show similar behavior.

15.4 Additional Comments for 0% Bypass

For the upper boundary (the 0% bypass line), it looks like the bypass damper is locked tight, and supply Cfm is reduced.

- The markers imply that supply air Cfm was modulated from 100% to 45%.

  Does the model use a representative model of a PSC blower curve? If so, is it steep? What are the upper and low limits for the approved operating range?

  If PSC performance is modeled, will the duct system operating point stay on the PSC blower curve as zone dampers close?

  Blower curves tend to be fairly steep. It is hard to believe that the operating point stays on the approved part of the blower curve for such a large change in blower Cfm.

- Supply Cfm goes down (significantly), as KW draw goes up (marginally), so % sensible EER goes down. What do we really learn here? Even if the denominator (KW) is constant (best case), making the numerator (sensible) smaller always makes sensible EER smaller.

- The coil is already operating near its low limit at 350 Cfm/Ton at 100% supply air Cfm, then Cfm is reduced.

  The maximum Cfm reduction depends on sensible Btuh capacity per Cfm of flow (B/C ratio).

  Sensible Btuh per Cfm and leaving air temperature depend on outdoor temperature (equipment capacity increases as outdoor temperature decreases).

  Sensible Btuh per Cfm and leaving air temperature depend on coil sensible heat ratio (the worst case is 1.00).

  The maximum Cfm reduction also depends on the OEM's value for low limit temperature (this may range from about 38°F to about 50°F).

- What were the values for the B/C ratio at AHRI rating conditions, the scenario's's outdoor temperature and coil sensible heat ratio, and the OEM's low limit temperature?

- If conditional sensible capacity and limit temperature are modeled, does the leaving air temperature stay above the OEM's low limit all the way down to 45% Cfm?

15.5 EER Comments for Bypass Air

Cooling coil temperature affects refrigeration cycle efficiency. Cooling coil temperature depends on zone damper movement, the type of air relief, and the amount of air relief. Air relief tends to stabilize blower Cfm, and blower KW.

- Bypass air produces a temperature ramp for leaving air (supply air at the coil). This transient will settle to a steady value in a matter of minutes. The settled value must not be less than the OEM's low limit value.

- The maximum bypass air Cfm for a given operating condition depends on sensible Btuh capacity per Cfm of coil air flow (B/C ratio).

  Sensible Btuh per Cfm depends on outdoor temperature (equipment capacity increases as outdoor temperature decreases).

  Sensible Btuh per Cfm depends on coil sensible heat ratio (the worst case is 1.00).

  The maximum bypass air Cfm value increases as the B/C ratio decreases (outdoor air temperature at the condenser gets warmer, and/or outdoor moisture increases the latent load on the cooling coil).

  350 Cfm/Ton at full air flow is not deniable.

- The maximum bypass air Cfm also depends on the OEM's value for low limit temperature (this may range from about 38°F to about 50°F).

- If bypass air is properly managed, the coil temperature will vary from a minimum, to a no bypass air value.

  With proper bypass damper design and controls, the minimum value will never be less than a degree or two warmer than the OEM's low limit value.

  The no bypass value is approximately equal to the single zone value.

Leaving air temperature is momentary event, so the affect on system efficiency must be integrated over seasonal time. A crude scope of work is provided here (for constant speed equipment).

- Define a default cooling season.
- Estimate the seasonal operating hours for a single zone system, and a capacity-equivalent zoned system.
- Estimate the average leaving air temperature for a default cooling season for a single zone system, and a capacity-equivalent zoned system.
- Determine relationship between leaving air temperature and compressor KW.
- Use average leaving air temperatures to compute average compressor KW values for a single zone system, and a capacity-equivalent zoned system.
- Compute and compare the seasonal KWH for a single zone system, and a capacity-equivalent zoned system.

Bypass air tends to stabilize momentary blower pressure and Cfm, and momentary blower KW. So, bypass air, (and other air relief measures) tend to equalize blower KW for a zone damper system vs. a single zone system.

- Bypass air, by itself, may not provide sufficient air management (correct design procedures produce an appropriate set of air relief measures for a given set application details).
- For sufficient air management, momentary zoned blower Cfm and pressure tends to be similar to momentary single-zone blower Cfm and pressure (increased back-pressure caused by zone damper closure is relieved by the air management strategy, and system Cfm is relatively constant).
- So, the primary difference (single zone vs. zoned) in system resistance is the pressure drop for an open zone damper.
- An open zone damper increases the design value for duct system resistance. This tends to increase blower KW, on a seasonal basis if duct airway size is not adjusted.
- If ***Manual D*** is used to size duct airways, the design friction rate for airway sizing will be somewhat smaller for a zone damper system, compared to no zoning.
- A smaller friction rate translates to larger airways for a zoned system, so zoned duct resistance will be comparable to a single zone system (see cells G18 to Q29 on the Excel spreadsheet -- Blower tab).

15.6 Additional Comments for Bypass Air

For the bypass curves, bypass air is incrementally increased from 0% to 50%. Then for each curve, it looks like modulating zone dampers move from some open position toward closed.

- Look at the 50% bypass curve. If 50% of the system air is bypassed, how can the supply Cfm to the conditioned space be greater than 50%? The other curves show the same behavior.
- Is the chart saying that the EER curves for bypass air Cfm are progressively lower than the no bypass curve because bypass air causes a lower coil temperature?
- The coil is already operating near its low limit at 350 Cfm/Ton at 100% supply air Cfm, then bypass air is activated.

  The maximum bypass Cfm value depends on sensible Btuh capacity per Cfm of flow.

  Sensible Btuh per Cfm and leaving air temperature depend on outdoor temperature (coil capacity increases as outdoor temperature decreases).

  Sensible Btuh per Cfm and leaving air temperature depend on coil sensible heat ratio (the worst case is 1.00).

  The maximum bypass Cfm value also depends on the OEM's value for low limit temperature (this may range from about 38°F to about 50°F).
- What were the values for outdoor temperature, coil sensible heat ratio, B/C ratio. and low limit temperature?
- What were the values for the B/C ratio at AHRI rating conditions, the scenario's's outdoor temperature and coil sensible heat ratio, and the OEM's low limit temperature?
- If conditional capacity and limit temperature are modeled, it would have to be very hot outdoors (say 105°F), and the OEM's low limit value would have to be about 38°F to 40°F, for 40% to 50% bypass air with no limit trip at 350 Cfm/Ton.

15.7 Comment on Momentary EER

If air zoning is applied to single speed equipment (compressor and OSC blower), the momentary sensible capacity at the supply air outlets depends on supply air temperature and outlet Cfm, and the momentary equipment KW depends on many variables.

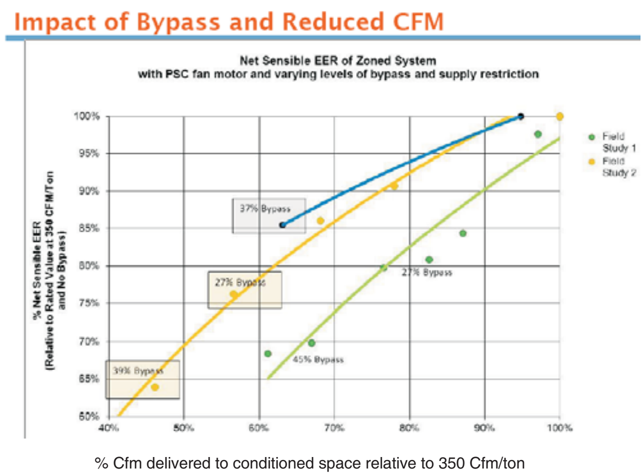
If we are going to compare single zone efficiency with multi-zone efficiency, wouldn't we want to integrate momentary power draw over cooling season time?

$$\Sigma \; \boldsymbol{KW_i \times HOURS}$$

In other words, if both systems provide comfort to the best of their ability (everything sized correctly), compare single zone KWH for the season with multi-zone KWH for the season.

Slide 16

Zoned Air Conditioning Data



16.1 General Comments

The title indicates that this is output from field tests. No information about the test procedure provided.

If this slide is consistent with Slide 8 (Sensible Capacity at the Register), EER is defined as supply air Btuh divided by system KW. **See Section 8.1 for comments on this practice.**

16.2 Understanding the Graph

What is this graph trying to say. Some questions provided here.

- Assume the blue line is for Field Study 3?
- Is 100% Cfm equal to the measured on-site blower Cfm with all zone dampers open, and the bypass closed?
- Shouldn't bypass air Cfm (per graph notes) plus Cfm delivered to the conditioned space (per x-axis label) equal 100%?

  What does "Relative to 350 Cfm per Ton" have to do with it?

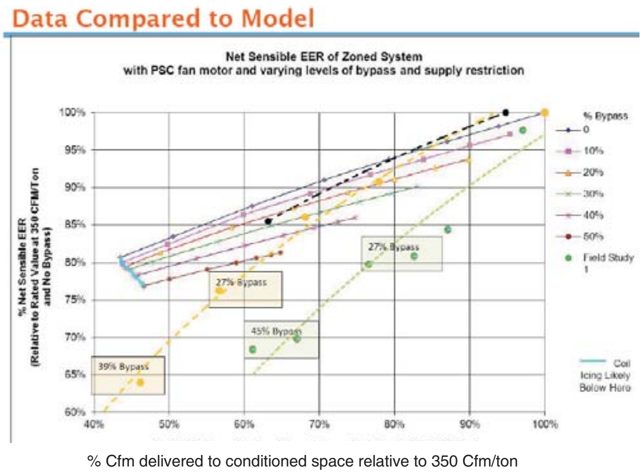
  Was the actual, measured on-site Cfm/Ton equal to, or different than 350 Cfm/Ton?
- There is a blue 37% bypass dot that shows 63% space Cfm and 86% EER; and an orange 39% bypass dot that shows 47% space Cfm and 64% EER.

  If bypass air is about 38%, why wouldn't the space Cfm be about the same for both cases (at 62%)?

  Same issue for the green 27% dot and the orange 27% dot.
- The graph shows the same general behavior as the model, but what else could happen if EER is defined as Btuh at the supply air outlets divided by input KW?
- All the comments for Slides 8, 9, 10, 12 and 15 apply here.

Slide 17

Zoned Air Conditioning



17.1 General Comments

The model and the tests are not compatible. Which one is correct?

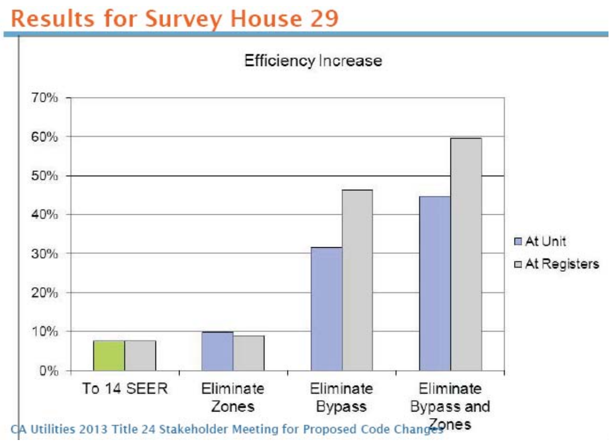
With no explanation for what slides 15 and 16 actually mean, it is not possible to comment on slide 17.

If this slide is consistent with Slide 8 (Sensible Capacity at the Register), EER is defined as supply air Btuh divided by system KW. **See Section 8.1 for comments on this practice.**

All the comments for Slides 8, 9, 10, 12 and 15 apply here.

Slide 18

Zoned Air Conditioning



18.1 General Comments

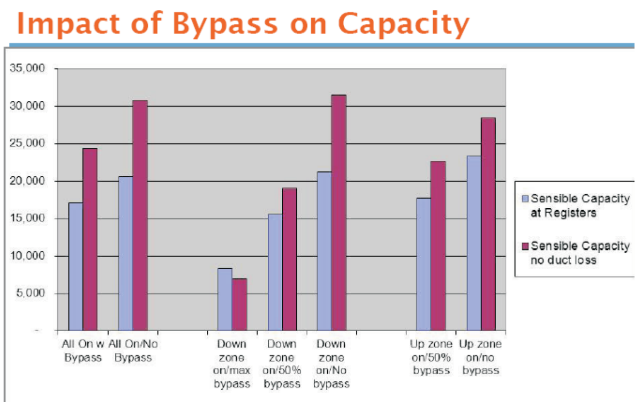
No explanation for the slides, so:

- How is efficiency defined.

  Seasonal or momentary?
  At unit = Total Btuh out / KW in?
  At registers = Sensible Btuh out / KW to unit?
- What does "To 14 SEER" mean? Is this the base case (single zone unit)?
- Wouldn't eliminating zones automatically eliminate Bypass?
- Do not know enough about the graph to comment.

Slide 19

Zoned Air Conditioning

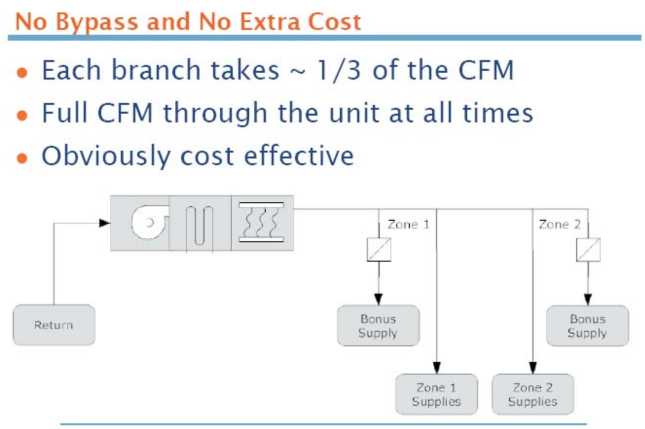


19.1 General Comments

No explanation for the slide, so it is not possible to comment on the slide. However; this is the first slide that says something about duct loss, so the complexity of the issues (and the explanation) are significantly increased.

Slide 21

Zonal AC System



21.1 General Comments

The Slide 21 solution (Figure 1) shows a duct system that has an undampered supply to each zone, and a dampered supply to each zone. There is no simple way to predict the behavior of this design.

- A zone may not need supply air, but some undetermined amount of supply air will be delivered to the zone (through the undamped supply) when the zone damper is closed.
- The slide says that 1/.3 of the air will flow to the zone that does not need the air (three ducts open, one duct closed, each duct at 1/3 of the blower Cfm).
- How is this going to happen? What are the duct sizing rules (for a particular blower table and set of pressure-dissipating devices, for a particular duct run geometry, and for a particular set of duct fittings); and/or what are the air balancing rules (the slide does not show hand dampers)?
- If a zone thermostat is satisfied, and if 1/3 of the air continues to flow to this zone, how long will it take for zone temperature to drop to an unacceptable level; and how does this transient compare to the time it will take ro satisfy the calling zone thermostat?
- Air outlet performance (throw and noise) depends on make, model and size. Some guidance is need here.
- In general, how can practitioners be sure that the proposed (Figure 1) design will always deliver adequate performance for any operating condition?

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Air-Conditioning, Heating, and Refrigeration Institute

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
aroy@ahrinet.org

Sacramento Zoned HVAC System Performance Test

Date: September 27, 2011

Performed By: Rick Chitwood
Chitwood Energy Management, Inc.
Office/Voice Mail (530) 926-3539
rick@chitwoodenergy.com

Summary

The goal of the testing was to quantify any loss of system performance due to the presence of a by-pass duct with a barometric damper. The test procedure used was developed as part of a recent California Energy Commission research project measuring the performance of new residential buildings and their HVAC systems. The procedure includes a measurement of sensible cooling at the equipment and the sensible cooling at the supply grilles (BTUs), and the energy consumption of the condensing unit and air handler (watts). This provides the actual operating efficiency and the actual delivered system capacity; both the efficiency and capacity are compared to the equipment manufacturer's performance data, to determine system performance.

Due to the complexity of the HVAC system in this home there was not enough time to collect the data required to quantify efficiency and capacity. The complexity included:

- Two return grilles
- Smart Vent, residential economizer
- 18 supply grilles; 8 downstairs, 10 upstairs
- Two stage furnace, 80%, 100,000/70,000 Btu/H
- Two stage condensing unit, SEER 16.0, 5 ton
- Two zones, upstairs/downstairs, with Honeywell HZ432 electronic system controller
- Bypass duct with barometric damper
- Honeywell electronic programmable thermostats, with Honeywell THM4000R wireless communication

Some data was collected on system performance, as were many observations and anecdotes.

The workmanship observed on this complex HVAC system installation was very high quality. I observed four technicians at this home, these technicians and their vehicles, are some of the best I have seen. Technician appearance, communication skills, response to customer questions, cleanliness and worksite care, and general skill level – were all top notch.

The system performance, though not thoroughly documented, appeared well below average. Below average performance can be attributed to the system complexity and the fact that equipment performance was not measured.

Measurements, Observations and Anecdotes

System Static Pressure

The system static pressure was measured in both the heating and cooling modes with both zones calling. The measured furnace external static pressure was 1.16" W.C. in the heating mode and 1.13" W.C. in the cooling mode. The manufacturer's maximum external static pressure rating is 0.80" W.C. This greater than allowed external static pressure causes low airflow and high fan watt draw, especially with an ECM motor, which this furnace has.

Commissioning Technician Only Measured Supply Plenum Pressure

Total external static pressure is a very helpful system performance metric and diagnostic aid. The commissioning technician only took one static pressure reading, in the supply plenum. Supply plenum static pressure is not a very helpful system performance metric.

Measured Airflow

With the system operating in the cooling mode, both zones calling, and with both thermostats set 10^{o}F below the room temperature, the airflow was measured at each supply grille. The sum of the supply grille flow is 1,352 CFM (270 CFM/ton, at 1.13" W.C. static pressure). Measurements were taken without the compressor running (dry coil). Target cooling airflow in this dry climate is over 500 CFM/ton with a wet coil.

I attempted to assure that the system was operating in the 2nd stage cooling mode by temporarily disconnecting the 1st stage control wire at the condensing unit, this would confirm 2nd stage operation as soon as the condensing unit turned on. The compressor never turned on even after calling for cooling for over 45 minutes. One of the technicians told me that it was the distance from setpoint that would start the 2nd stage cooling but that does not seem to be the case on this system.

This complex HVAC system has system controllers at five locations. Most of the controllers have installer programming options. System control functions reside in~~;~~ the two Honeywell thermostats, the Honeywell zone control board, the Z-Tech Smart Vent system control, the Rheem furnace, and the Rheem condensing unit. Complicated interactions can exist between these five controllers – which may explain 2nd stage cooling not turning on within 45 minutes.

High Fan Watt Draw

The measured fan watt draw for this system (at the above measured airflow) was 949 watts (or 0.70 watts per CFM). The California standard for fan watt draw is 0.58 watts per CFM. This high fan watt draw (with both zones calling) is caused by high static pressure and the presence of a bypass duct that bypasses some air even with both zones calling. High fan watt draw drives up operating costs and lowers system efficiency.

Commissioning Technician Could Not Measure Airflow

I asked the commissioning technician what the system airflow was and he did not know or have the test equipment (capture hood or flow plate) to measure the airflow.

Commissioning Technician Not Concerned About High Superheat

The commissioning technician did not have subcooling or superheat on his list of system performance tests, even though they are important performance metrics. High side and low side refrigerant pressures were on the technicians list of measurements. While he was measuring pressures, I asked him to also measure the refrigerant temperatures so subcooling and superheat could be calculated. The subcooling was 9.8^{o}F, within the normal range for most equipment. The superheat was 21.6^{o}F, well above the typical range of 5^{o}F to 10^{o}F. The high superheat did not seem to be a concern of the commissioning technician.

Low Supply Grille Velocity

Many of the supply grilles have low delivery velocity. Three things should be happening at each supply grille: 1) the grille must deliver the BTU's (CFM and delta-T) to meet the loads for the room; 2) deliver air at a high enough velocity to mix the room air and minimize stratification (usually 500 FPM to 700 FPM); and 3) deliver air so that it does not create drafts on the occupants. The living room grille in this home is a large 14" x 8" one direction curved blade grille with an airflow of 161 CFM. The delivery velocity is only about 250 FPM, or about 40% of the velocity required for good room air mixing. This size of a grille would deliver 377 CFM if it had a delivery velocity of 600 FPM, our typical design velocity. Delivering air at a low velocity contributes to low comfort levels in the home due to temperature stratification – especially in the heating mode. Hot air delivered at a low velocity at the ceiling of the first floor rises to the ceiling of the second floor due to its buoyancy, making the second floor too hot, and leaving the floor level of the first floor too cold.

Furnace Cycles Off on High Limit

In an attempt to warm up the house a little before switching to the cooling mode test the system was operated in the heating mode with both zones calling and the thermostats set 10^{o}F above the room temperature. During this call for heat, the furnace burners cycled off repeatedly. I assume this was a high temperature limit shutdown due to low airflow and some bypass air (even though both zones were calling). Cycling on high limit reduces system efficiency and can shorten the life of the furnace's heat exchanger.

Return Grille Noise

The return air grille was very noisy. The connection between the furnace and the return grille is short (about 3.5 feet) rectangular sheet metal duct.

Commissioning Technician Thought a Single Stage Condensing Unit Was Installed

When I first arrived, I asked the technician if the SEER 16 condensing unit had a one-stage or a two-stage compressor – he said one-stage – but it actually had a two-stage compressor.

Evaporator Condensate Leak Repair

One technician was working on a condensate-handling problem on the evaporator coil condensate pan. The technician was not sure if the condensate pan was leaking, the drain-pipe had a leak, there was condensate blow-off (the pan is not catching all of the water due to the high air velocity), or if the drain-pipe vent was not large enough. The technician focused in the drain-pipe vent size, even though there should not be a vent in the pipe and that could not have anything to do with this leak.

<u>Conclusion</u>

This HVAC contractor is clearly a great contractor; with a good business model and top-notch technicians, and is nationally recognized by AHRI as a company that is doing zoning correctly. It is surprising that the system performance characteristics measured show below average system performance, especially since statewide average system performance is already so low. The primary reason for the low performance is system complexity, and the lack of testing of the system performance factors.

The ASHRAE Handbook of Fundamentals tells us that we should not design or specify systems that require exceptional skill or workmanship from the workforce available. Following this logical concept would preclude the installation of systems of this complexity.