DOE 2.2 - Work-Arounds to Hidden Problems:

eQUEST - A half hour to learn;
three years to master.

Building Energy Simulation Forum (BESF)

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Announcing: Final draft available

PECI™
Energy for Change™

Demand Controlled Ventilation (DCV) Measure Analysis Guide

covers:

DCV Modeling of
- Movie Theaters
- Meeting/Lecture
- Classrooms
- Gym/Fitness
- Retail

Includes prototype eQUEST models

Research Funded by:

Bonneville Power Administration

Final Draft available at:
http://energytrust.org/Business/building-energy-simulation/

Final soon available at: www.peci.org & www.bpa.gov/energy/n/commercial.cfm

DOE 2.2 – Work-Arounds to Hidden Problems: eQUEST - A half hour to learn; three years to master.

Learning Objectives:

- Understand correct modeling of outside air economizers for single-stage dX cooling
- Distinguish commercial ventilation requirements and modeling: *What to do if you aren't in California!*
- Understand DOE 2 limits and proper modeling of unoccupied fan operation
- List issues for proper VFD fan modeling
- Know how to model DCV

Understand Correct Modeling of Outside Air Economizers for Single-Stage dX RTUs

DOE 2.2 – Work-Arounds to Hidden Problems:
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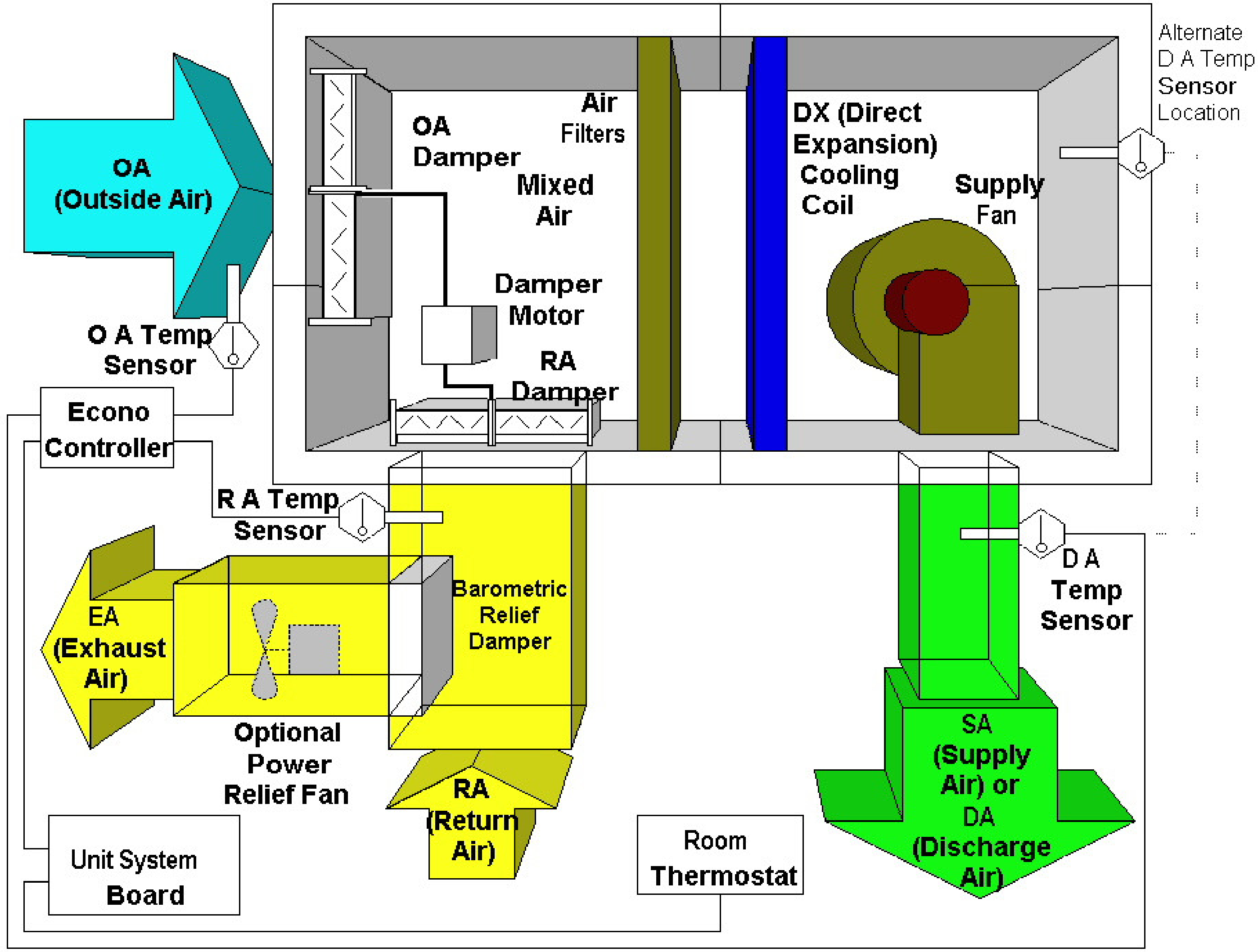
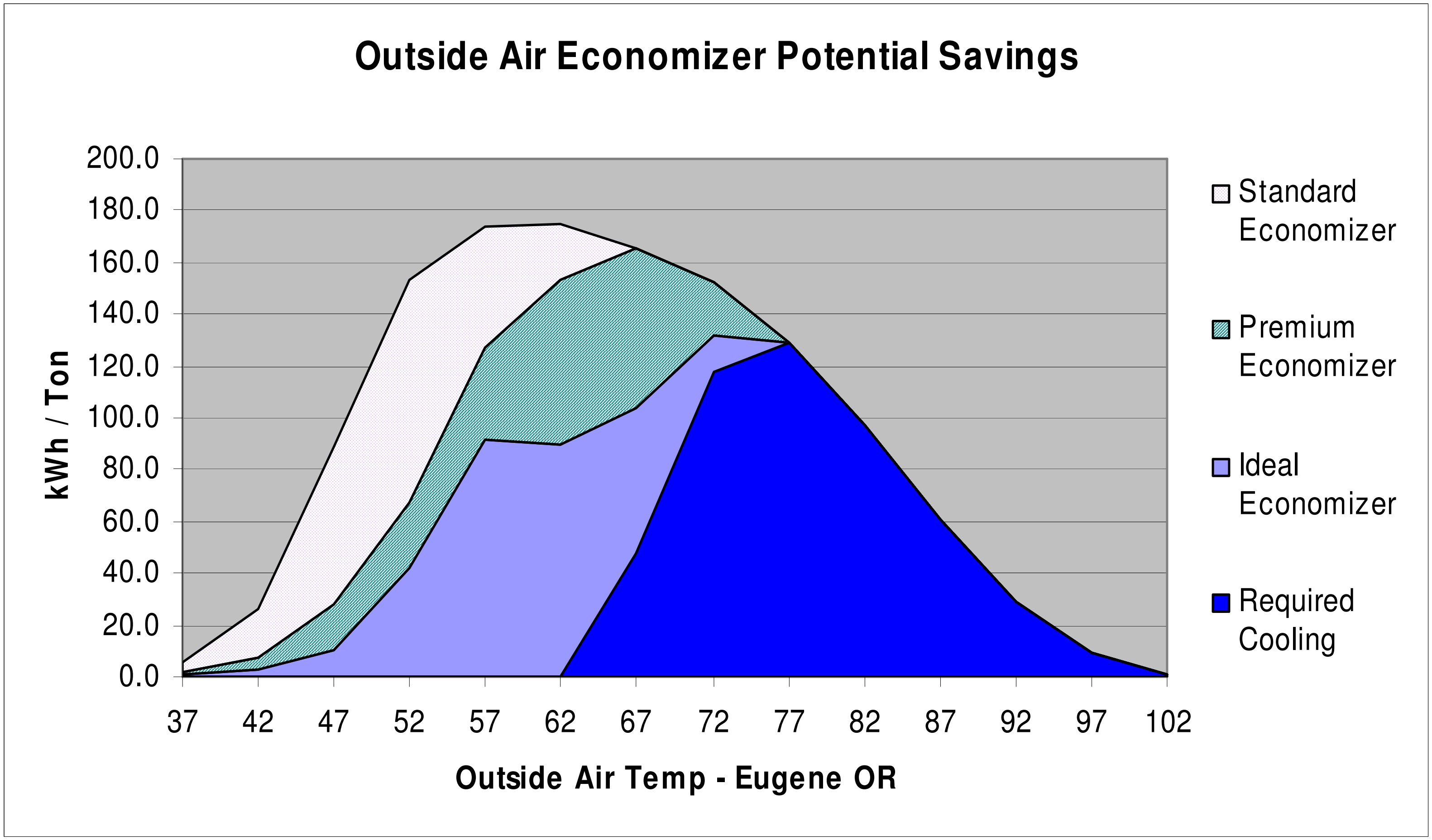
Basic OSA Economizer Idea

Packaged Unit (DX cooling) Outside Air Economizer



Savings from Economizer Levels



Ideal economizer does not happen in RTUs.

Get most of savings with alternating integration.

Integrated Economizer for RTUs

- Thermostat with dedicated stage for economizer.
  - Single stage at thermostat requires "either/or" operation with low (55 F) changeover setting. Result: very little savings.
  - Dedicated first stage on thermostat allows economizer to operate before compressor.

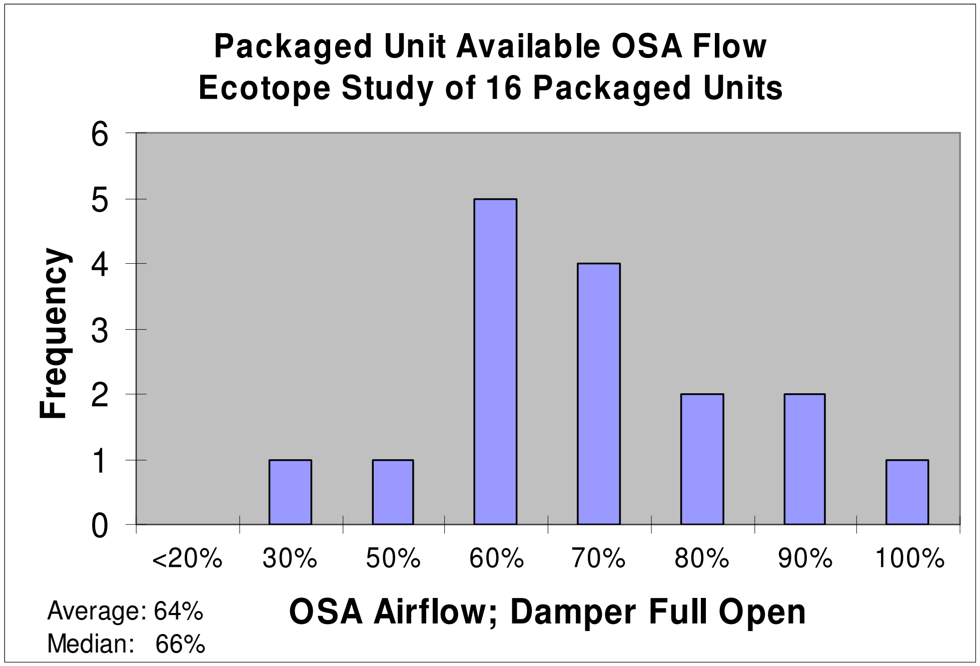
- For dX cooling with minimal stages; must locate primary control sensor in discharge air, not mixed air position.
  - Mixed air sensors notoriously inaccurate (poor mixing)
  - Otherwise DA too cold, economizer will be disabled

Three specific economizer modeling issues:

- Maximum outside air is usually less than the 100% default included in eQUEST
- The minimum outside air position is often greater than required
- Single zone packaged units with single-stage dX cooling coils operate in alternating integration while DOE 2.2 PSZ system incorrectly models full integration

Max OA Fraction

- Code may call for 100%; do you get it?



- 70% is more reasonable for RTUs without exhaust fans

Suggested Max_OA_Fraction

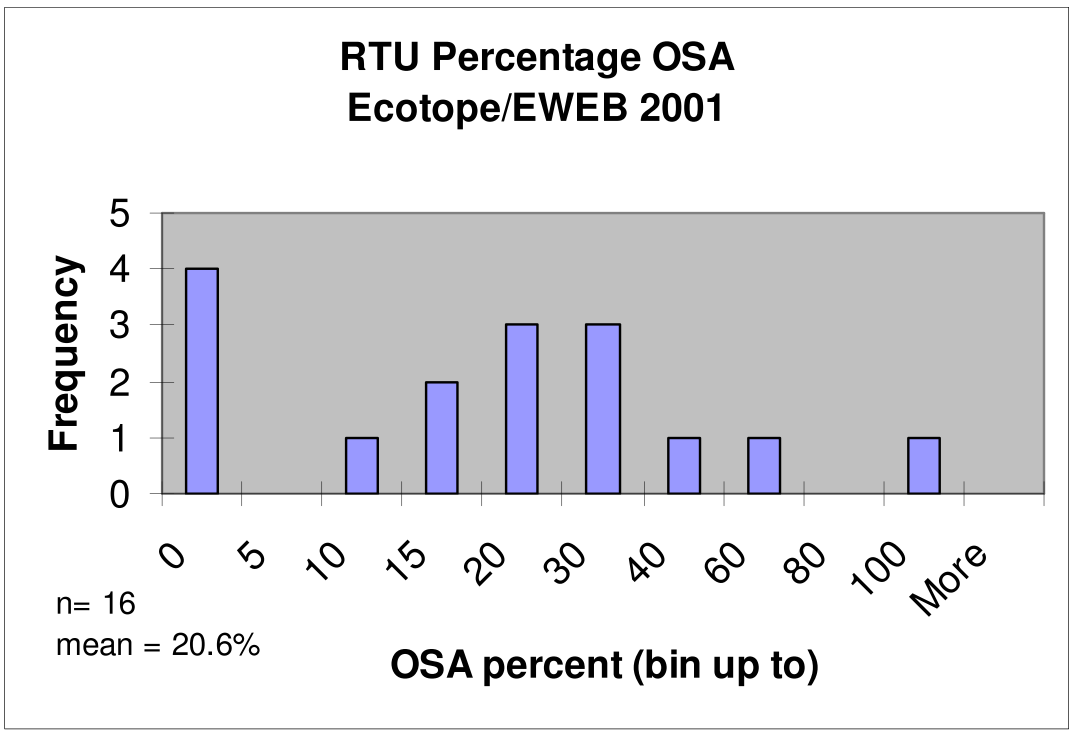
OA fraction improved with:

- Barometric relief damper
- Motorized exhaust damper
- Seals and good closure on return damper
- Powered exhaust fan
- Return fan with return damper seals

Configuration	MAX_OA_FRACTION
No relief damper	0.50
Barometric relief damper (most common)	**0.55 to 0.75 (0.70 suggested)**
Motorized relief/exhaust damper	0.60 to 0.80 (0.75 suggested)
Good seals on return damper	0.75 to 0.85 (0.80 suggested)
Powered exhaust fan activated at 50% OSA	0.80 to 0.90 (0.85 suggested)
Return Fan with good seals on return damper	0.80 to 0.90 (0.85 suggested)

Minimum Ventilation Rate

- MIN-OUTSIDE-AIR at system level
- ZONE keywords take precedence
  OUTSIDE-AIR-FLOW, OA-CHANGES,
  OA-FLOW/PER or EXHAUST-FLOW
- Office typically 7% to 13% required; Actual ~20%

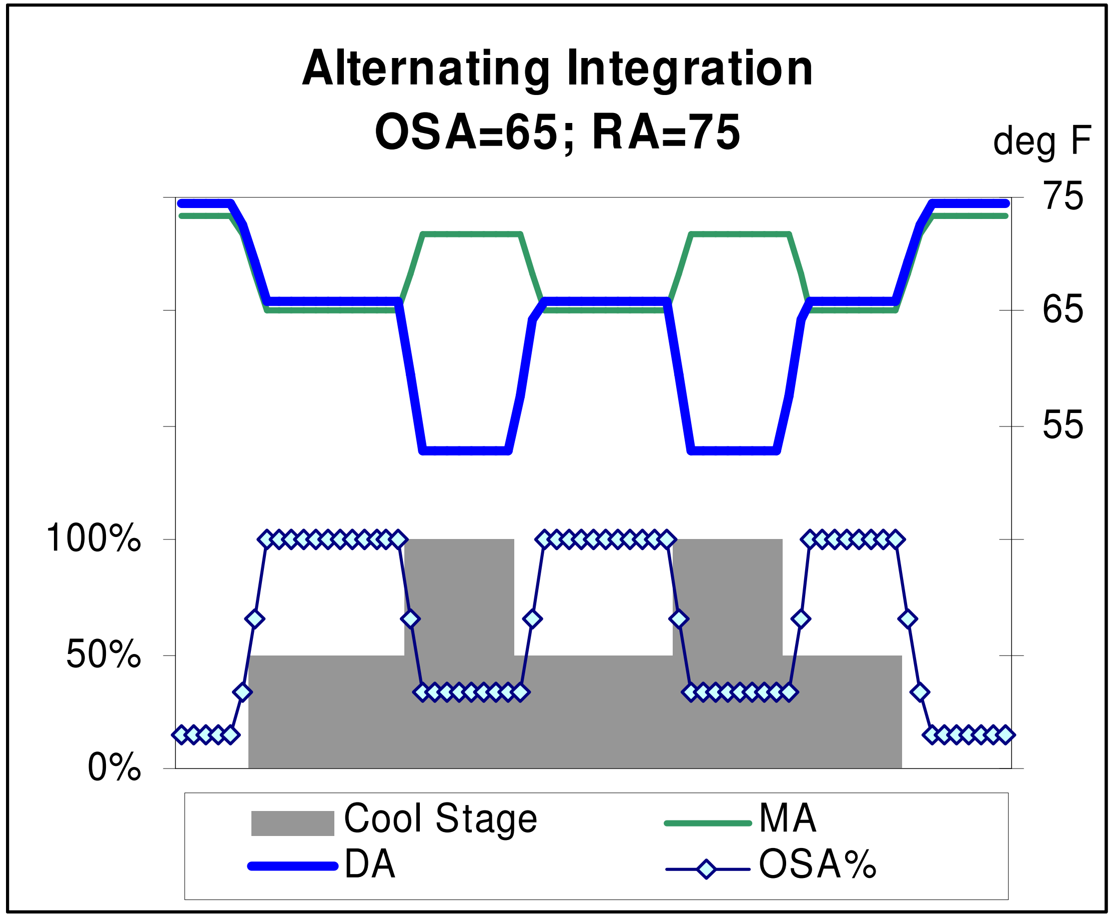


Baseline?

- Use 20%? May be appropriate for low density buildings
- Prototypes use ASHRAE 90.1 area + 50% default people base case; matching most codes

Alternating Integration Fix

- Rtus (PSZ) with staged dX do NOT have full integration
- DOE 2 says they do
- Economizer must throttle back when dX is on . . . Why?
- Need to model all modes during each hour
  - Economizer only
  - Econo + Stage 1 dX
  - Econo + Stage 2 dX
  - Min OA + Stage 1 dX
  - Min OA + Stage 2 dX



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OSA High Limit or "changeover" sequence

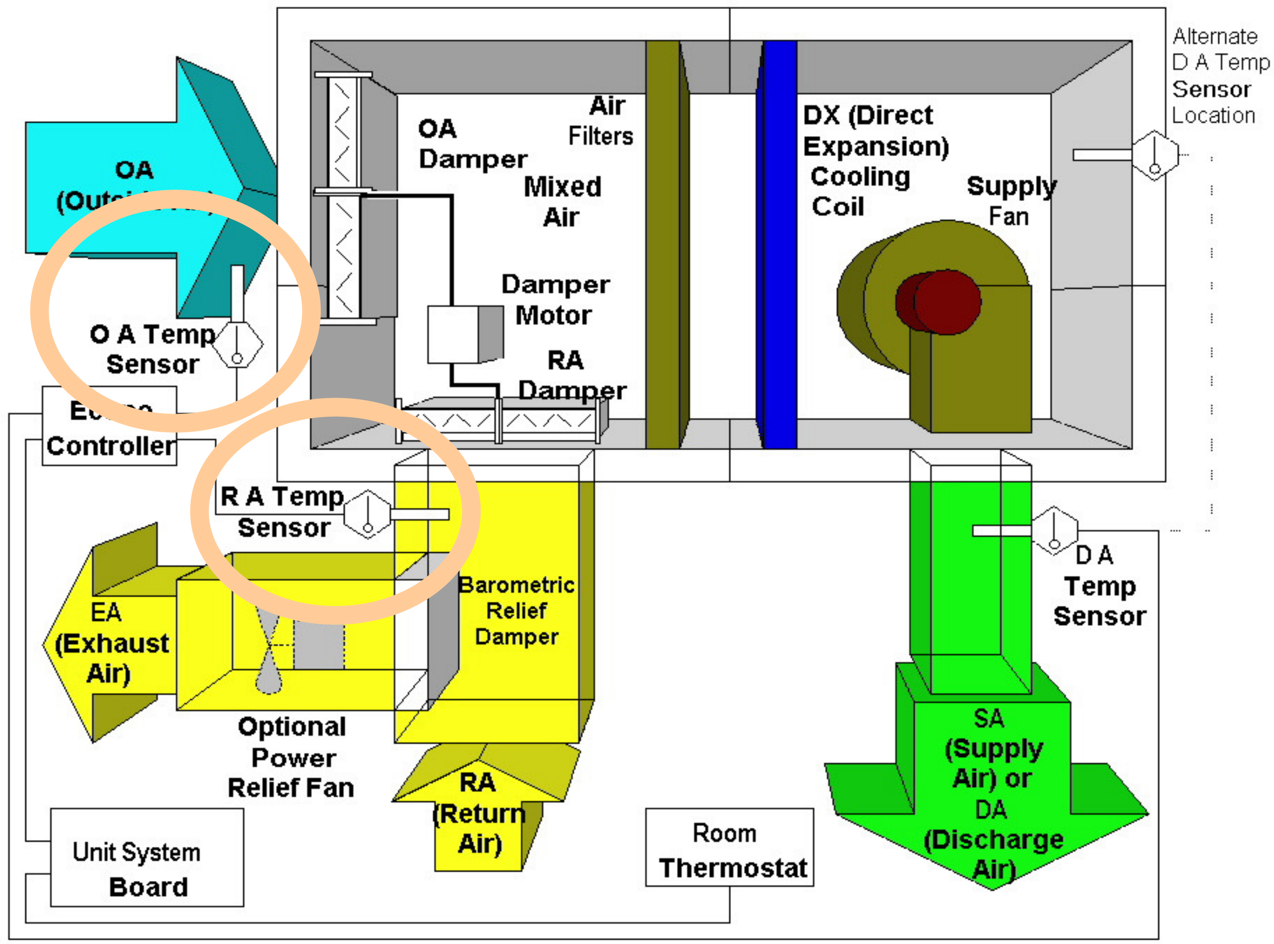
When is it too hot to use economizer?

- Mode of Control
  - Single (OSA only) fixed (snap disc)
  - Single (OSA only) adjustable (analog)
  - Differential or Comparative (OSA vs. RA)
- Type of Sensor
  - Dry-bulb measures temperature only
  - Enthalpy adjusts for humidity
  - Separate dry-bulb and humidity sensors

Dry-Bulb Differential Changeover

- Dry-bulb analog OA & RA sensors
- Differential changeover logic
  - Both OA & RA sensors pprovided
  - NO snap discs
- Single changeover OK with aggressive setpoint

Packaged Unit (DX cooling) Outside Air Economizer

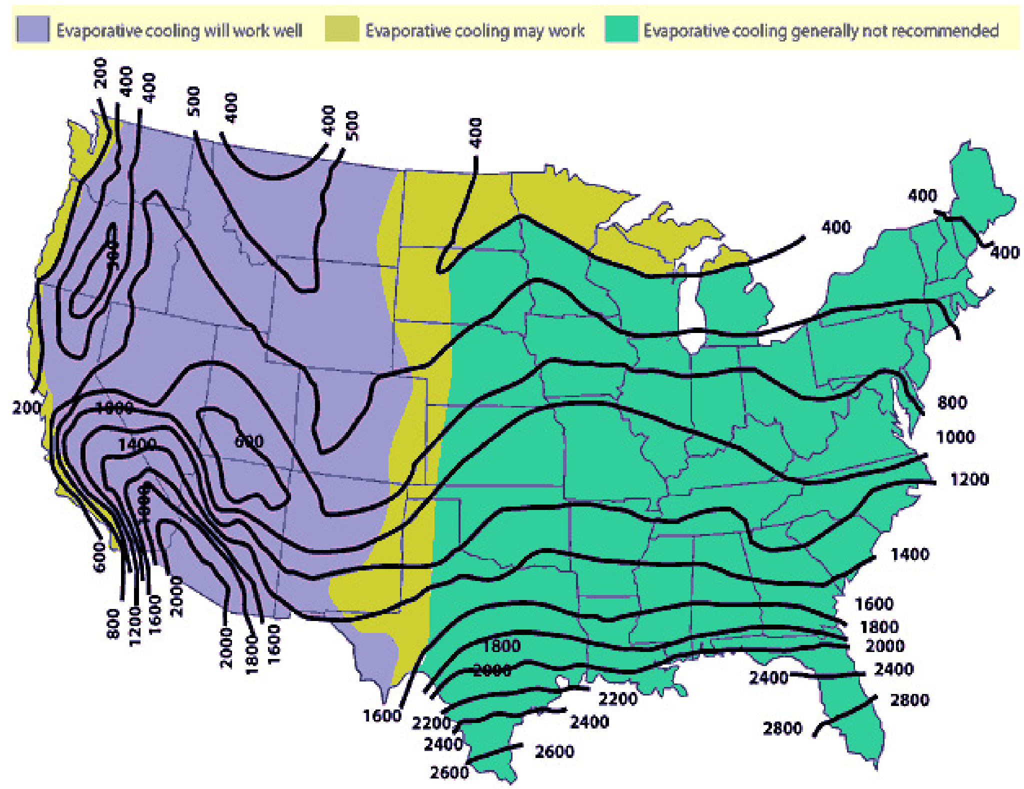


Differential Changeover

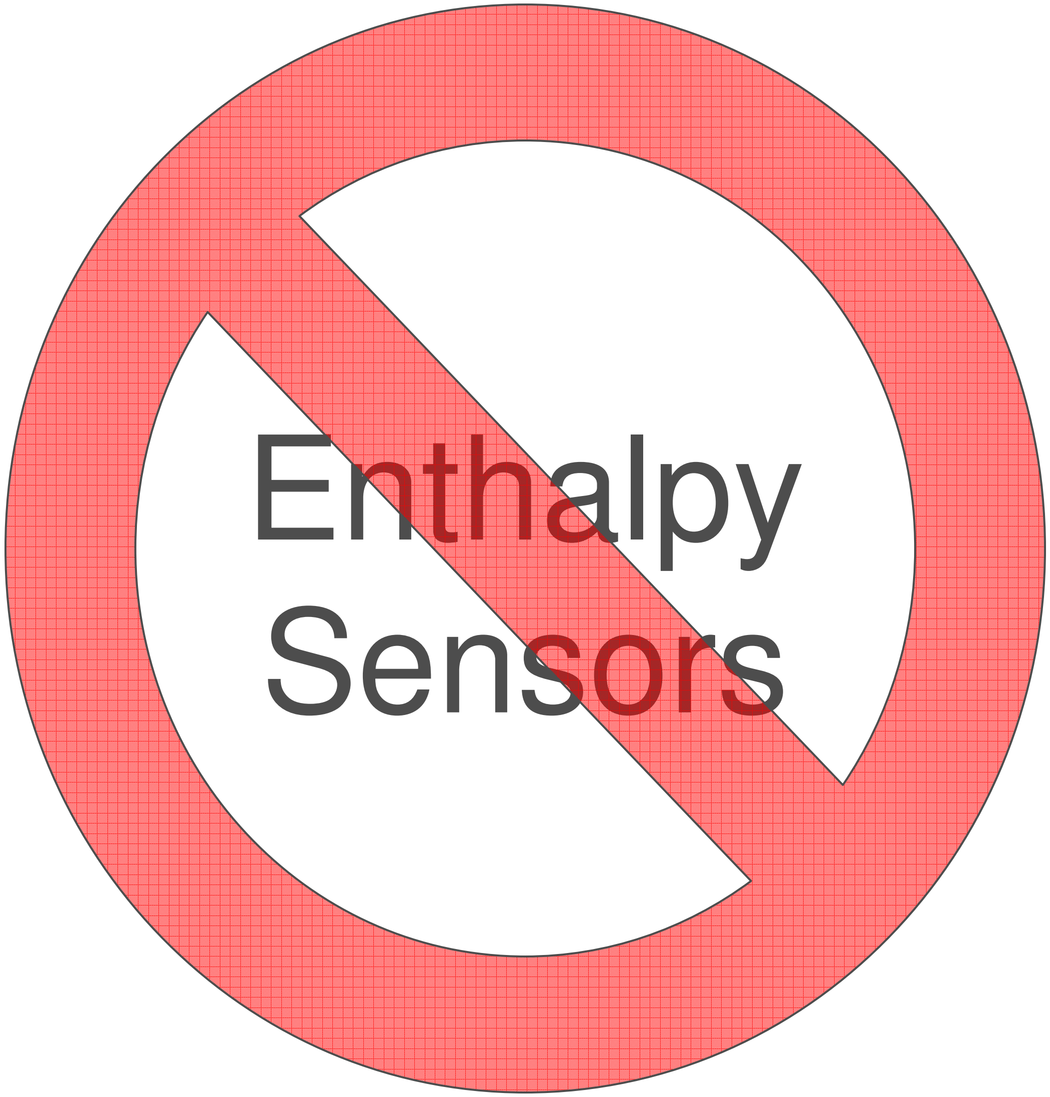
- Differential or Comparative Changeover
  - RA & OSA sensor
  - Avoids confusion; no guesswork
  - Adapts to loads; avoids callbacks
- How it works
  - Compares temperature or energy content (enthalpy) of both RA & OSA
  - Allows economizer if OSA is cooler or has less enthalpy or heat energy depending on sensor
  - Maximize integration or time both the economizer can function with cooling

Dry-bulb Changeover in the West

- ARI map shows where evaporative cooling works.
- Economizer climates in West (ASHRAE 90.1):
  - 95% dry
  - 5% intermediate
  - 0% humid
- Changeover Sensor:
  - West: dry-bulb
  - East: enthalpy



Map of Summer Cooling Load Hours
Reprinted from pages 16 and 17 of the ARI Unitary Directory, August 1, 1992 - January 31, 1993
Courtesy of the Air-Conditioning and Refrigeration Institute, ARI.



See Iowa Humidity Sensor Testing: NBCIP *PTR: Duct Mounted Relative Humidity Transmitters.* April 2004; Rev. November 2004

www.energy.iastate.edu/Efficiency/Commercial/download_nbcip/PTR_Humidity_Rev.pdf

Modeling dX Cooling Economizer in eQUEST

pecl™

- DOE 2.2 will overstate economizer energy savings for Packaged Single Zone (PSZ) and PVVT systems using single stage DX cooling coils.
- DOE 2.2 models a fully integrated economizer strategy instead of actual alternating integration.
- In actuality, a single-stage dX cooling unit must throttle back the outside air during integrated operation.
- In order to simulate an alternating economizer strategy in DOE 2.2 a work around has been developed.
  - Adjust the high limit to reduce economizer savings
  - Described in Appendix A of DCV modeling guide

High Limit Adjustment – Low Density Occupancies

Adjusting DOE 2.2 PSZ from full integration to alternating integration

Low Density Occupancies such as offices

OAT	Adjusted High Limit Input, °F			Reduction in High Limit, °F		
Balance:	57	52	47	**57**	**52**	**47**
OAT High Limit	**Light**	**Med**	**Heavy**	**Light**	**Med**	**Heavy**
75.0	71.3	69.6	68.6	**3.7**	**5.4**	**6.4**
72.5	70.2	69.2	68.2	**2.3**	**3.3**	**4.3**
70.0	69.1	68.2	67.2	*0.9*	*1.8*	**2.8**
67.5	66.5	65.7	64.8	*1.0*	*1.8*	**2.7**
65.0	64.5	64.2	63.7	0.5	*0.8*	*1.3*
62.5	62.1	61.7	61.2	0.4	*0.8*	*1.3*
60.0	59.8	59.6	59.3	0.2	0.4	*0.7*
57.5	57.5	57.1	56.7	0.0	0.4	*0.8*
55.0	55.0	54.8	54.5	0.0	0.2	0.5

Internal loads are characterized as light, medium and heavy.

Heavy: Lighting at 2.3 Watts/square foot with high occupancy; Call center

Medium: Lighting at 1.7 Watts/square foot; moderate occupancy; open office

Light: Lighting at 0.7 Watts/square foot with low density occupancy

High Limit Adjustment – High Density Occupancies

Adjusting DOE 2.2 PSZ from full integration to alternating integration

High Density Occupancies (with increased ventilation)

OAT	Adjusted High Limit Input			Reduction in High Limit		
Balance:	52	47	37	**52**	**47**	**37**
Hi Limit	Light	Med	Heavy	Light	Med	Heavy
75.0	70.7	69.5	67.8	**4.3**	**5.5**	**7.2**
72.5	69.8	69.1	67.9	**2.7**	**3.4**	**4.6**
70.0	69.1	68.4	66.4	*0.9*	*1.6*	**3.6**
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55.0	54.9	54.7	54.1	0.1	0.3	*0.9*

Internal loads are characterized as light, medium and heavy.

Heavy: Retail with high lighting or appliance and people density

Medium: Moderately full classrooms, meeting rooms, and lecture halls

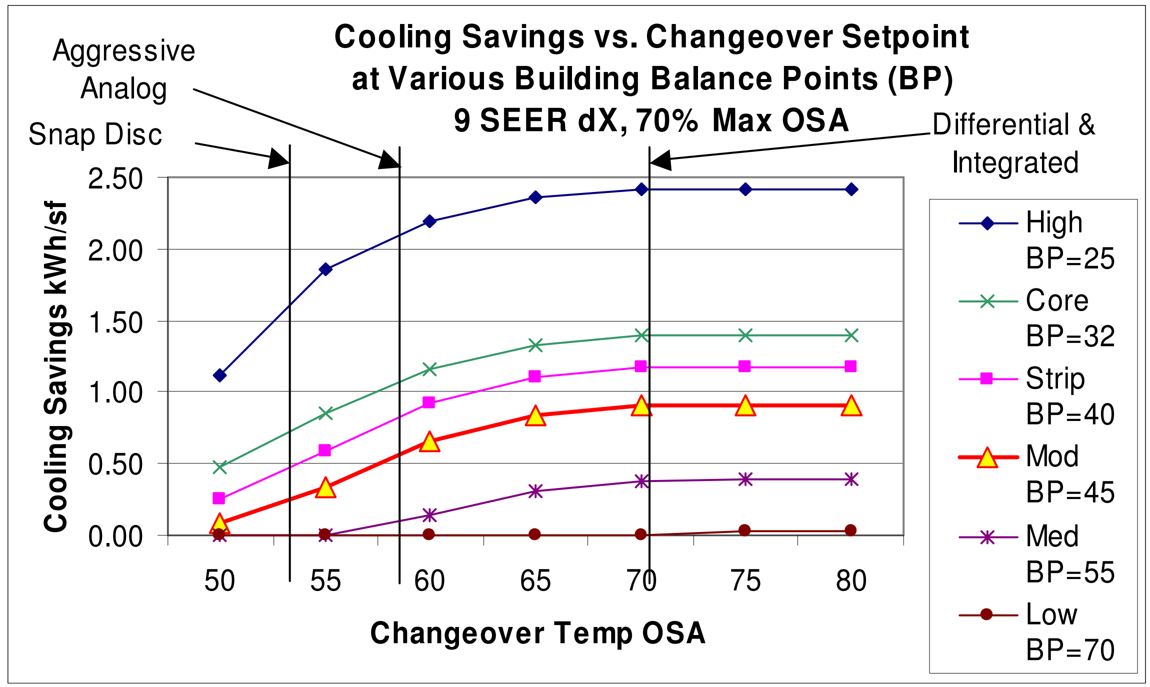
Light: Theatre or assembly with intermittent occupancy, low light levels

Development of Economizer High Limit Adjustment Values

- Use a simplified bin method to find:
  - Bin cooling loads and occupied hours
  - OSA % for 53°F DAT with mechanical cooling
  - Time of economizer only operation in each bin
- Percentage of full integrated economizer delivered by alternating integration
- Match those percentage reductions in savings to parametric DOE2 runs.

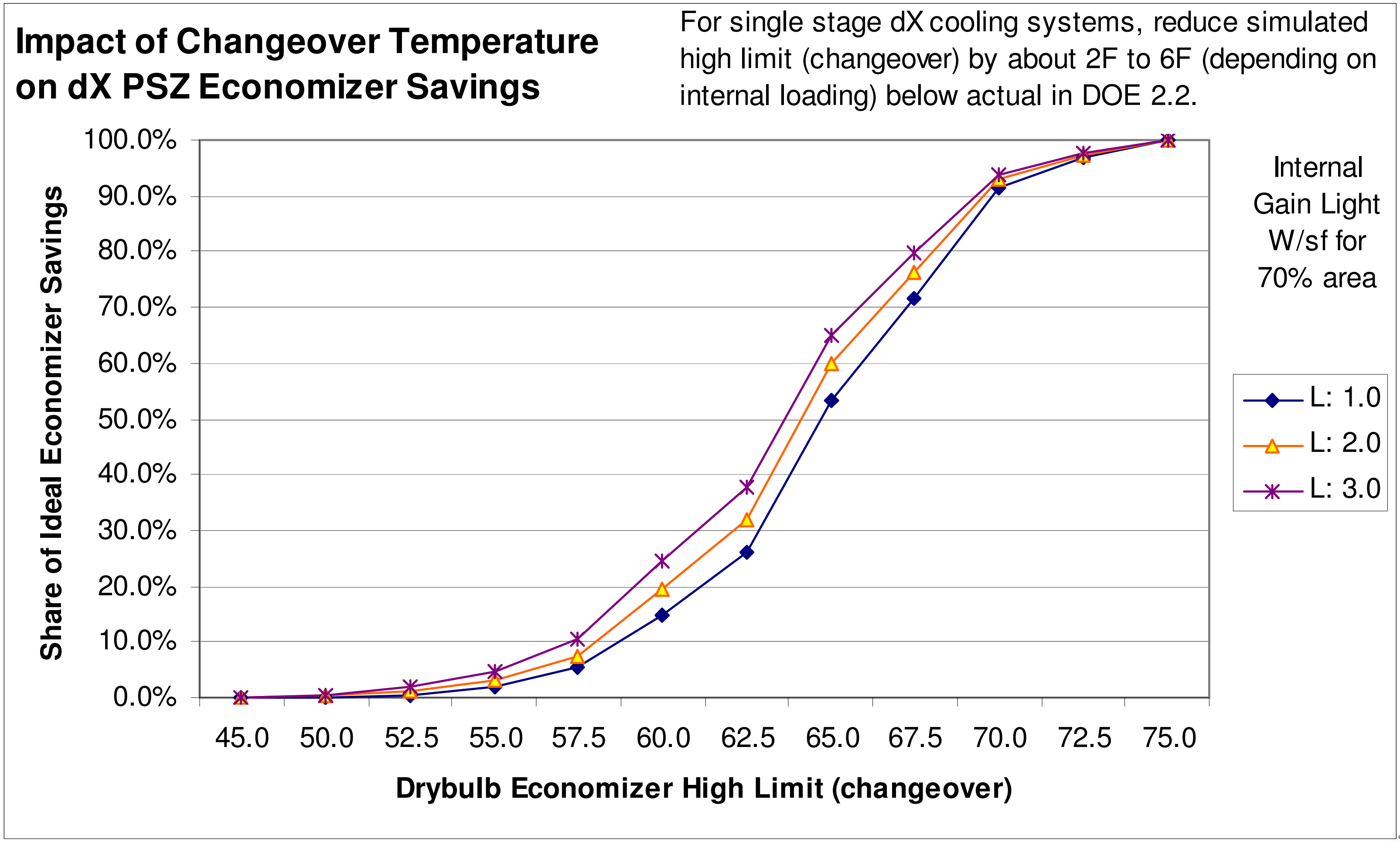
Integration Increases Savings

- Cooling load:
  - +Internal gains
  - + Solar
  - - Heat loss
  - - Ventilation
- Balance Point: cooling not needed
- Internal gains are reducing
  - – Efficient lights
  - – LCD monitors

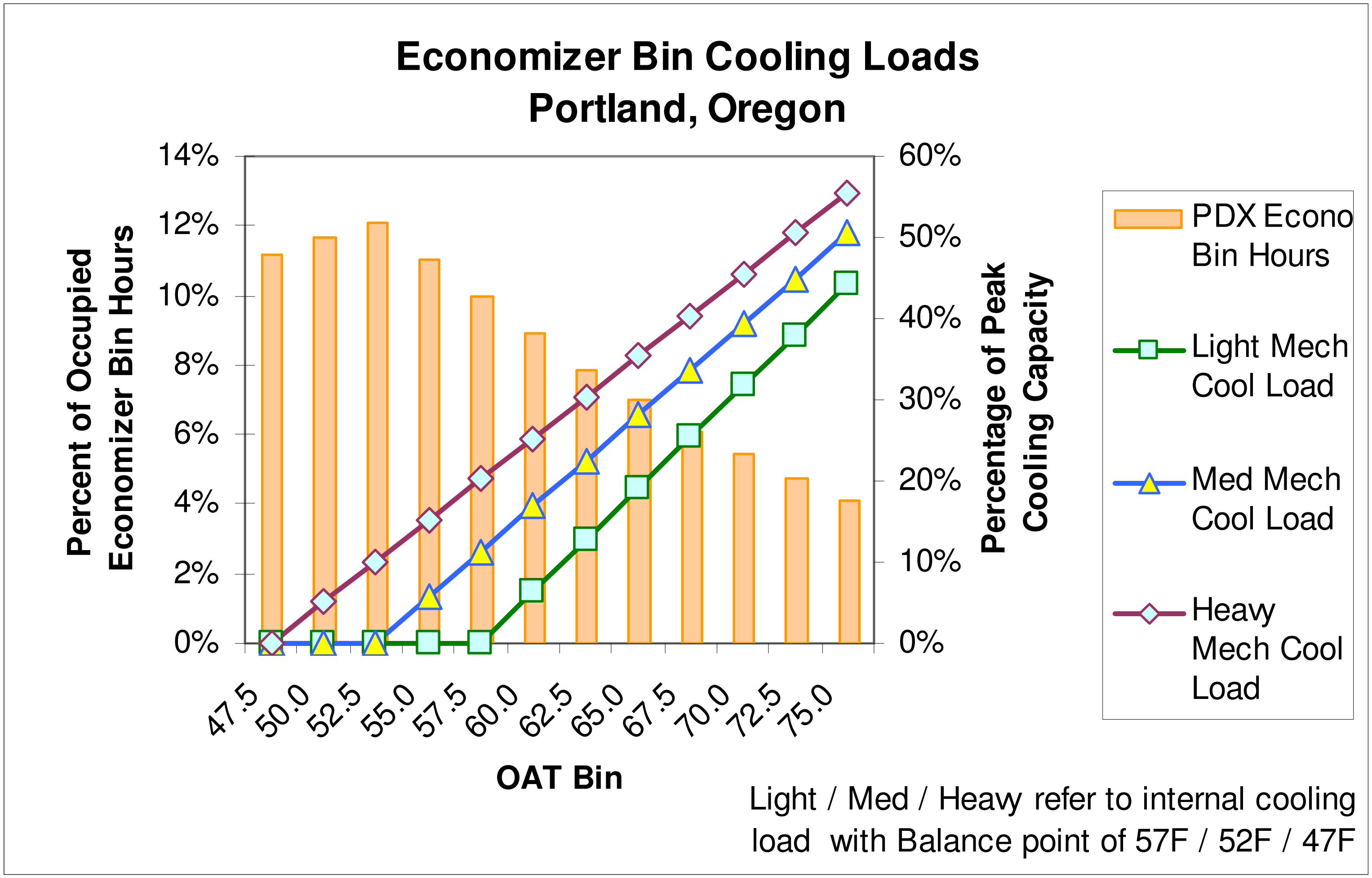


- Save more with higher OSA changeover
  - – High changeover requires integration
  - – Little savings at snap disc changeover points

Impact of Changeover Temperature on % dX PSZ Economizer Savings



Portland Cooling Loads in Economizer Range



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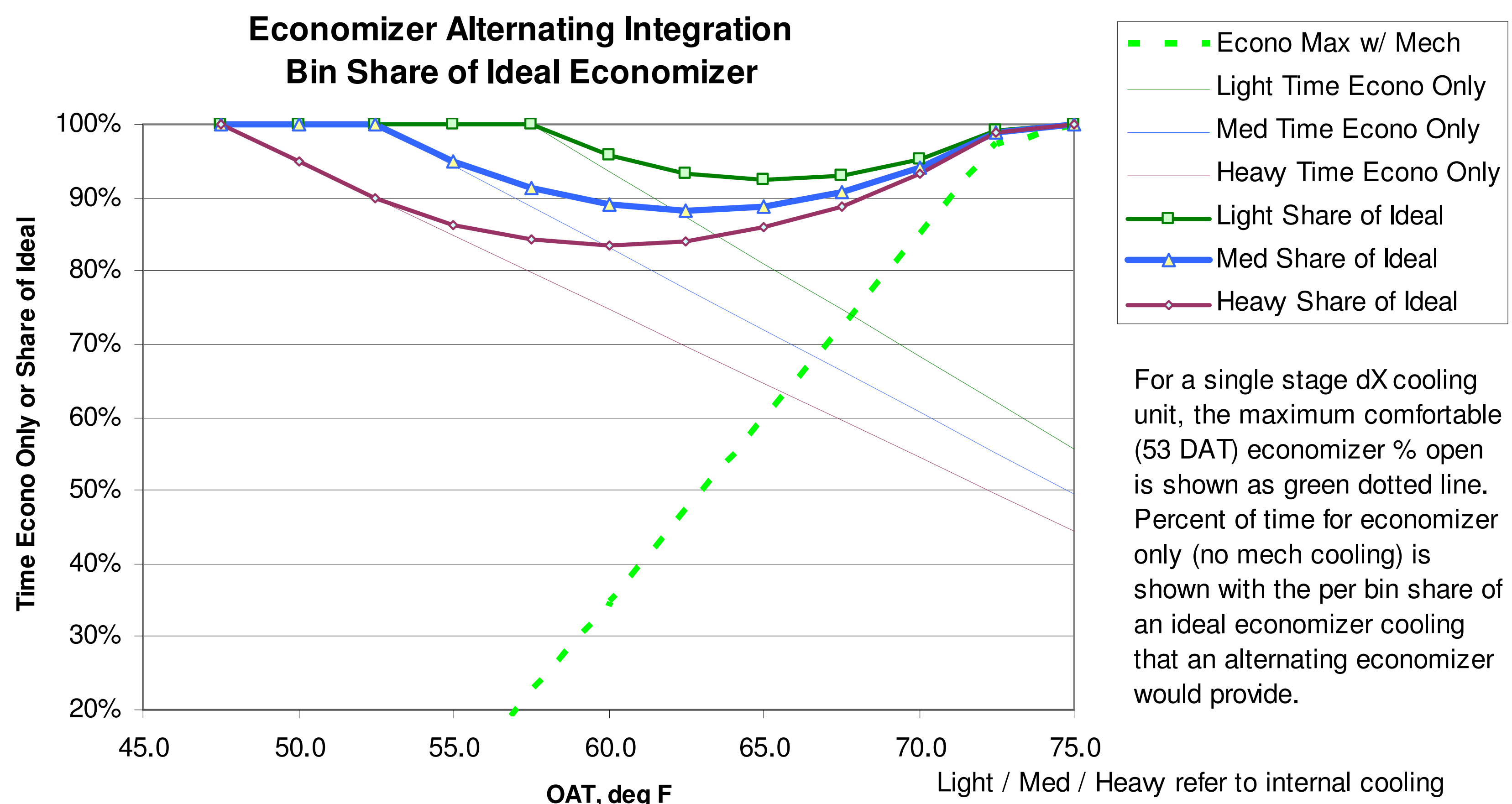
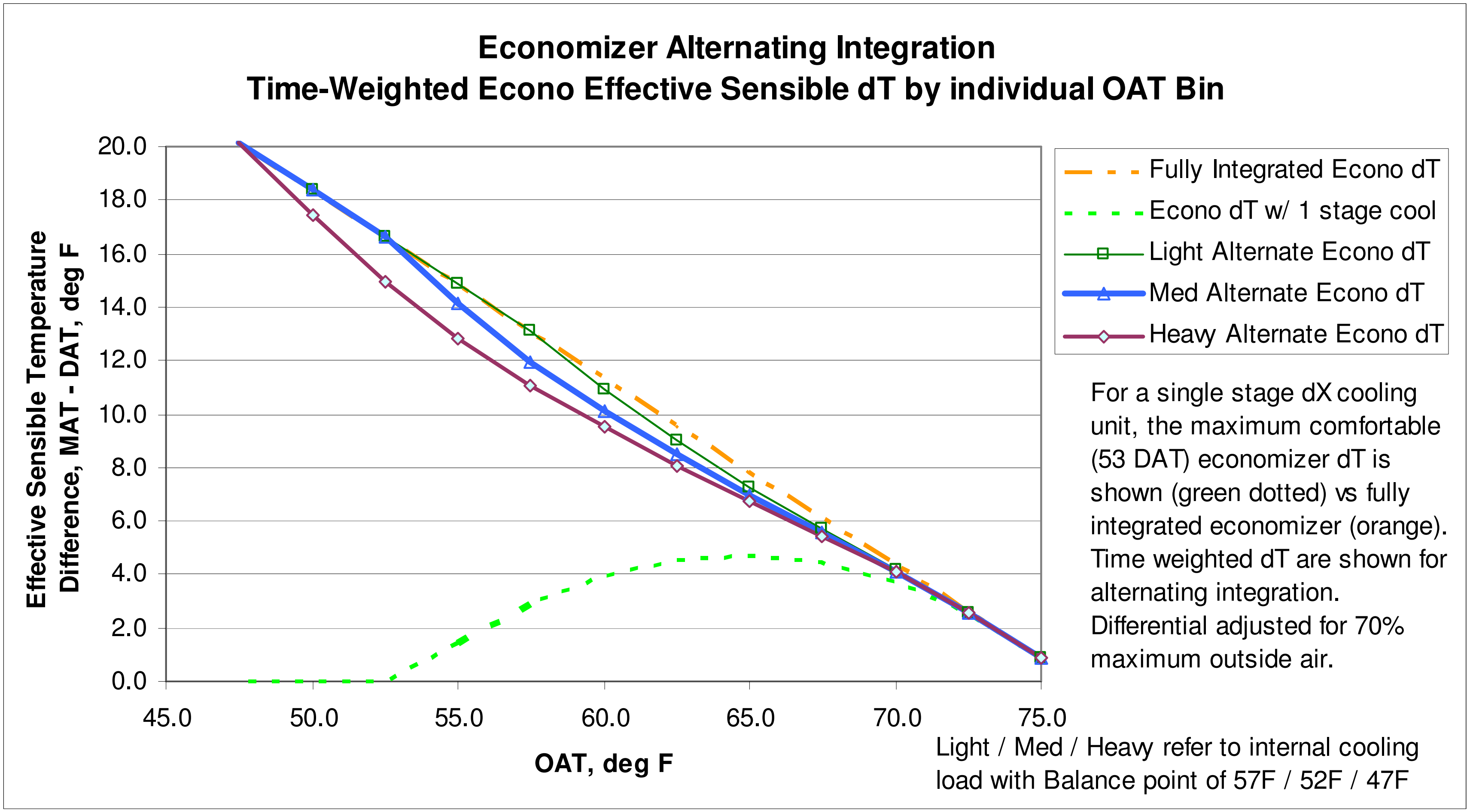
Economizer Time and Share of Ideal by OAT Bin



For a single stage dX cooling unit, the maximum comfortable (53 DAT) economizer % open is shown as green dotted line. Percent of time for economizer only (no mech cooling) is shown with the per bin share of an ideal economizer cooling that an alternating economizer would provide.

Light / Med / Heavy refer to internal cooling load with Balance point of 57F / 52F / 47F

Economizer Time–Weighted Effective dT by OAT Bin



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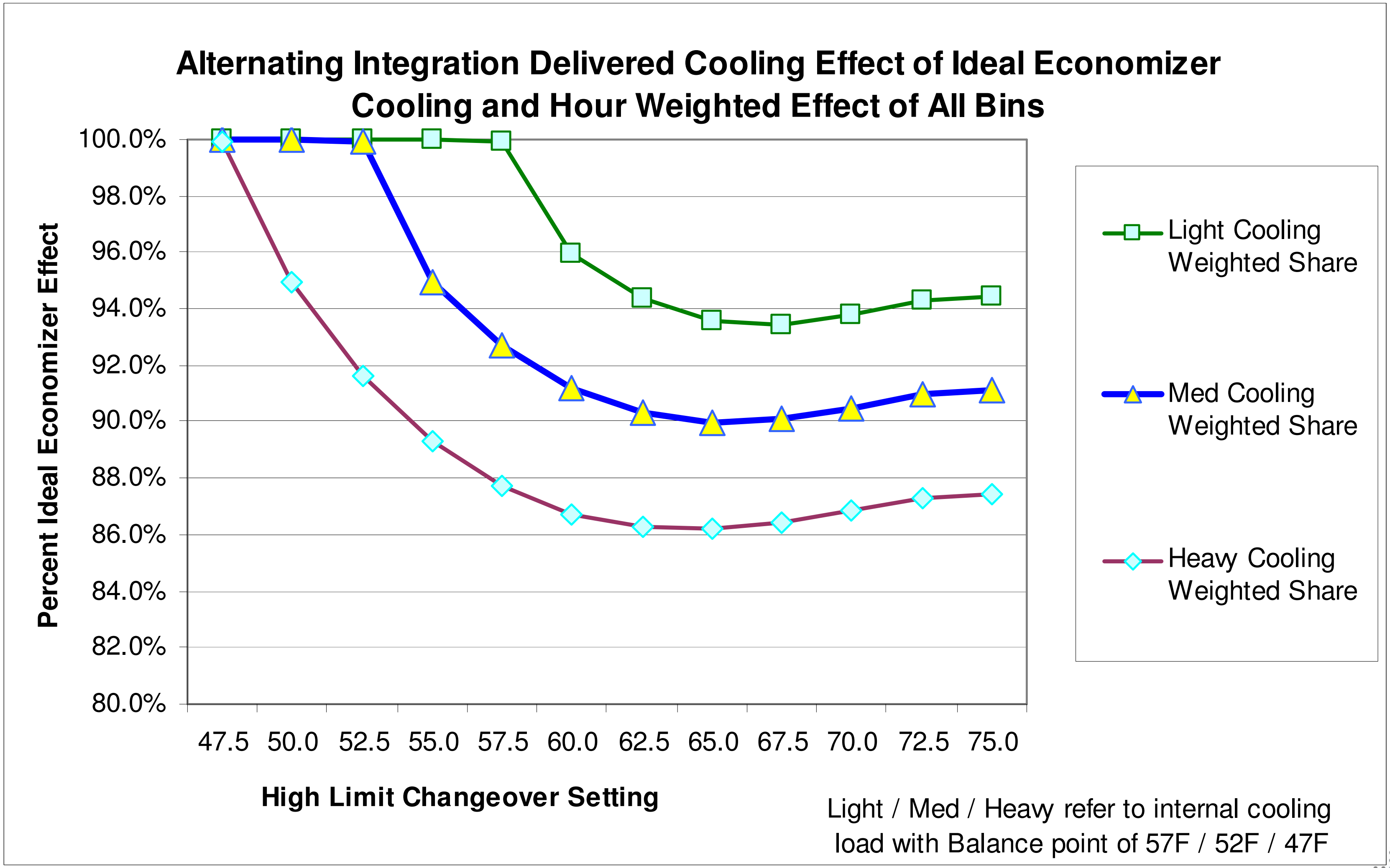
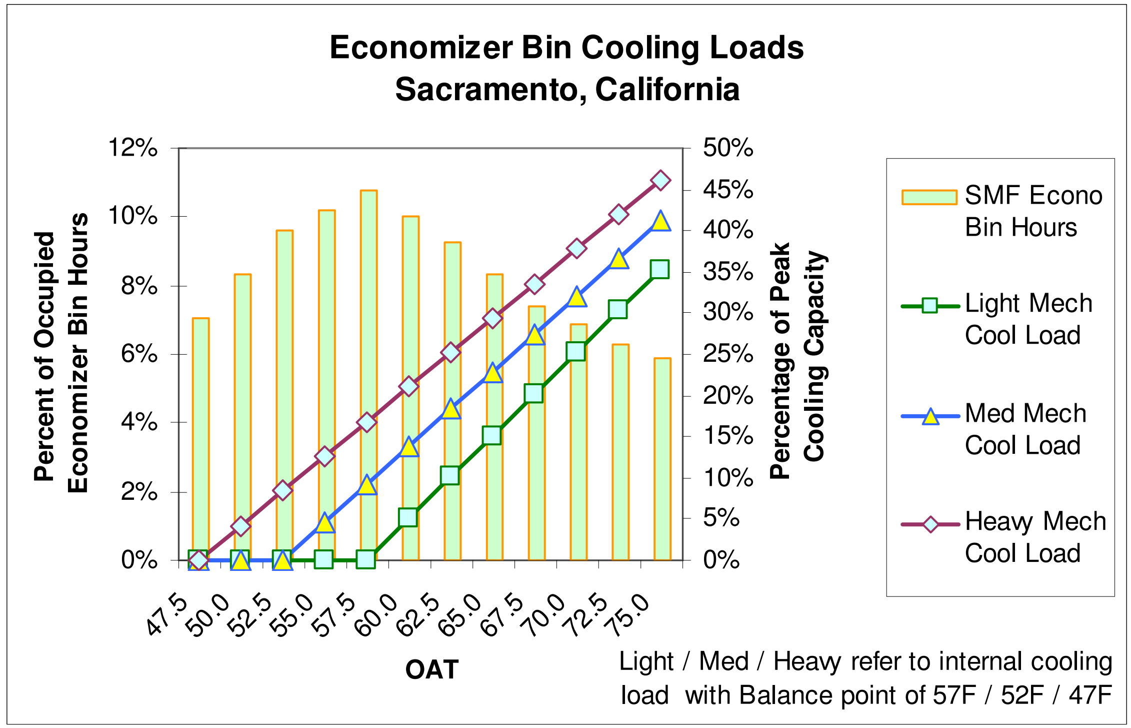
Economizer Delivered Cooling Sensible Effect by OAT Bin



Sacramento Cooling Loads in Economizer Range

- Climate does NOT matter for adjusted high limit inputs
- eQUEST provides climate adjustment
- Portland and Sacramento Adjustments all within 0.75°F



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Distinguish Commercial Ventilation Requirements and Modeling:

What to do if you aren't in California!

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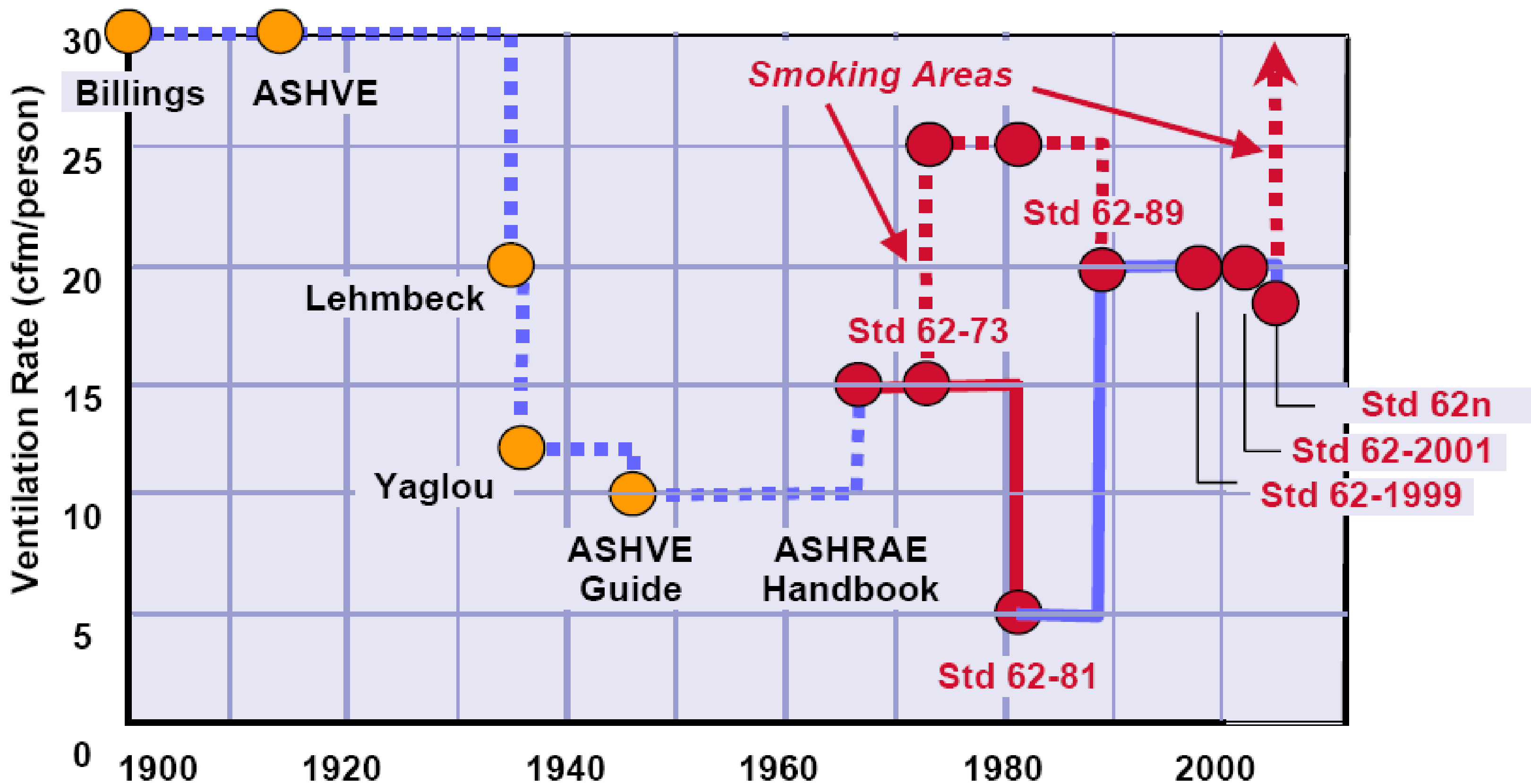
Ventilation Rate History

For office spaces...



Source: Fred Kohloss; ASHRAE Summer Meeting 2003

Health or Stinky Air?

- Primary purpose is “acceptable indoor air quality”
  - Primary unit in testing is “olf” (olfactory unit)
- Standard has a purpose to “minimize negative impacts on health”
  - Difficulty in proving objective healthy concentrations for contaminants of concern in commercial
  - Second hand smoke major restriction
  - Most commercial occupancies do not have significant health contaminant issues, with exception of dry cleaning, nail salons, light manufacturing and perhaps retail and warehousing

Standard 62.1 Procedures

- Ventilation Rate Procedure (Prescriptive Path)
  - For given occupancy, provide minimum ventilation when occupied
- IAQ Procedure
  - Provide a ventilation and air cleaning strategy that:
    - Addresses contaminants of concern
    - Is found acceptable based on survey or measurement
  - Now requires site specific investigation and monitoring
- Natural Ventilation Procedure
  - Limited to almost always require mixed mode

ASHRAE Standard 62.1
Ventilation Rate Procedure

- Determine an outside air ventilation rate and provide when occupied
- Ventilation now has 2 components
  - Area ventilation rate
  - People ventilation rate
- These 2 components are used to establish the total or "full" ventilation rate
  - Full ventilation rate = Area + People
  - There is NOT a requirement to provide the area rate to unoccupied areas!

People + Area Based Rate

- Occupancy from
  - Default tables (ASHRAE 62.1 or code); usually high
  - Observation or architectural program
  - Owner information
- Most codes set a floor for people basis to 50% of stated table default people densities
- Less important to get people accurate if using DCV

- Example; Conference Room, 1000 square feet:
  - Area: 1000 x 0.06 cfm/sf = **60 cfm**
  - People: 50 x 5 cfm/per = **250 cfm**
  - Total = **310 cfm** or 26% of 1200 unit or zone cfm

Ventilation Codes

- Most energy codes require DCV in high density spaces
- Most PNW mechanical codes either
  - Allow ASHRAE 62.1 as an alternate
  - Or are modeled after 62.1
- Occupancy type lists are generally shorter than ASHRAE; tied to fire occupancies
- California is different
  - Ventilation is maximum of area or people rate
    - Generally 0.15 cfm/sf area rate
    - Generally 15 cfm/person people
  - Requires pre-purge and continuous fan operation
  - DCV has general 1000 ppm setpoint
  - CO_2 sensor in space with readout required

California is Different (of course)

Standard: Requirement	ASHRAE 62.1 (since 2001n)	California Title 24 (2008/10 Compliance Manual)
People; Area rates	**Sum**	**Max (DOE 2.2 default)**
High density cfm per person	down to 5	stay at 15
General area: cfm/sf	0.06	0.15
Ventilate when	“actually occupied”	“Usually occupied”
Intermittent fan	Short periods (~30 min)	Only 5 minutes / hour
DCV C_R Setpoint	Match space (800-1800)	Always 1000 ppm
DCV Sensors	CO_2 return air OK; schedule, counters	CO_2 only, in space, readout, trended if DDC
Space purge	Not required (recommended if non human contaminants)	1 hour or 3 air changes before occupancy
Multiple zone	Ventilation efficiency	Min for space; Vent at fan

Space & Zone OA Specs ASHRAE 62.1 vs CA T24

Area Rate:

INF-FLOW/AREA

Unoccupied Cfm/sq ft; not wind corrected

Enter unoccupied; use INF-SCHEDULE to multiply for occupied

0.0263 + 0.12 = 0.1463

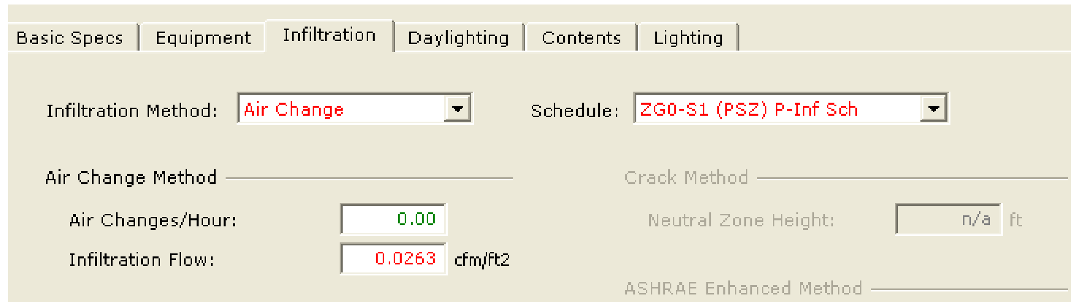
0.0263 x 5.56 = 0.12

People Rate:

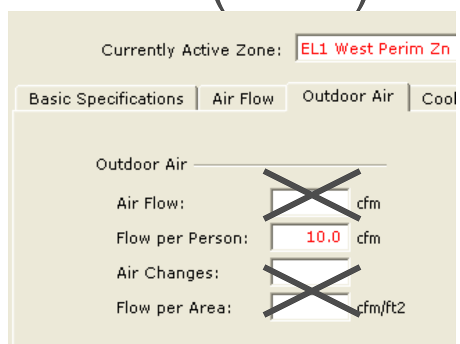
OA-FLOW/PER

(Not: OA-FLOW/AREA)

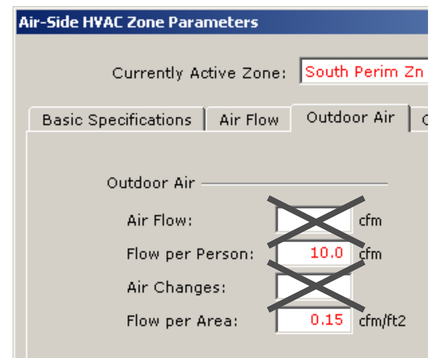
Space:



Zone (62.1):



CA T24:

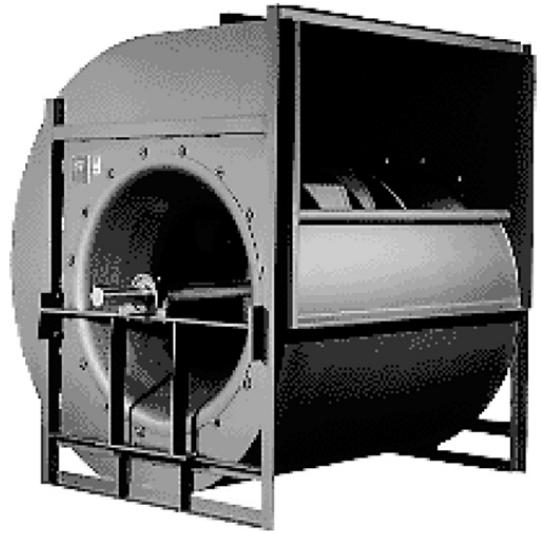


Infiltration works for area as OSA is OSA for SZ units

Prototype Minimum Ventilation

- The minimum ventilation requirements used for the prototype models follow ASHRAE 62.1-2010 Ventilation Rate Procedure.
- Ventilation = People Outdoor Air Rate + Area Outdoor Air Rate

Space Type	People Ventilation Requirement (CFM/Person)	Area Ventilation Requirement (CFM/SQFT)
Classroom	10	0.12
Gym	3.75	0.18
Lecture Hall	7.5	0.06
Movie Theater	10	0.06
Movie Theater Lobby	5	0.06
Retail	7.5	0.12



Understand DOE 2 Limits and Proper Modeling of Unoccupied Fan Operation

DOE 2.2 – Work-Arounds to Hidden Problems:
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eQUEST Defaults at Work

Night Cycle Fan Control Default:

- “Stay Off” is default
- eQUEST adds 2 hours/day to fan schedule

Options:

- “Cycle on Any” runs fan full hour on ANY load
- No option for
  - “on” during occupied and
  - intermittent” during unoccupied
- Intermittent with a 24 hour fan schedule is intermittent all the time

Solution: Use the Fan Schedule

- Hourly outputs analyzed for unoccupied load
- Monthly equivalent fan hours determined
- Schedules built up for prototypes

Not perfect,

but closer than

“cycle on any”

	Class room	Retail	Gym	Movie Theater	Lecture Hall
January	1	3	4	3	2
February	1	2	3	2	2
March	1	2	2	2	1
April	1	2	2	1	1
May	0	1	1	1	0
June	0	0	1	0	0
July	0	0	1	0	0
August	0	0	2	0	0
September	0	0	1	0	0
October	0	1	2	1	0
November	1	2	2	2	1
December	2	2	3	2	1

Longer Run Hours with VFD

- Similar process
- Lower speed results in longer hours

	Class room	Retail	Gym	Movie Theater	Lecture Hall
January	5	All	All	All	All
February	2	All	All	All	All
March	4	All	All	All	6
April	3	8	All	All	5
May	1	4	6	3	1
June	0	1	4	1	0
July	1	0	5	0	0
August	2	0	8	0	0
September	0	0	5	1	0
October	1	5	7	5	2
November	4	7	All	All	5
December	All	All	All	All	5

List Issues for Proper VFD Fan Modeling

DOE 2.2 – Work-Arounds to Hidden Problems:
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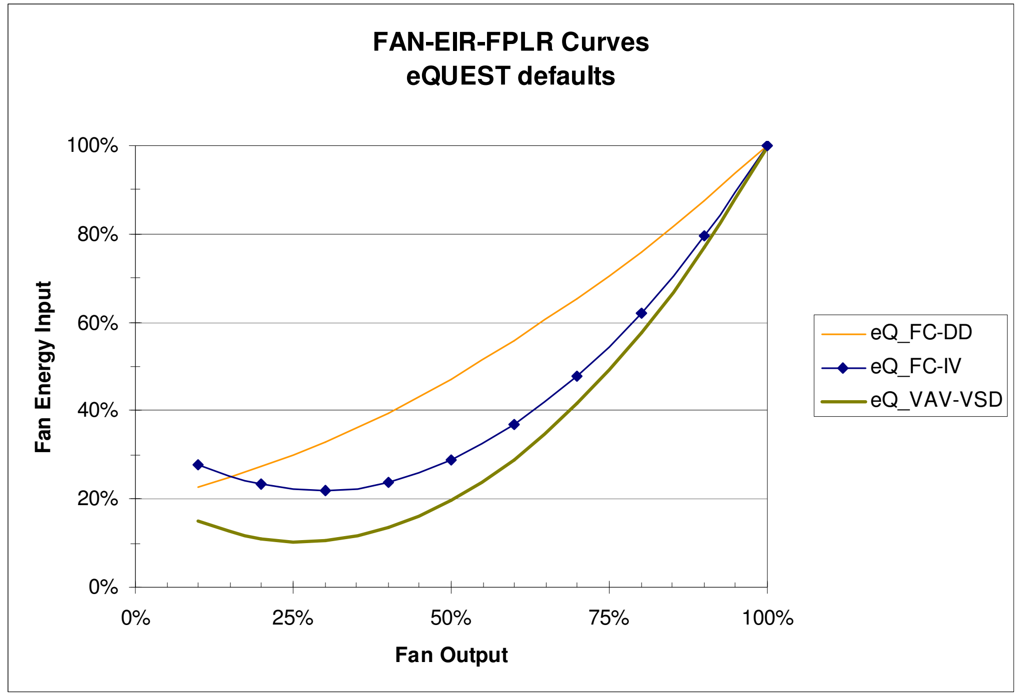
FAN EIR-fPLR

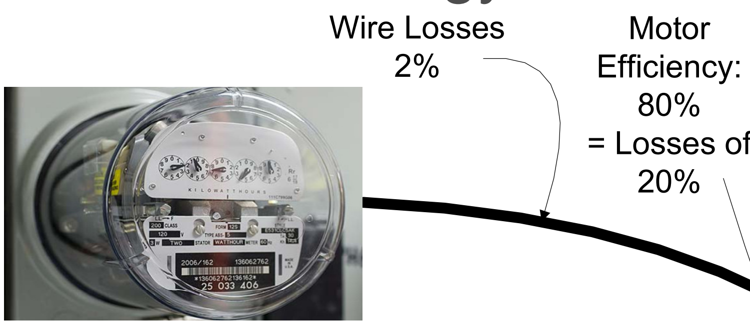
- EIR-fPLR:
  - Energy Input Ratio
  - as a function of
  - Part Load Ratio
- Dimensionless
  - Other formulas or inputs define 100% energy input
  - Systems finds CFM for the hour

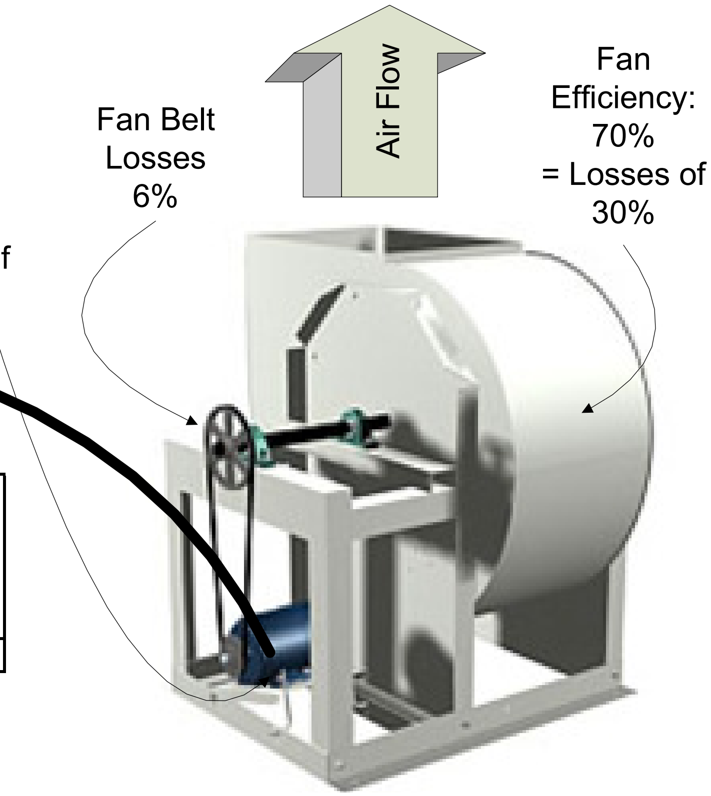


- EIR-fPLR curve quickly translates hourly fan output (cfm) to energy input
  - Does not require complexity of a ductwork model inside DOE2 or TRACE
  - Allows any separately calculated relationship to be input
- Default curves in the system for:
  - Forward Curve or Air Foil; Discharge Dampers or Inlet Vanes;
  - Any VSD; VaneAxial

Overall Fan System Efficiency
Multiple Conversions = Multiple Losses

Losses occur for each conversion of energy



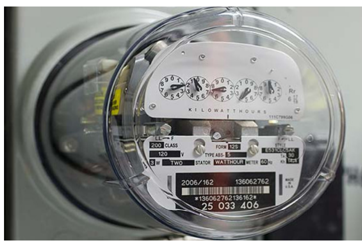


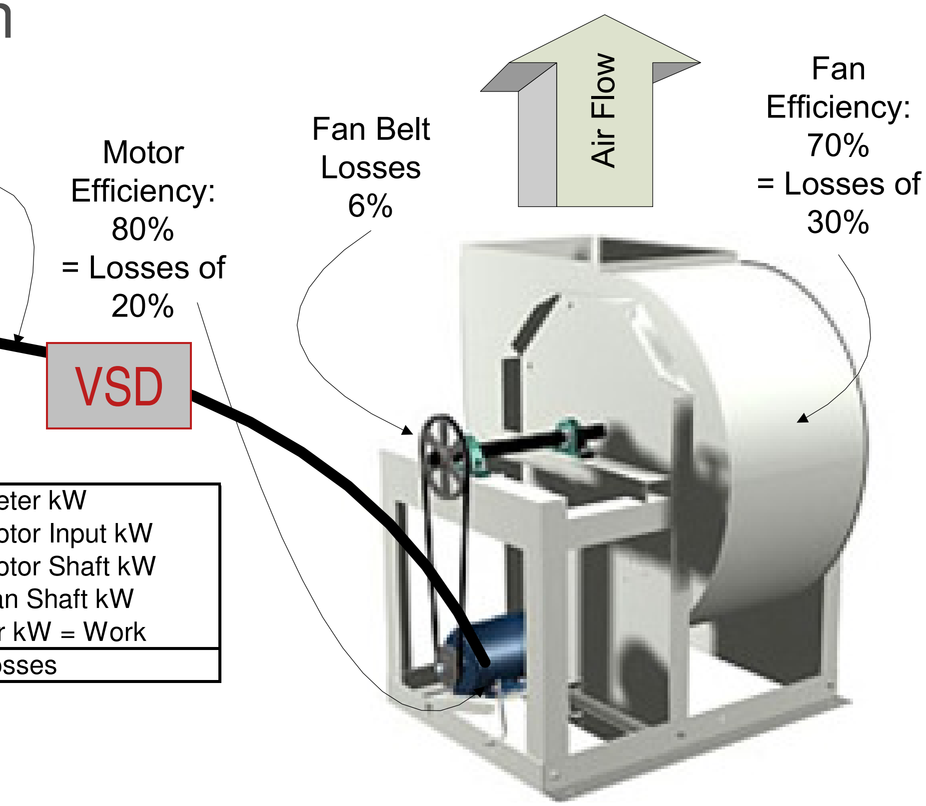
	Efficiency	Loss	kW	
Meter			**9.65**	Meter kW
Wire		2%	9.46	Motor Input kW
Motor	80%	20%	7.57	Motor Shaft kW
Belt		6%	7.14	Fan Shaft kW
Fan	70%	30%	**5.00**	Air kW = Work
System	52%	48%	4.65	Losses

Overall system efficiency = [5.0 kW work] / [9.65 kW in] = 52%

Overall Fan System Efficiency Multiple Conversions = Multiple Losses

- Losses occur for each conversion of energy





	Efficiency	Loss	kW	
Meter			**9.65**	Meter kW
Wire		2%	9.46	Motor Input kW
Motor	80%	20%	7.57	Motor Shaft kW
Belt		6%	7.14	Fan Shaft kW
Fan	70%	30%	**5.00**	Air kW = Work
System	52%	48%	4.65	Losses

What is impact of VSD?

- The old rule of thumb is added constant losses of 5-10% of peak input
- Newer or larger units closer to 2%-3% of peak

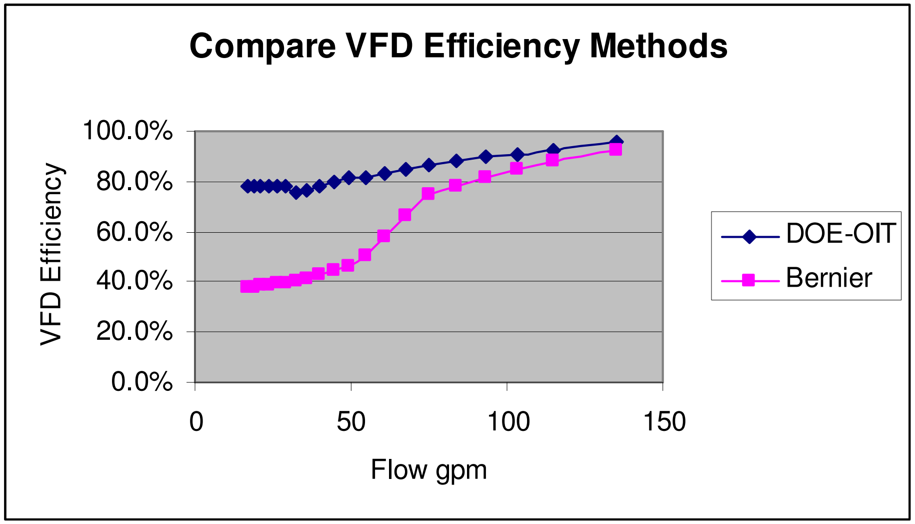
VFD Efficiency – A Research ?

VFD losses must be included in EIR-fPLR curve

- Two available references for VFD efficiencies were used in FanSysCalc
- They have different part load curves, as shown at right

- The DOE-OIT method is based on a table published in three references:
  - USDOE OIT. Ask the clearinghouse: Variable speed drive part-load efficiency. January 2002. Energy Matters. Office of Industrial Technologies, US Department of Energy. http://www.ornl.gov/sci/buildings/20070205_AuthorsManual_Jan2007.pdf
  - Chan, Tumin. August 2004. "Beyond The Affinity Laws." Engineered Systems Magazine. http://www.esmagazine.com/Archives?issue=65496
  - Rooks, JA & Wallace, AK. 2003. Energy efficiency of variable speed drive systems. Pulp and Paper Industry Technical Conference, 2003.

- The Bernier method uses an equation published in
  - Bernier, MA & Bourret, B. December 1999. Pumping energy and variable frequency drives. ASHRAE Journal.



Further Reading:

Development of a Correlation for System Efficiency of a Variable-Speed Pumping System (ASHRAE 2008)

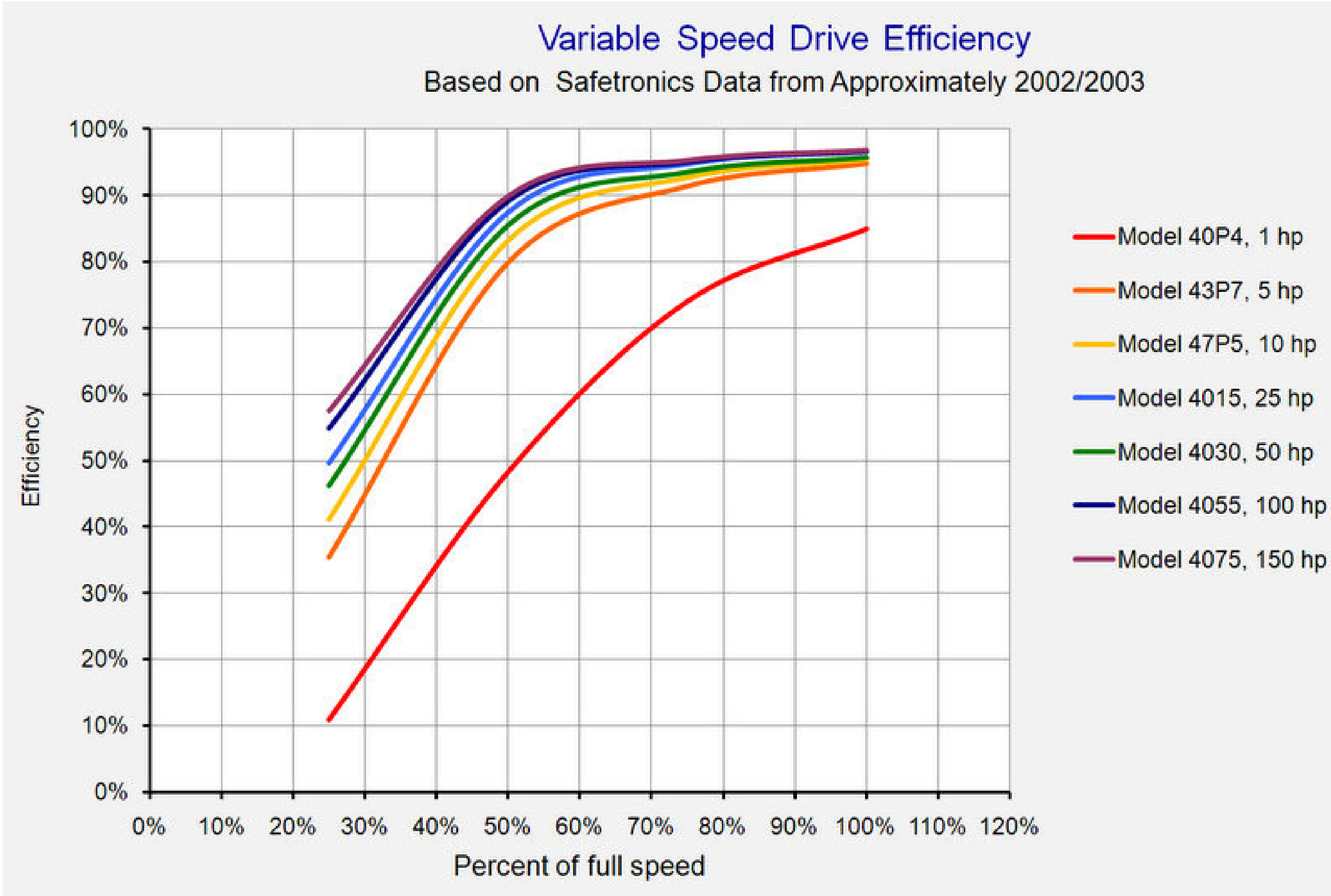
Efficiencies of an 11.2 kW Variable Speed Motor and Drive (ASHRAE 2001)

Non-Dimensional Pumping Power Curves for Water Loop Heat Pump Systems (ASHRAE 1999)

Utilizing VFD for Building HVAC System Performance (ASHRAE 2008)

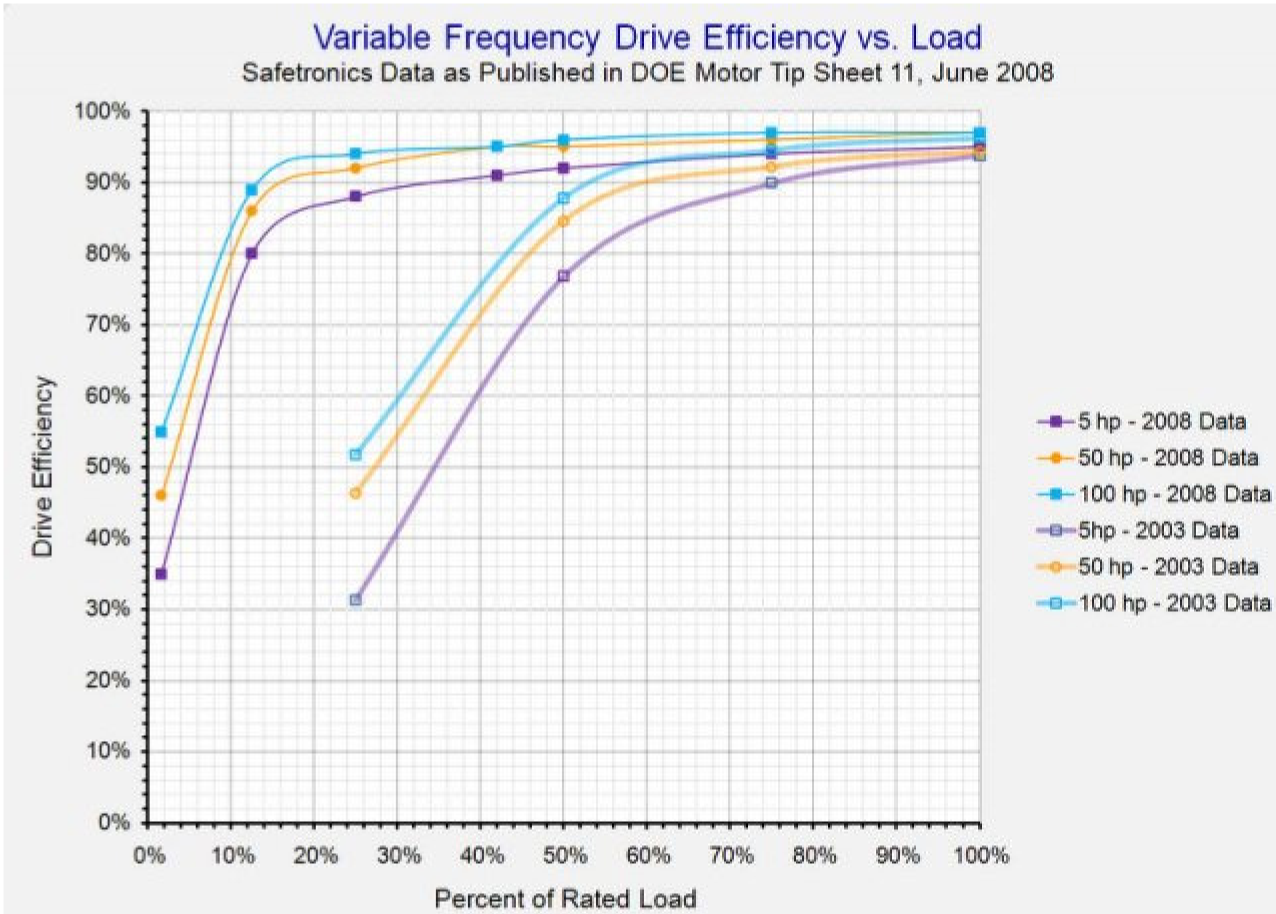
Variable Flow in Chilled Water Systems-Benefits and Controls (ASHRAE 2006)

VSD: Size Matters



Sellers, David. A Field Perspective on Engineering. Dec. 18, 2010.
http://av8rdas.wordpress.com/2010/12/18/variable-frequence-drive-system-efficiency/

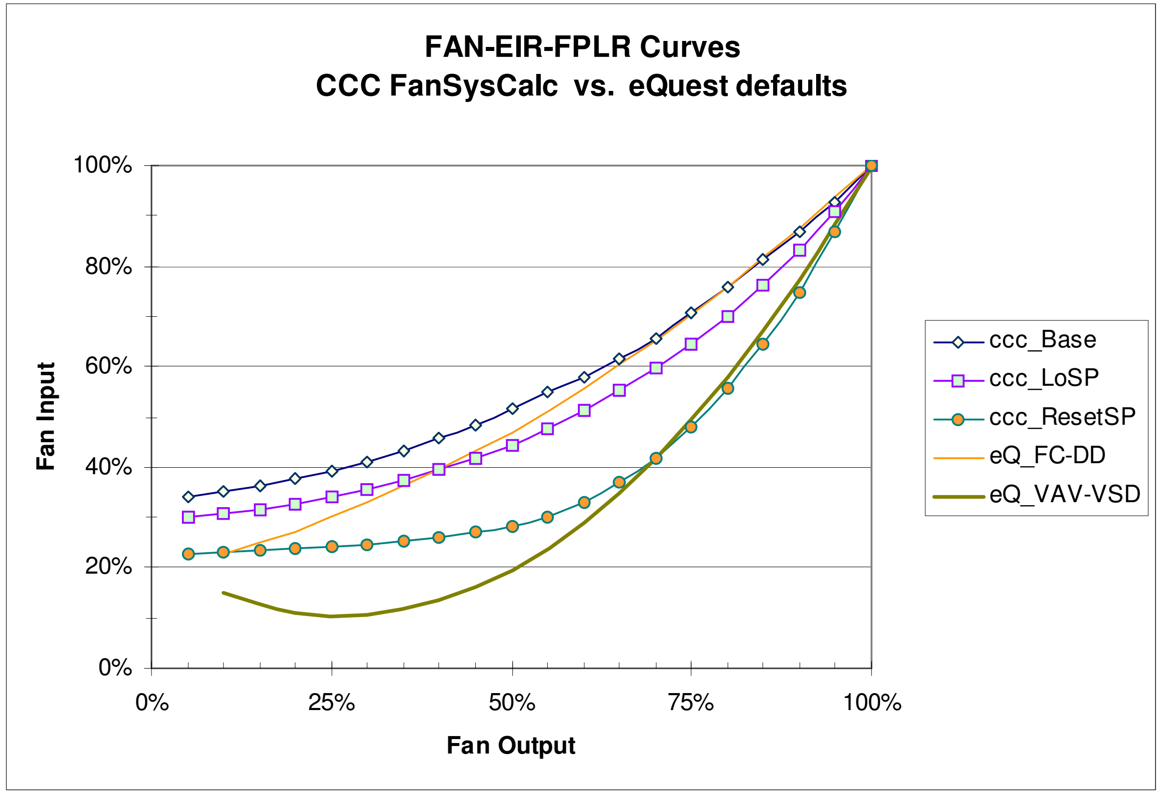
VSD: Vintage Matters



Sellers, David. A Field Perspective on Engineering. Dec. 18, 2010.
http://av8rdas.wordpress.com/2010/12/18/variable-frequence-drive-system-efficiency/

Simulating Static Pressure Reset in eQUEST/DOE2; TRACE

- Temperature reset
  - very easy in wizard
- Static Pressure reset
  - requires a new
  - FAN-EIR-fPLR curve
- The CCC FanSysCalc tool
  - is a good source to generate a realistic EIR-fPLR curve



Fan Cycling and VSD Control

- Each DCV prototype was modeled with three different fan control methods:
  - constant volume,
  - fan cycling, and
  - variable speed drive.
- The fan cycling and variable speed drive fan control methods respond to an increase or decrease in space temperature.
- Most packaged rooftop units are controlled such that the fan will operate during night time hours to meet heating and cooling loads.

Aspiration Duct Probe
TR9293-L

Know How to Model DCV

DOE 2.2 – Work-Arounds to Hidden Problems:
eQUEST - A half hour to learn;
three years to master.

Demand Controlled Ventilation (DCV)

- Idea here is to provide
  - area ventilation rate when occupied,
  - then proportion people ventilation rate to actual occupancy
- Measure occupancy by:
  - Carbon dioxide (CO_2) concentration
  - Counters or security system
  - Schedule approximation
  - Camera technology

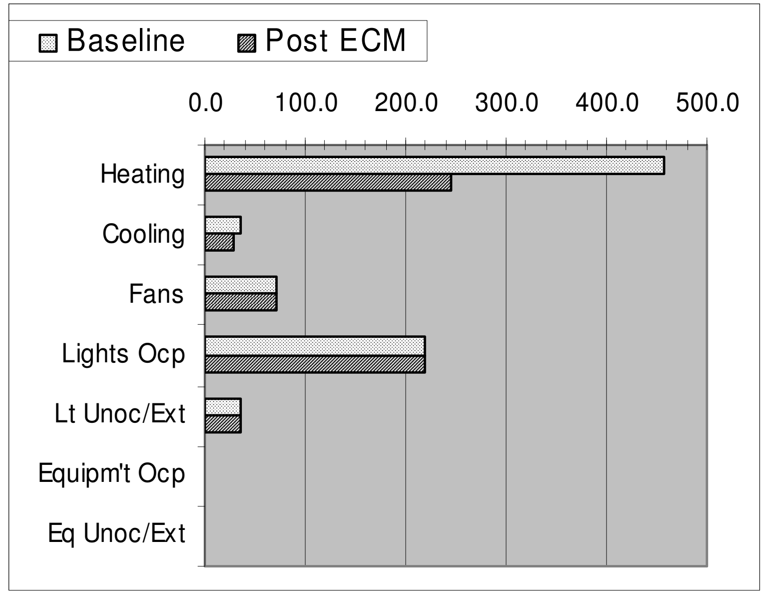
Demand Controlled Ventilation

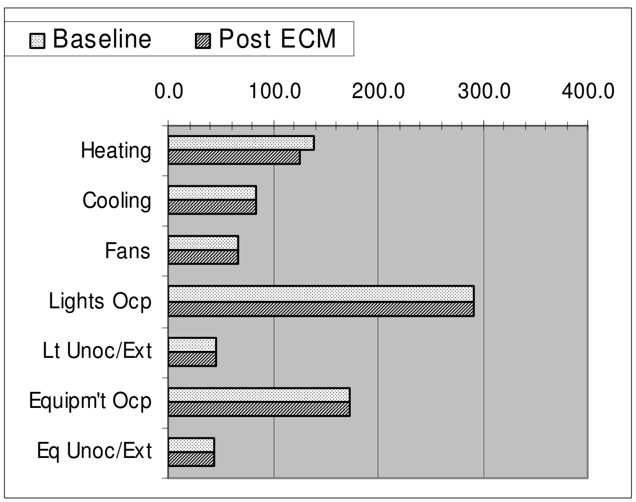
- Use CO_2 as a proxy for adequate ventilation with CO_2 sensors
  - Odor: ~700 above background
  - Target 1000 to 1200 ppm
  - For 2-10 V CO_2 sensor where 2 V is 800 ppm, we want damper to start at 4 V or 1100 ppm
  - Damper is full open at 10 V or 2000 ppm
- For smaller areas:
  - Use occupancy sensors
  - Use scheduled ventilation

DCV Savings

Gym is a great application

- Go from 38% average OSA to 15%
- Save $2200 per year, mostly heat





Office saves less, but may pay

- Only need 10-15% OSA; might reduce by 5%
- Save $153 per year
- Save more if base case is over ventilated (typically is 20%)

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CO_2 Sensors

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- Why are they used?
  - Energy Savings:
    - Only when not in OSA economizer mode
  - Eliminate over-ventilation when occupancy is low
  - Coordinated with OSA economizer maximizes benefit of ventilation control and savings
  - Establish transient space occupancy
- Alternate methods
  - Schedules, camera visualization, counters
  - VOC sensors (if co-related to CO_2)

CO_2 as Proxy for Air Quality

- Carbon Dioxide is JUST a proxy
  - Does not account for other chemicals
  - Does reflect space occupancy
  - Is not a “contaminant of concern”
- What do CO_2 levels mean?
  - Background outdoors 350 to 450 ppm
  - Equivalent to 15-20 cfm/person: 1000 ppm
  - OSHA limit for exposure > 8 hrs: 5000 ppm
  - Submarines: 7000 normal, 30,000 dive!

CO_2 Target Is NOT 700 ppm!

- DCV CO_2 concentration setpoint
  - Earlier standard called for 700 ppm CO_2 over ambient as target level
  - Ambient at 400 means 1100 ppm setpoint
- Actually is based on specific occupancy
- Explained well in the 62.1-2007 users manual
  - Steady state CO_2 C_R

$$C_R = C_{OA} + \frac{8400 E_z m}{R_p + R_a A_z / P_z}$$

- Concentration lags occupancy
  - DCV allowed to control for steady state
  - Pre-purge not required; may be good

Find C_R for your space

$$C_R = C_{OA} + \frac{8400 E_z m}{R_p + \frac{R_a A_z}{P_z}}$$

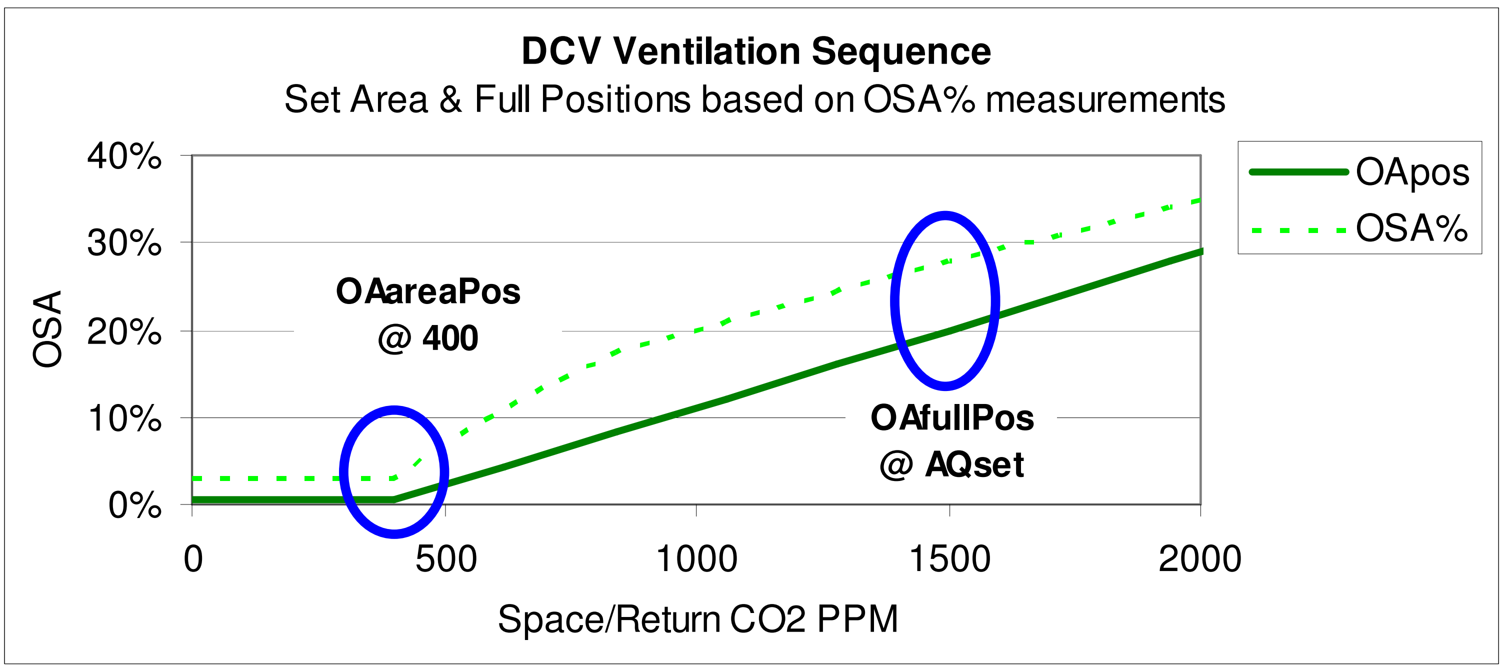
- C_R is target CO_2
  - Think OA% “area” & “full”, not minimum
  - Met varies due to activity
  - Large ratio indicates more DCV savings potential
  - Where possible, use actual people & CFM, not defaults!

ASHRAE 62.1-2010 Area Type	Pz	sf/p	met	Rp cfm/p	Rs cfm/sf	**CR ppm**	OA% area	OA% full	Ratio
Art Classroom	20	50	1.2	10	0.18	**824**	19%	40%	2.1
Office - default	5	200	1.2	5	0.06	**874**	6%	9%	1.4
Office - open	7	143	1.2	5	0.06	**994**	6%	10%	1.6
Class (age 9+)	35	29	1.2	10	0.12	**1001**	13%	49%	3.9
Retail Sales	15	67	1.5	7.5	0.12	**1050**	13%	24%	1.9
Grocery	8	125	1.7	7.5	0.06	**1162**	6%	13%	2.0
Call Center	12	83	1.2	5	0.06	**1206**	6%	13%	2.0
Lecture Class	65	15	1.1	7.5	0.06	**1278**	6%	57%	9.1
Movie Theater (actual)	77	13	1.0	5	0.06	**1563**	6%	46%	7.4
Conference	50	20	1.1	5	0.06	**1592**	6%	32%	5.2
Restaurant	70	14	1.4	5	0.18	**1643**	19%	55%	2.9
Assembly	150	7	1.0	5	0.06	**1644**	6%	84%	13.5
Rock Concert (dance)	100	10	2.0	5	0.06	**2800**	6%	58%	9.3

*For all types: area is 1000 square feet, COa = 400 ppm, Ez is 80%, unit cfm 1.2 cfm/sf

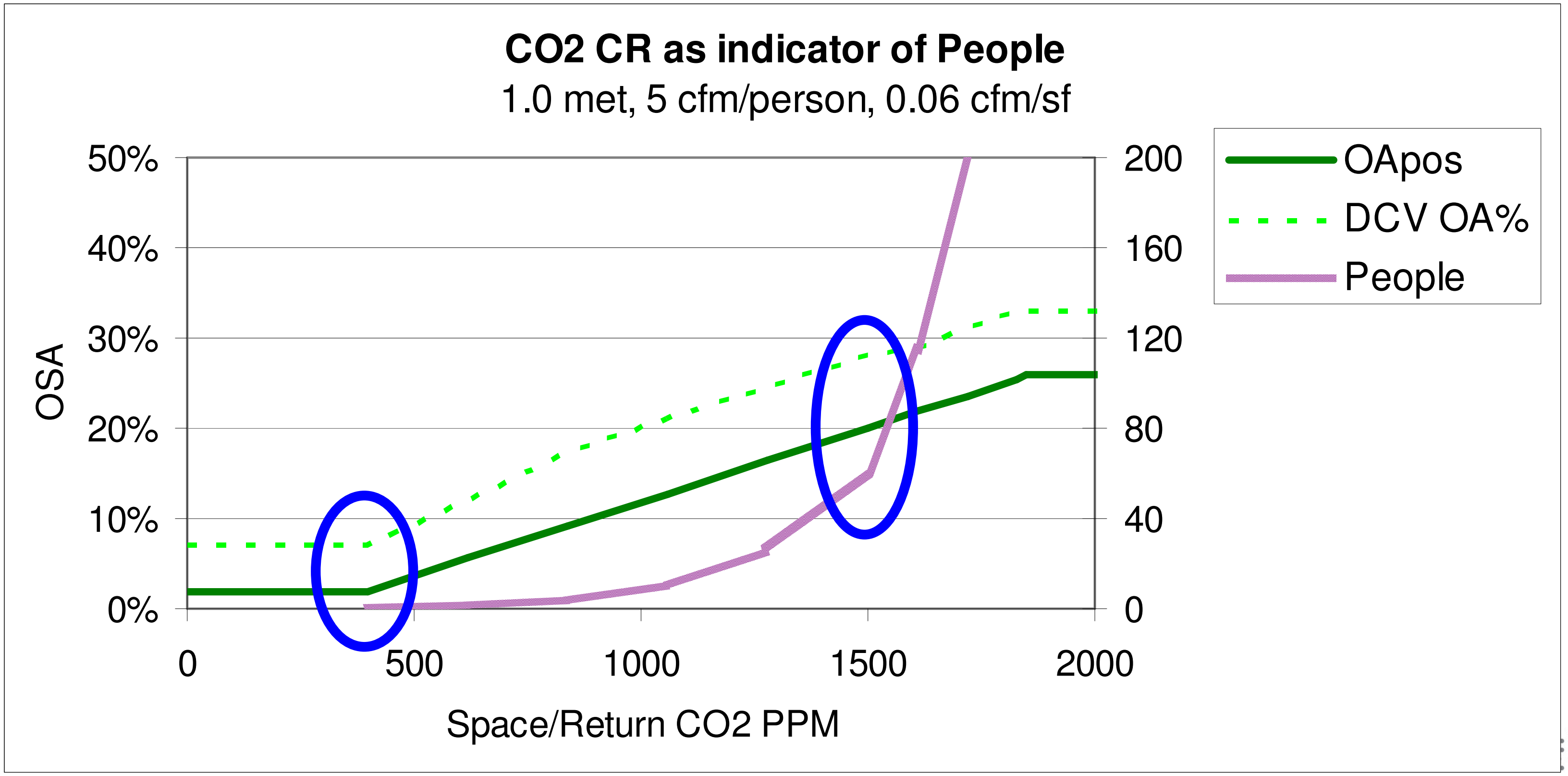
DCV: Step vs Analog Control

- Step control:
  - "full %" when approaching C_R or concentration target
  - Return to "area %" @ point between C_R and 400 ppm
- Analog control recommended by 62.1 User Manual
  - Area rate at 400-450 ppm
  - Area + people (full) rate at C_R target ppm
  - Note difference in damper position vs OSA%
- Analog is how DOE 2 models DCV according to people schedule



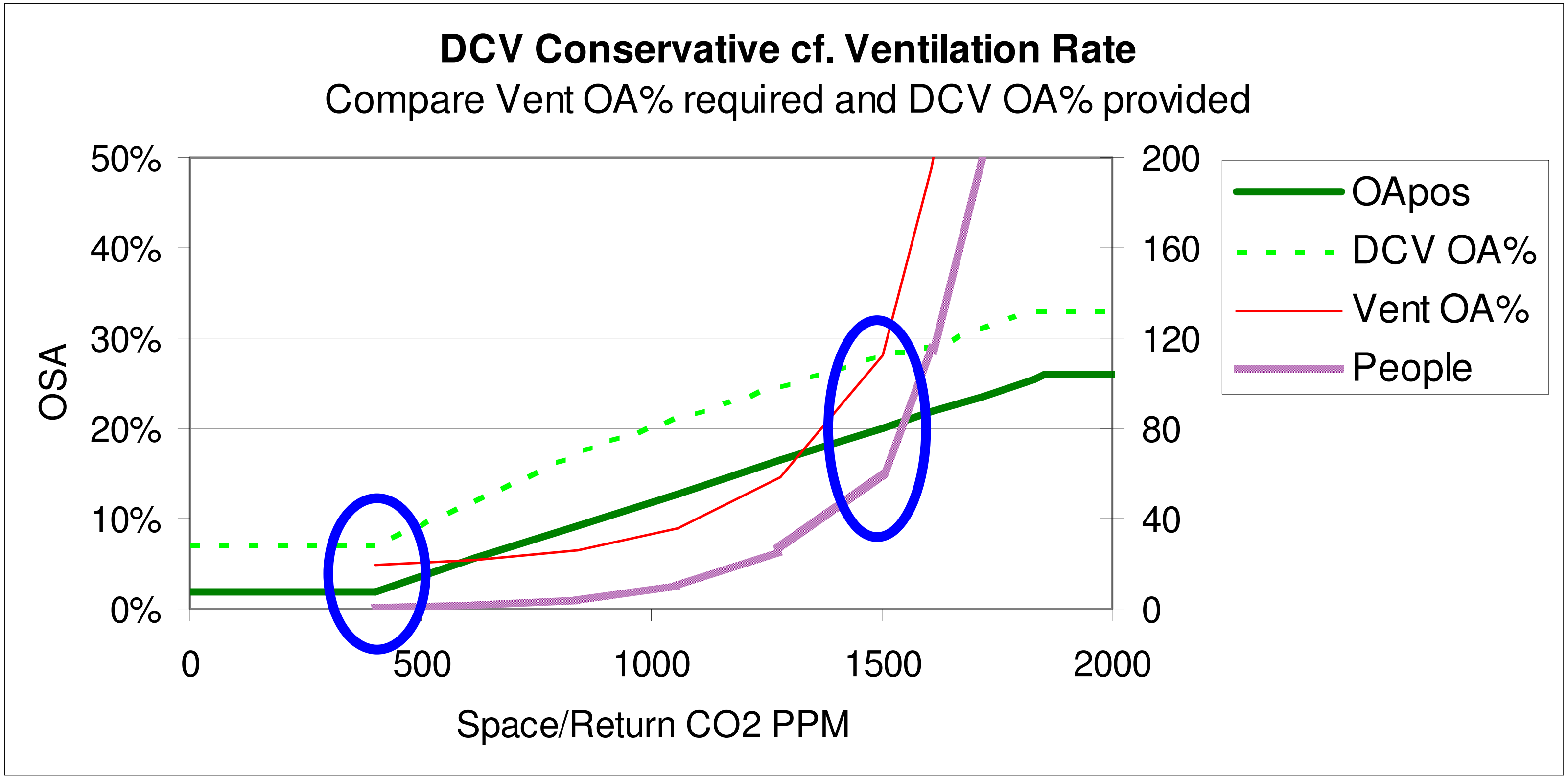
C_R as Indicator of People Concentration

- C_R good up to typical high density
  - Curve goes exponential at maximum densities
  - 200 people per 1000 sf is 5 sf a person – rock concert SRO
  - This is movie theater or assembly occupancy



Linear Control is Quite Conservative

- Compare DCV to prescriptive rate
  - High at lower proportional occupancy
  - Helps offset lag due to space volume



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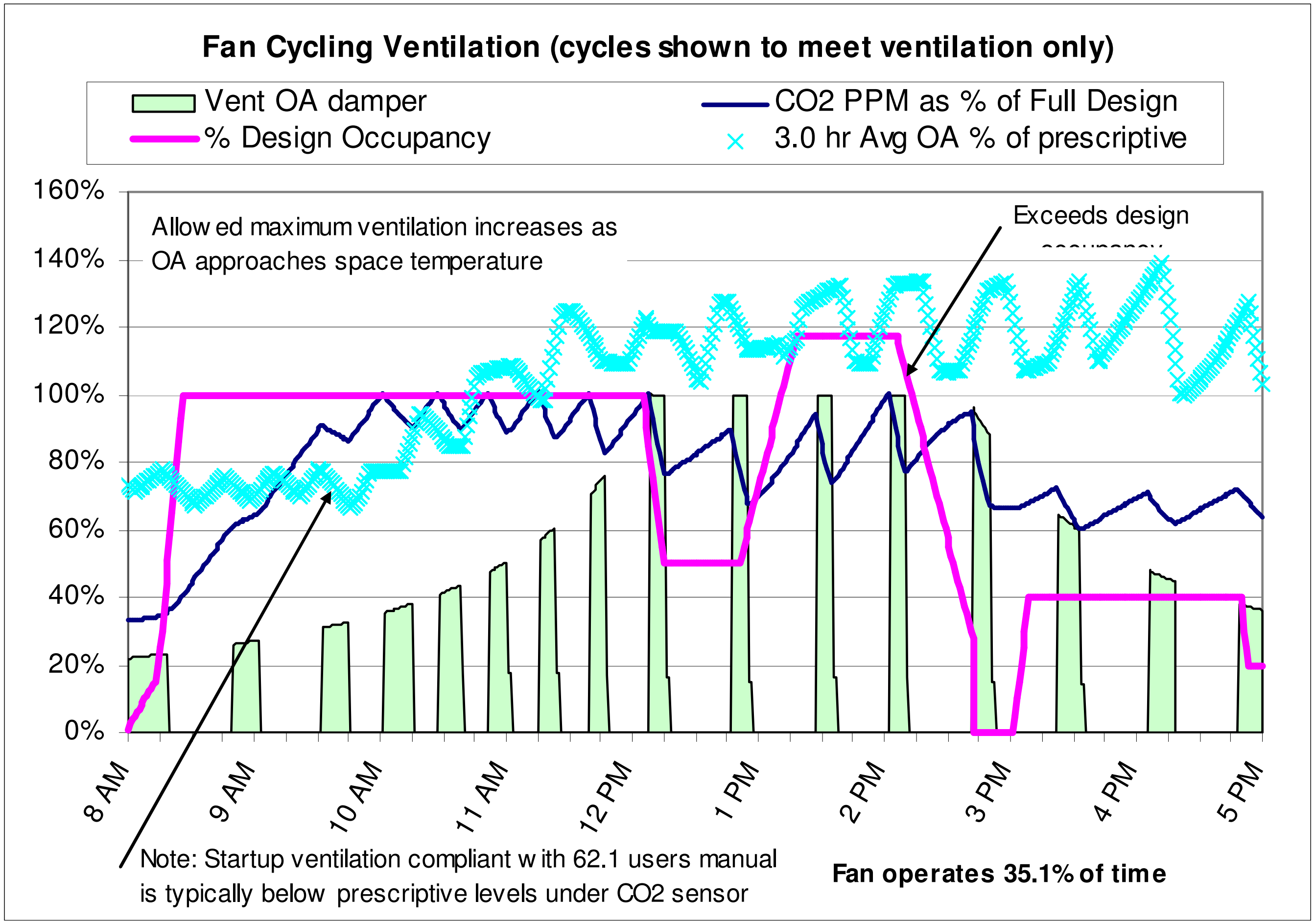
DCV Integrated Fan Control (DCV-IFC)

Yes, ASHRAE Standard 62.1 allows fan cycling

- General thinking: commercial fans must be ON during the occupied period
- Section 6.2.6.2 of ASHRAE Standard 62.1-2010 allows short-term interruption of ventilation if ventilation levels are maintained on average



Approach:

-Record CO_2 at fan off

-Provide % air relative to comfort

-Operate time needed to match ventilation rate

-Duty cycle if off for 30 minutes with no heat or cool

Aspiration Duct Probe

RTU DCV Ventilation Issues

Improve ventilation

CO_2 Transmitter Accuracy

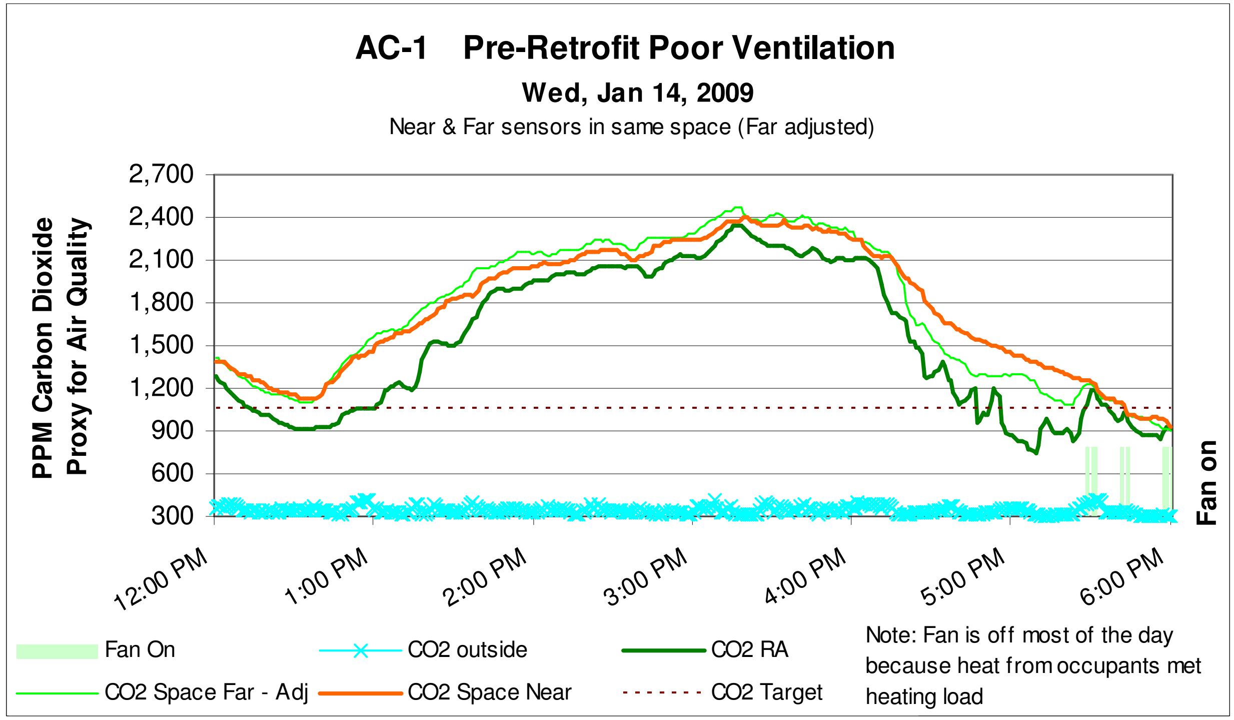
Fan cycling

Control setup and adjustment

Alterations & Code

Current RTU Conditions

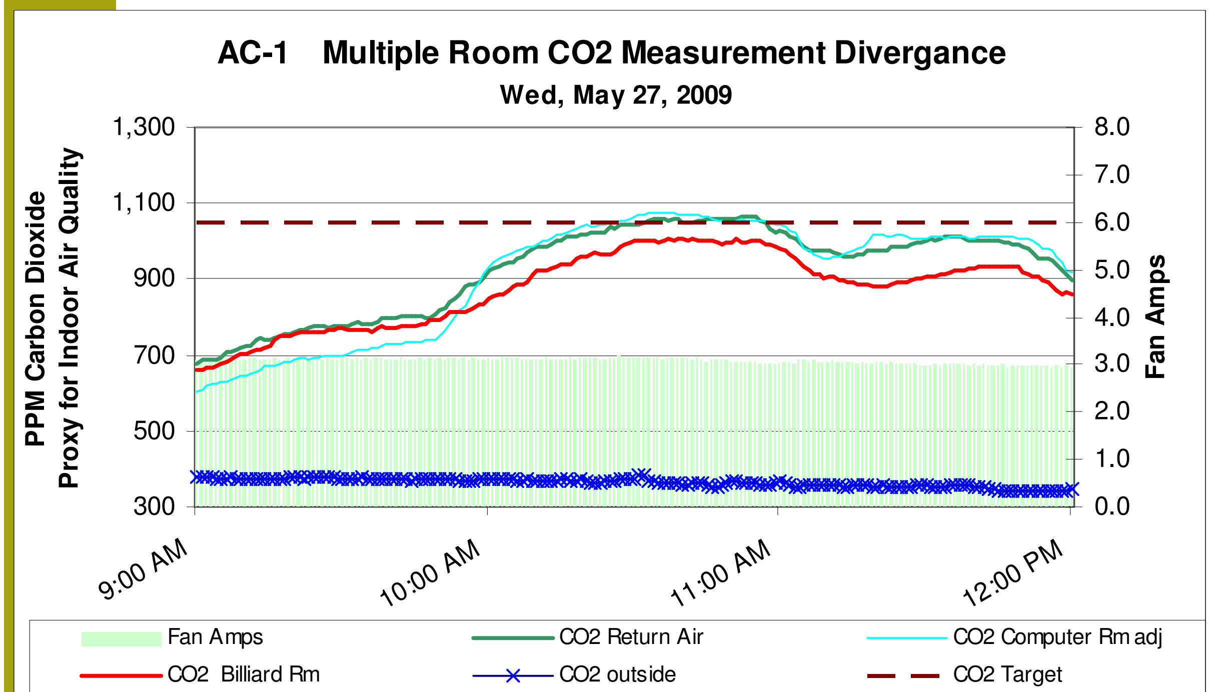
- TAB rare for RTUs
- OA minimum often set much too high
- Specific study of senior center



- Like here, fan is often in auto position (~40% RTUs)
- During January, little daytime fan operation
  - Heating load offset by internal gains
- CO^2 concentration almost triple desired level

RTU DCV Brings Significant Ventilation Improvements

- Fan slows when not heating or cooling, but remains on
- CO2 sensor modulates damper to maintain proper ventilation rate



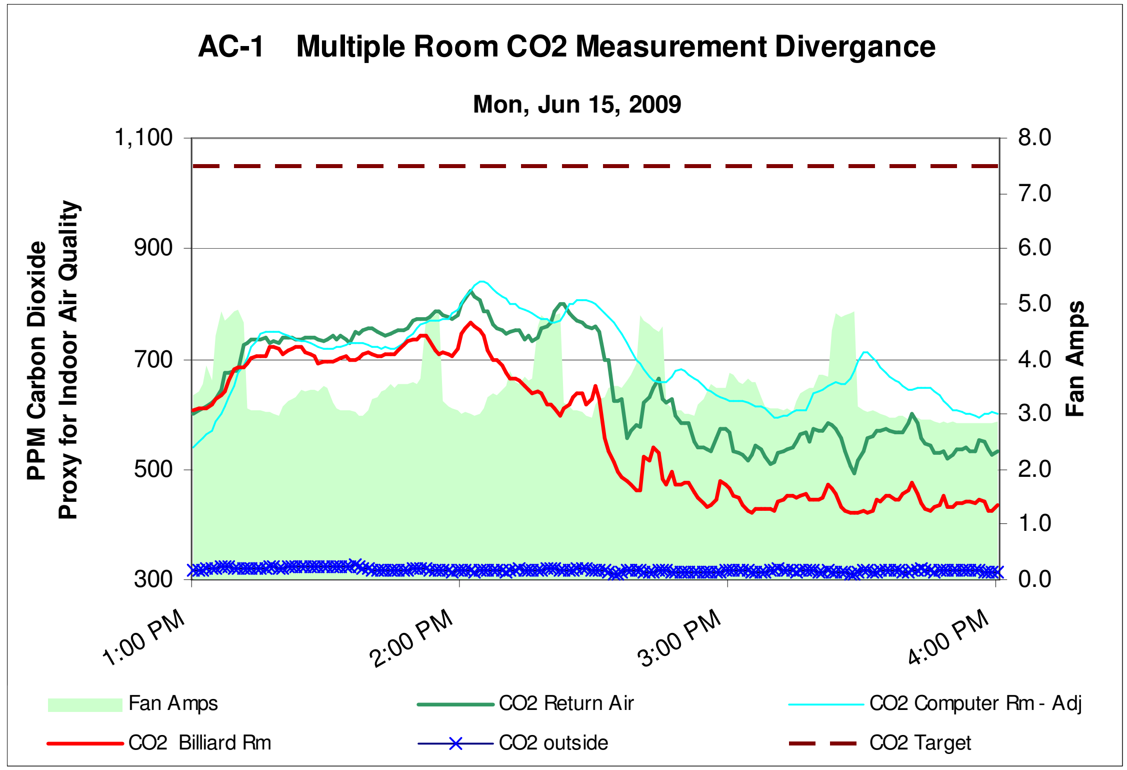
- After retrofit, highest CO2 concentration within target limits.
- Single return air sensor worked for multiple rooms:
  - Computer lab
  - Billiard room

Single CO² Sensor Does It

Easiest placement of the sensor is in the return air

Concerns:

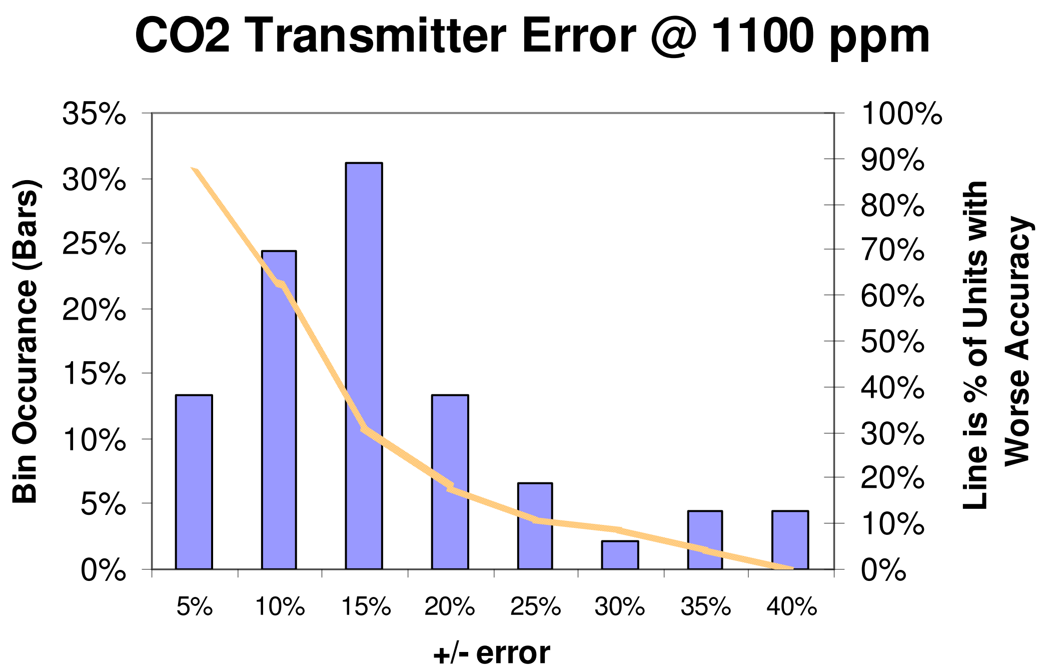
- Studies suggest putting sensor in the breathing zone (California T24 code required)
- When multiple rooms are served by one unit, an imbalance in ventilation quality may occur with uneven occupancy
- With VSD, less air throw at the diffuser may reduce ventilation effectiveness.



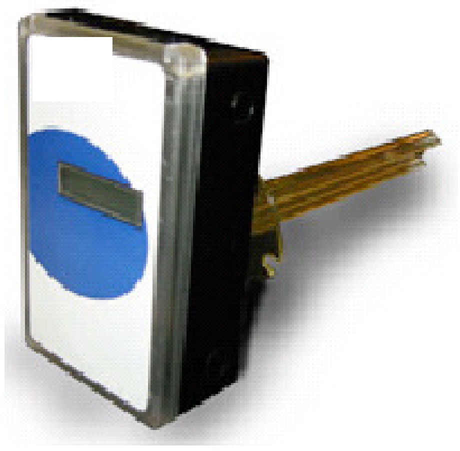
Testing results:

- For smaller RTUs, even with one room heavily occupied, **return air sensing** can meet requirements with a slight ventilation setpoint adjustment
- Testing multiple places in one room showed little variation with lower airflow

CO_2 Transmitter Accuracy



Source: National Building Controls Information Program. PTR. Iowa EC & CEC. June 2009.



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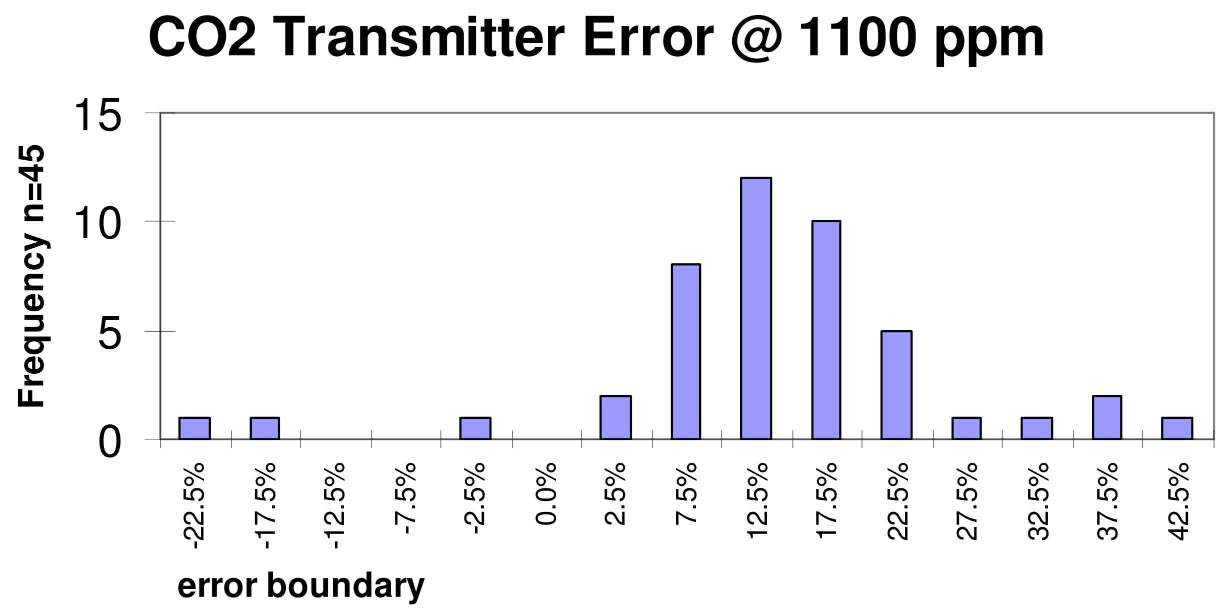
Source:
http://www.energy.iastate.edu/Efficiency/Commercial/download_nbcip/PTR_CO2.pdf

NBCIP/ Iowa Energy Center Testing

No manufacturers had all three samples within specs!

- Count on being within 200 ppm, usually high
- Many sensors saturate above 1700 ppm
- Self calibrating sensors improve persistence

Closer Look at Accuracy Impact



Source: National Building Controls Information Program. PTR. Iowa EC & CEC. June 2009.

Eliminating 3 of 13 manufacturers keeps error within +/- 21%

- Sensors are generally high; (mean = + 12%)
- Relative accuracy will maintain with self-calibration
- Particular manufacturers are at extremes

Questions ?

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