Trane Engineers Newsletter Live

Chilled-Water Terminal Systems

Presenters: John Murphy, Mick Schwedler, Eve London, Jeanne Harshaw (host)

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Agenda

Trane *Engineers Newsletter Live* Series

Chilled-Water Terminal Systems

Abstract

In this program, Trane applications engineers will discuss system design and control strategies for various types of chilled-water terminal systems, including fan-coils, chilled beams, and radiant cooling. Topics include: types of terminal equipment, variable-speed terminal fan operation, dedicated OA system design, chilled-water system design, and complying with ASHRAE 90.1 requirements.

Presenters: Trane applications engineers John Murphy, Mick Schwedler, Eve London

After viewing attendees will be able to:

1. Summarize design and control strategies that can save energy in various types of chilled-water terminal systems, including fan-coils, chilled beams, and radiant cooling
2. Understand the latest fan motor technology being used in chilled-water terminal units
3. Apply design and control strategies in a dedicated OA system as part of a chilled-water terminal system
4. Learn how to design and control the chilled-water plant for various types of terminal units
5. Understand how the requirements of ASHRAE Standard 90.1 apply to chilled-water terminal systems

Agenda

- Types of chilled-water terminal units
  - Fan-coils / blower coils
  - Chilled beams
  - Radiant
- Dedicated OA system design
- Chilled-water system configurations and control
- Summary

Presenter biographies

Chilled-Water Terminal Systems

John Murphy | applications engineer | Trane

John has been with Trane since 1993. His primary responsibility as an applications engineer is to aid design engineers and Trane sales personnel in the proper design and application of HVAC systems. As a LEED Accredited Professional, he has helped our customers and local offices on a wide range of LEED projects. His main areas of expertise include energy efficiency, dehumidification, dedicated outdoor-air systems, air-to-air energy recovery, psychrometry, and ventilation.

John is the author of numerous Trane application manuals and Engineers Newsletters, and is a frequent presenter on Trane's *Engineers Newsletter Live* series. He has authored several articles for the ASHRAE Journal, and was twice awarded "Article of the Year" award. As an ASHRAE member he has served on the "Moisture Management in Buildings" and "Mechanical Dehumidifiers" technical committees. He was a contributing author of the *Advanced Energy Design Guide for K-12 Schools* and the *Advanced Energy Design Guide for Small Hospitals and Health Care Facilities*, a technical reviewer for the *ASHRAE Guide for Buildings in Hot and Humid Climates*, and a presenter on the 2012 ASHRAE "Dedicated Outdoor Air Systems" webcast.

Mick Schwedler | applications engineer | Trane

Mick has been involved in the development, training, and support of mechanical systems for Trane since 1982. With expertise in system optimization and control (in which he holds patents), and in chilled-water system design, Mick's primary responsibility is to help designers properly apply Trane products and systems. Mick provides one-on-one support, writes technical publications, and presents seminars.

A recipient of ASHRAE's Distinguished Service and Standards Achievement Awards, Mick Chairs ASHRAE's Advanced Energy Design Guide (AEDG) Steering Committee and is past Chair of SSPC 90.1. He also contributed to the ASHRAE GreenGuide and is a member of the USGBC Pilot Credits Working Group. Mick earned his mechanical engineering degree from Northwestern University and holds a master's degree from the University of Wisconsin Solar Energy Laboratory.

Eve London | product manager | Trane

Eve London joined Trane in 1998 and is the Product Manager for Unit Heater and Terminal Products. She is responsible for all activities leading to the utilization of terminal fan coil, blower coil, unit ventilator and unit heater products.

London received a Bachelor of Industrial Engineering from Georgia Institute of Technology and a Master of Science in Engineering from Mercer University. She is a member of the USGBC and the AHRI Room Fan Coil Compliance Committee.

Chilled-Water Terminal Systems

Trane Engineers Newsletter Live Series

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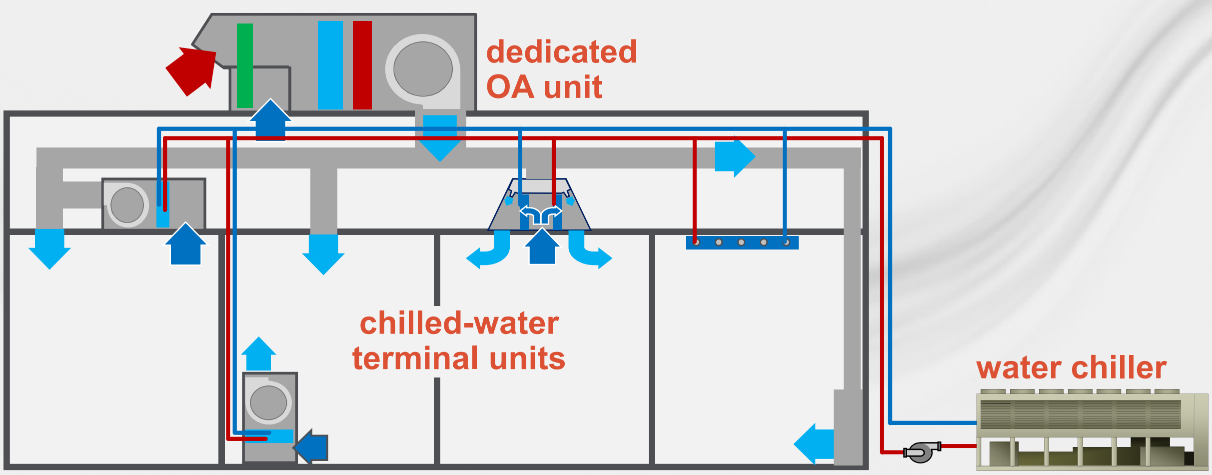
Learning Objectives

- Summarize design and control strategies that can save energy in various types of chilled-water terminal systems, including fan-coils, chilled beams, and radiant cooling
- Understand the latest fan motor technology being used in chilled-water terminal units
- Apply design and control strategies in a dedicated OA system as part of a chilled-water terminal system
- Learn how to design and control the chilled-water plant for various types of terminal units
- Understand how the requirements of ASHRAE Standard 90.1 apply to chilled-water terminal systems

Chilled-Water Terminal System



Agenda

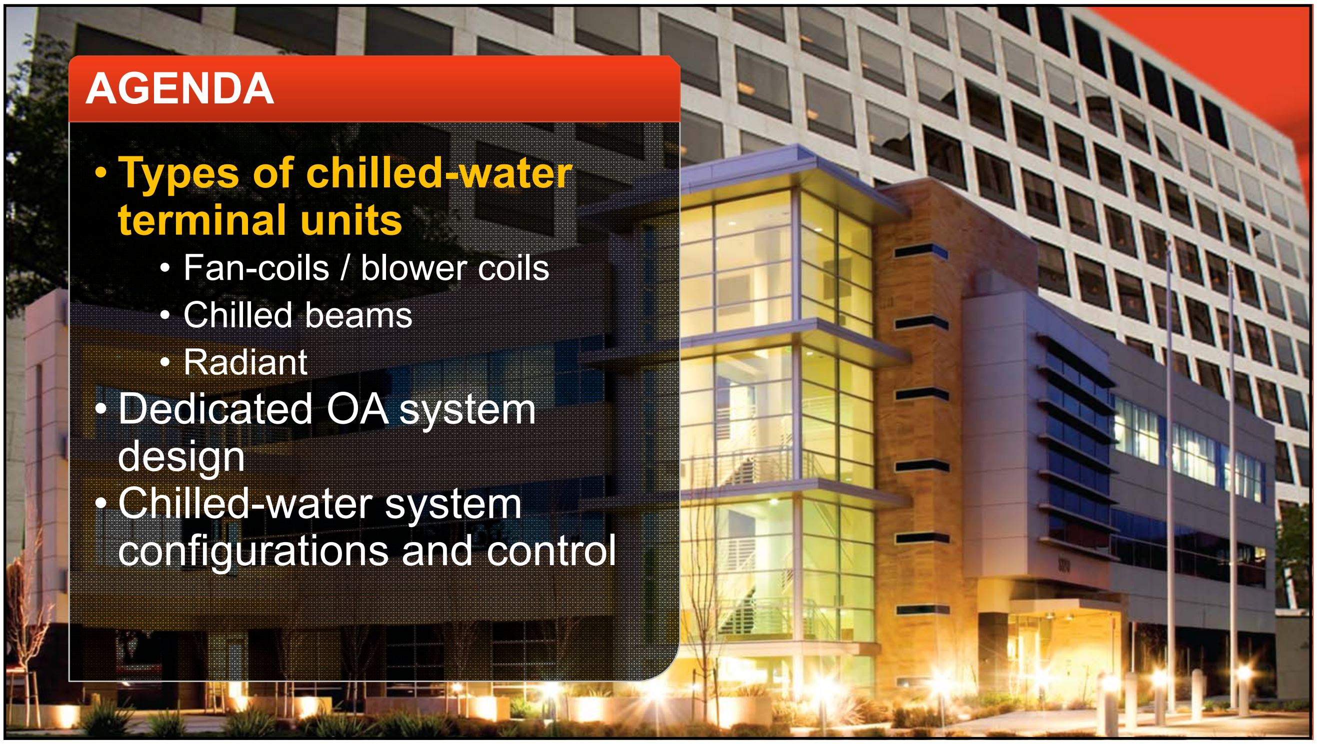
- Types of chilled-water terminal units
  - Fan-coils / blower coils
  - Chilled beams
  - Radiant
- Dedicated OA system design
- Chilled-water system configurations and control

Today's Presenters

Eve London
Product Manager

John Murphy
Applications Engineer

Mick Schwedler
Applications Engineer


AGENDA
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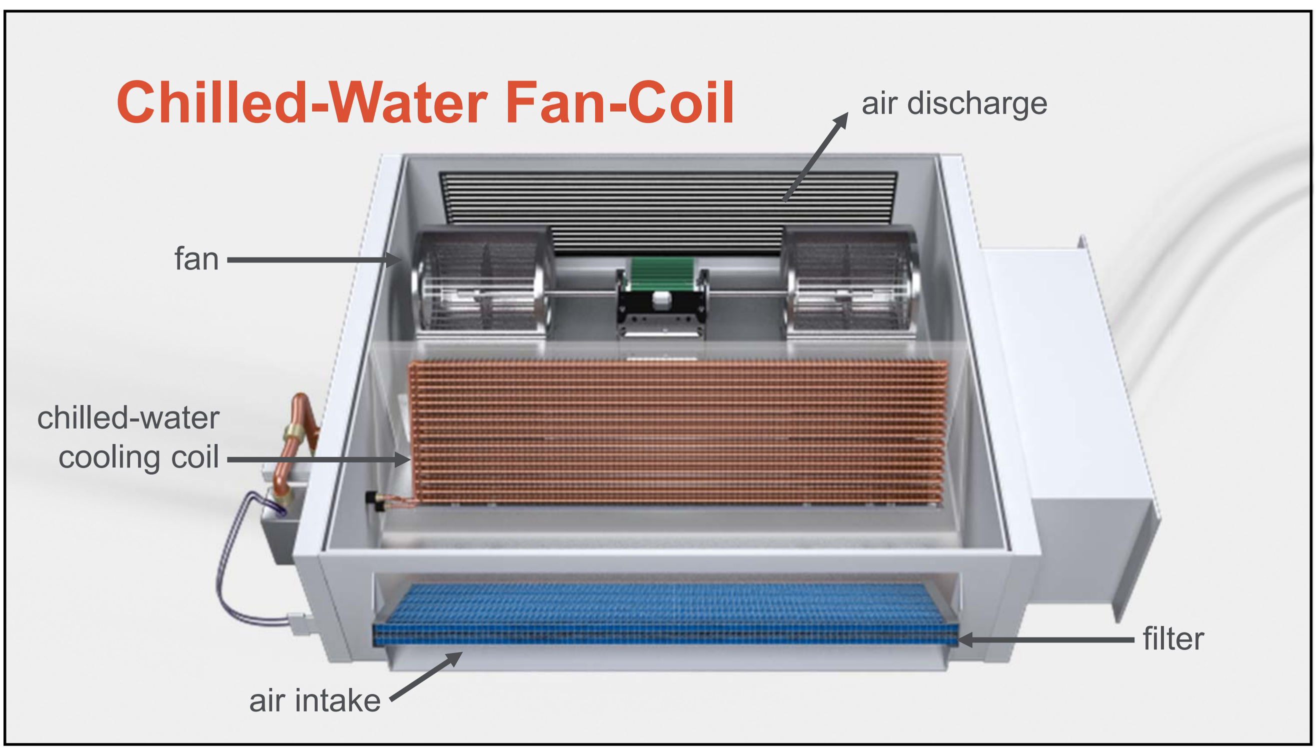

Chilled-Water Fan-Coil
air discharge
fan
chilled-water cooling coil
filter
air intake

Examples of Various Fan-Coil Styles

vertical cabinet

horizontal concealed

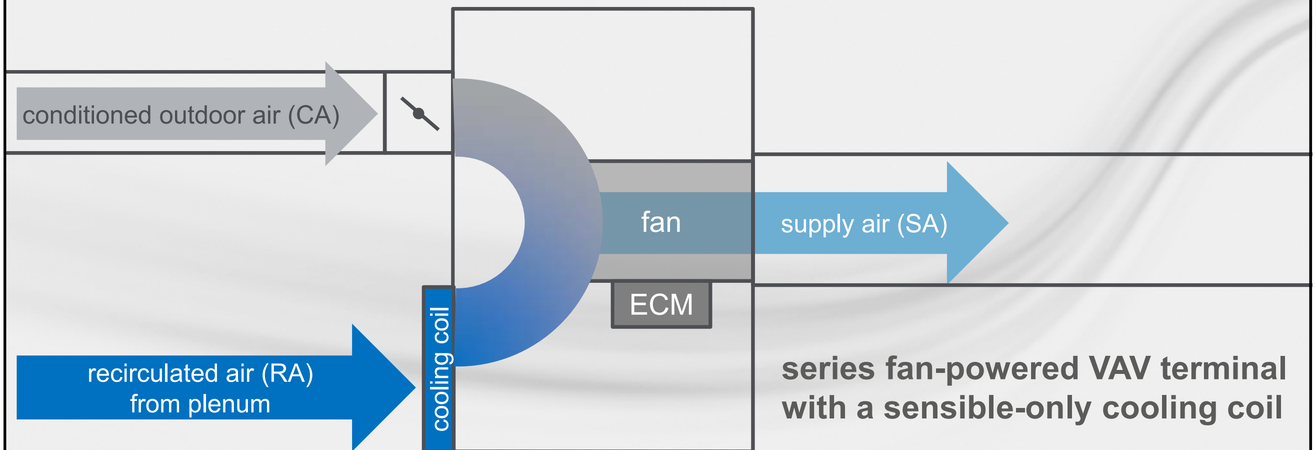
vertical stack

Similar Chilled-Water Terminal Units

blower coil air handlers

classroom unit ventilator

Similar Chilled-Water Terminal Units



series fan-powered VAV terminal with a sensible-only cooling coil

Standard Motor Technology

Permanent Split Capacitor (AC) Motor

- The function of an electric motor is to convert electrical energy into mechanical energy
- Fractional-horsepower, single-phase AC motors are relatively inefficient
- AC motors are designed to run most efficiently at the rated voltage and speed
  - Multiple-speed capability has traditionally been achieved with multiple winding taps

Advanced Motor Technology

Electronically-Commutated Motor (ECM)

- Brushless technology extends motor service life and reduces maintenance
  - Brushes no longer need to be cleaned, and dust from brushes is eliminated
  - Eliminates speed restrictions inherent with “brushed” DC motors
- Commutator doesn’t carry current to rotor
  - Eliminates brushes and their wear-related drawbacks

Reduces maintenance and increases service life

Advanced Motor Technology

ECM Performance

Constant-volume application

- Motors can be used with traditional thermostats
- Soft ramp between speeds
  - Less noticeable by occupants
- Programmability
  - Motor speeds (rpm) can be adjusted to minimize acoustical levels

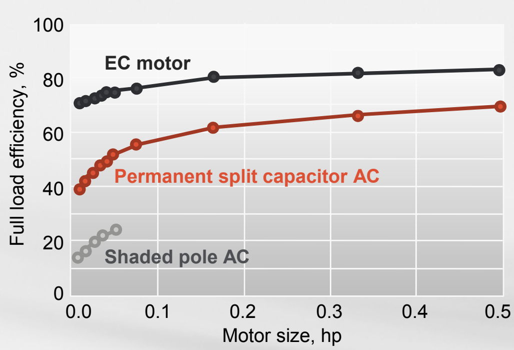
Advanced Motor Technology

ECM Performance

Variable-volume application

- Operates at lowest speed necessary to meeting the heating or cooling load
- Programmability
  - High and speeds can be adjusted
- Soft ramp in auto mode
- Longer run times at lower speeds improves dehumidification

Advanced Motor Technology



ECM Efficiency

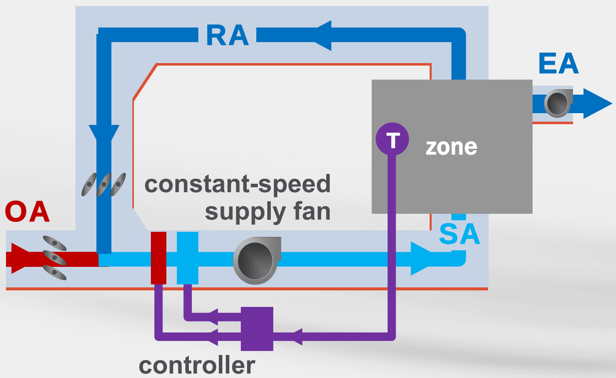
Conventional Permanent Split Capacitor (PSC) motor technology

- Full-load efficiency is typically 55% to 65%
- Performance degradation at lower speeds, down to 15% to 20%

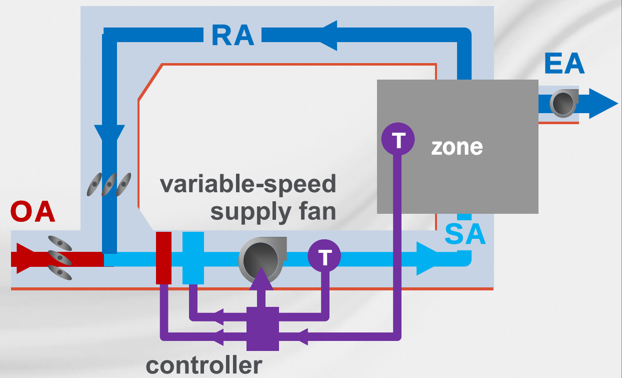
EC motor technology (brushless DC)

- Full-load efficiency can be 70% or better
- Real advantages come at part load, where efficiency can be two or three times better than conventional PSC motors

System Configurations

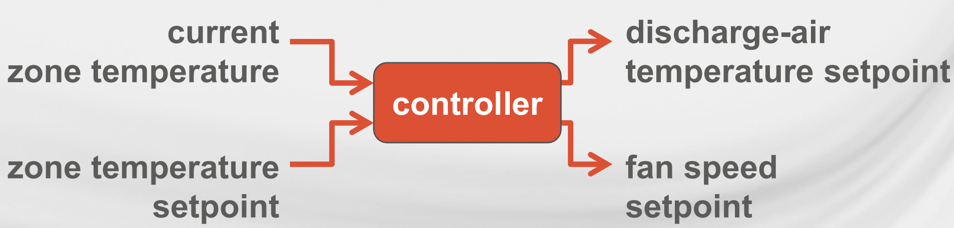


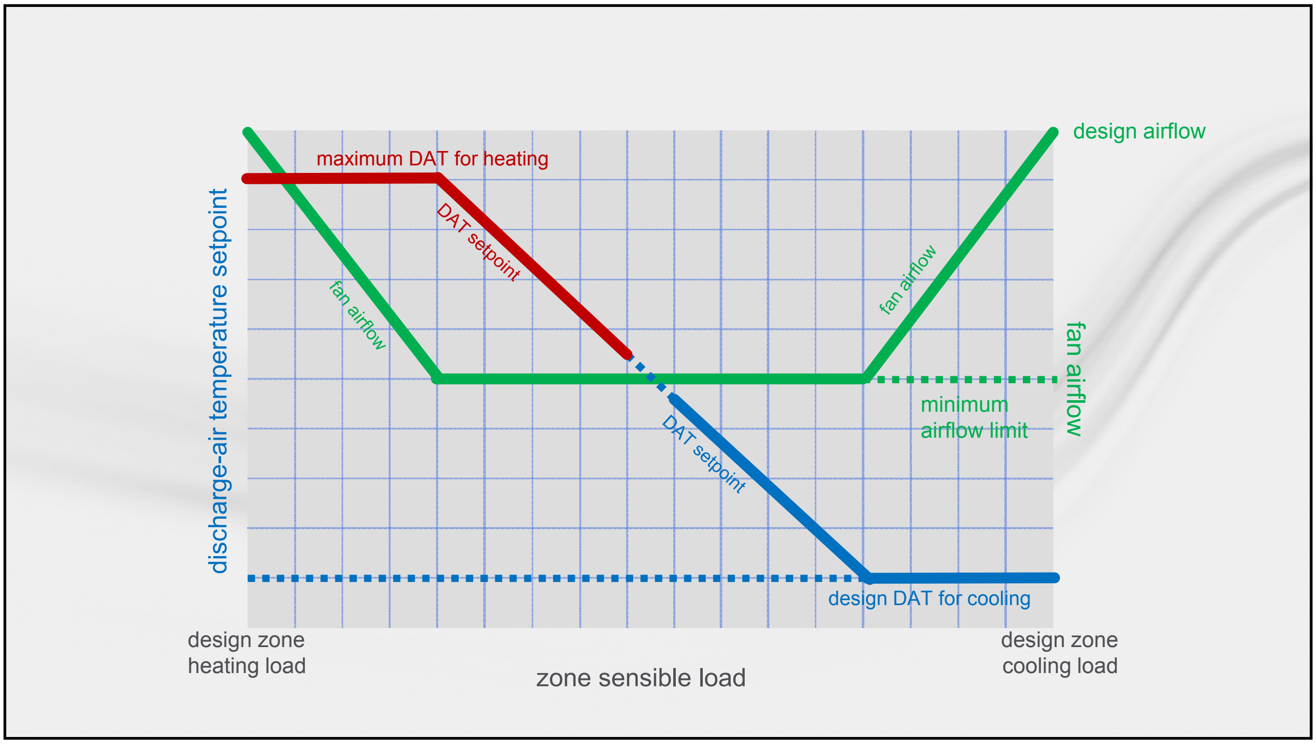
conventional, constant-volume
terminal unit control

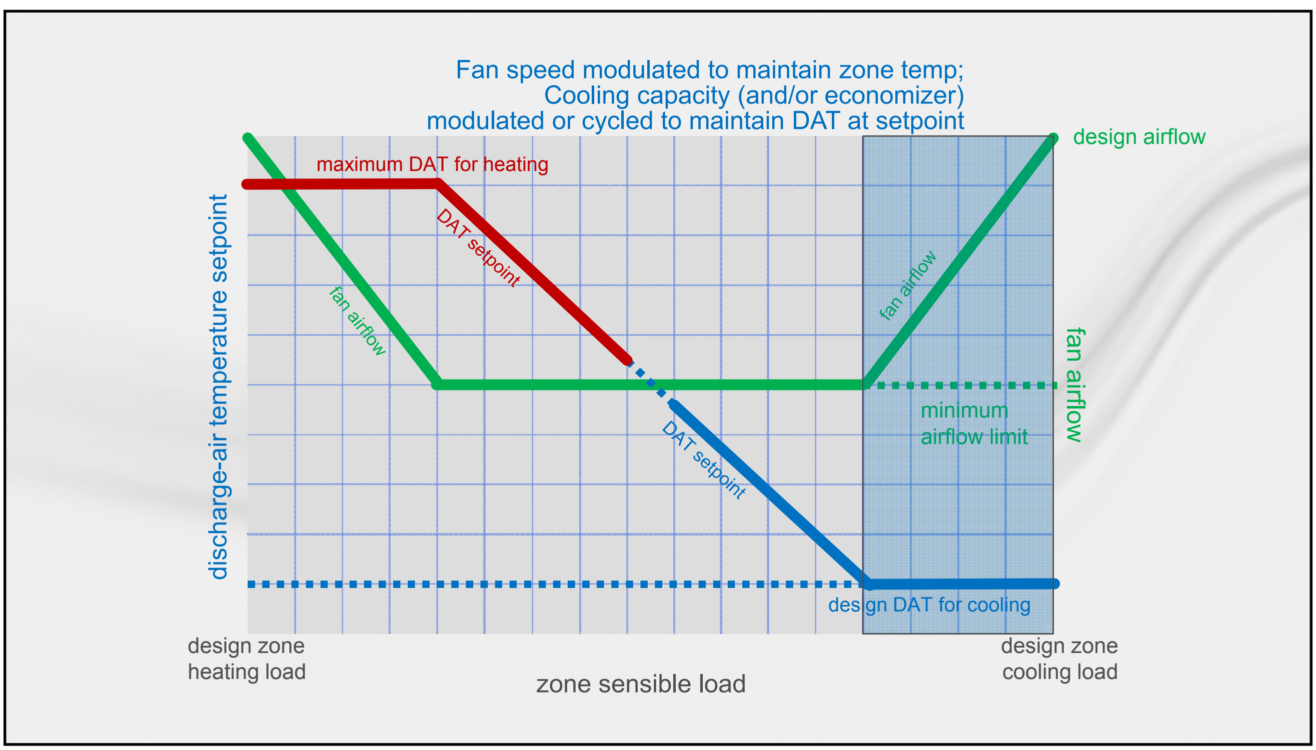


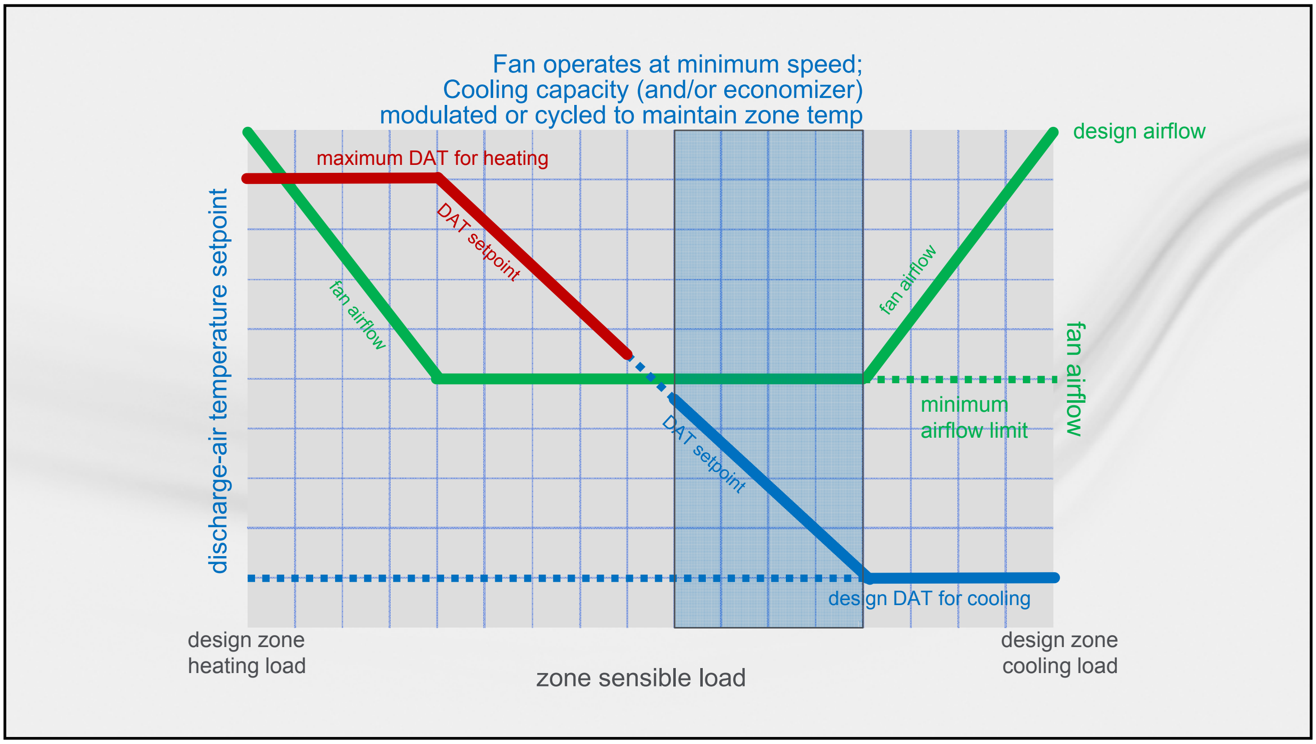
new, single-zone VAV
terminal unit control

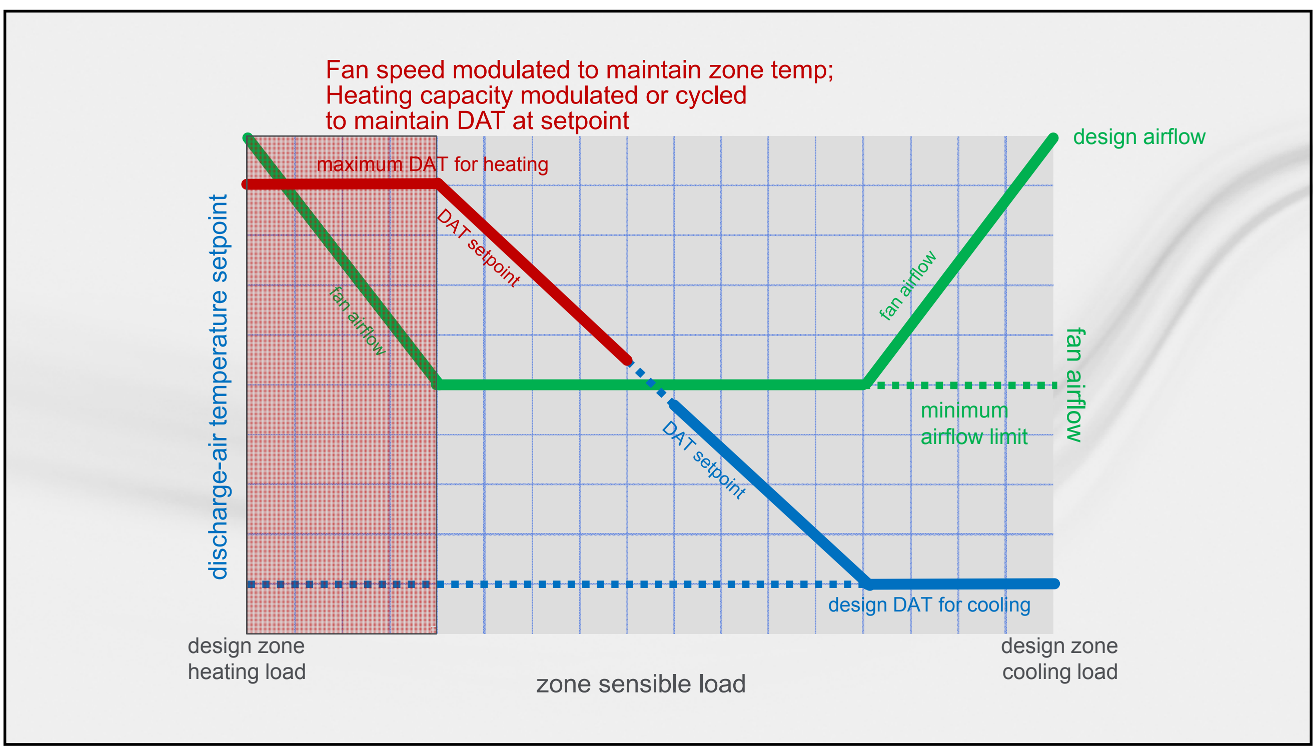
Single-Zone VAV Control




design airflow
maximum DAT for heating
DAT setpoint
fan airflow
fan airflow
discharge-air temperature setpoint
fan airflow
minimum
airflow limit
DAT setpoint
design DAT for cooling
design zone
heating load
design zone
cooling load
zone sensible load


Fan speed modulated to maintain zone temp;
Cooling capacity (and/or economizer)
modulated or cycled to maintain DAT at setpoint
design airflow
maximum DAT for heating
DAT setpoint
fan airflow
fan airflow
discharge-air temperature setpoint
fan airflow
minimum
airflow limit
DAT setpoint
design DAT for cooling
design zone
heating load
design zone
cooling load
zone sensible load


Fan operates at minimum speed;
Cooling capacity (and/or economizer)
modulated or cycled to maintain zone temp
design airflow
maximum DAT for heating
DAT setpoint
fan airflow
fan airflow
discharge-air temperature setpoint
fan airflow
minimum
airflow limit
DAT setpoint
design DAT for cooling
design zone
heating load
design zone
cooling load
zone sensible load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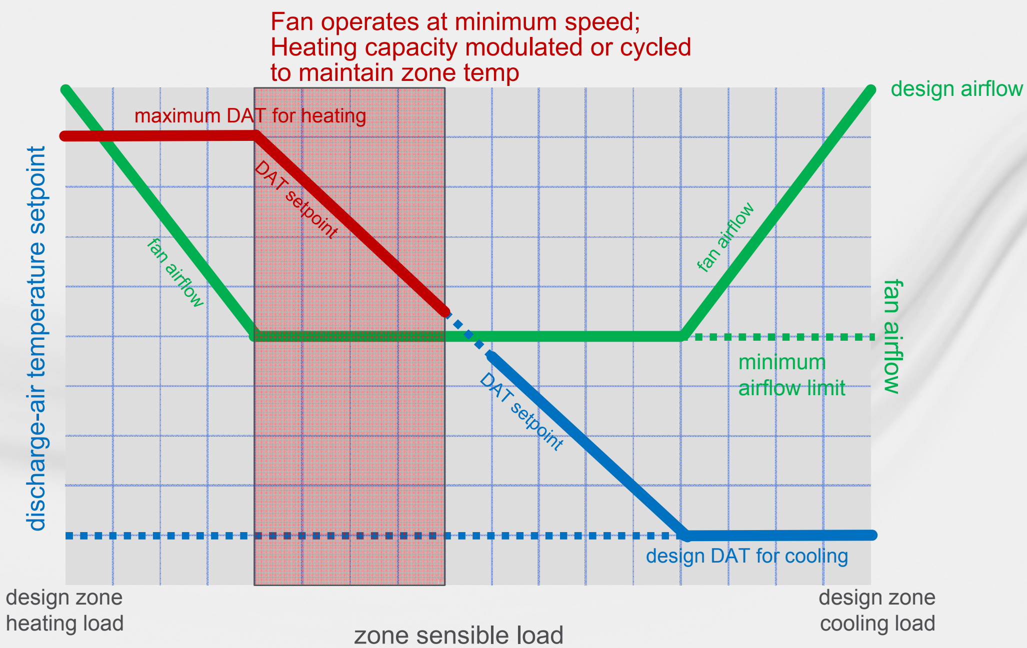
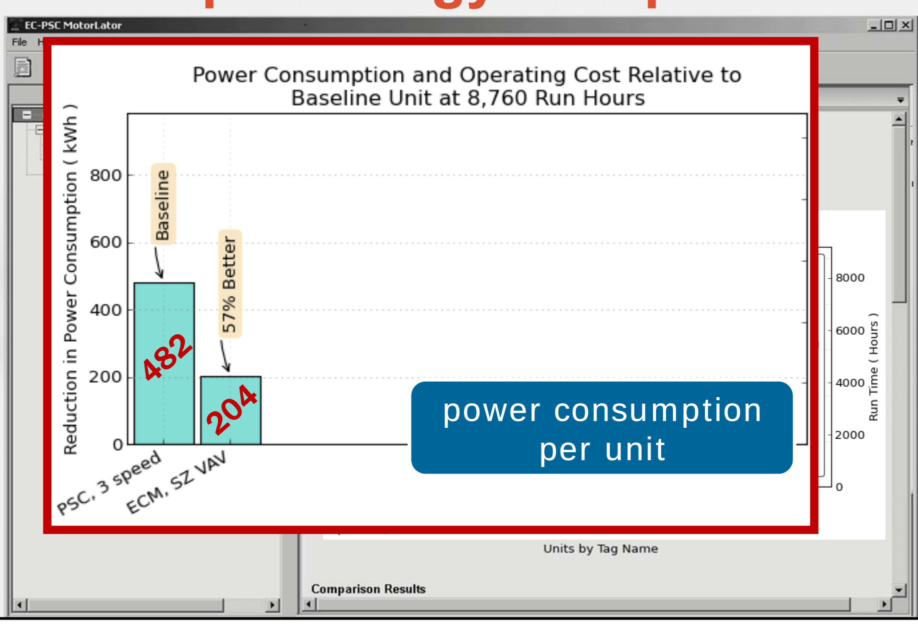


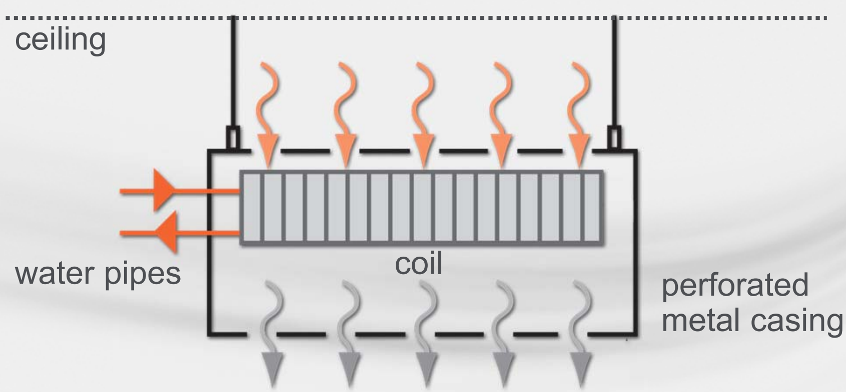
Example Energy Comparison

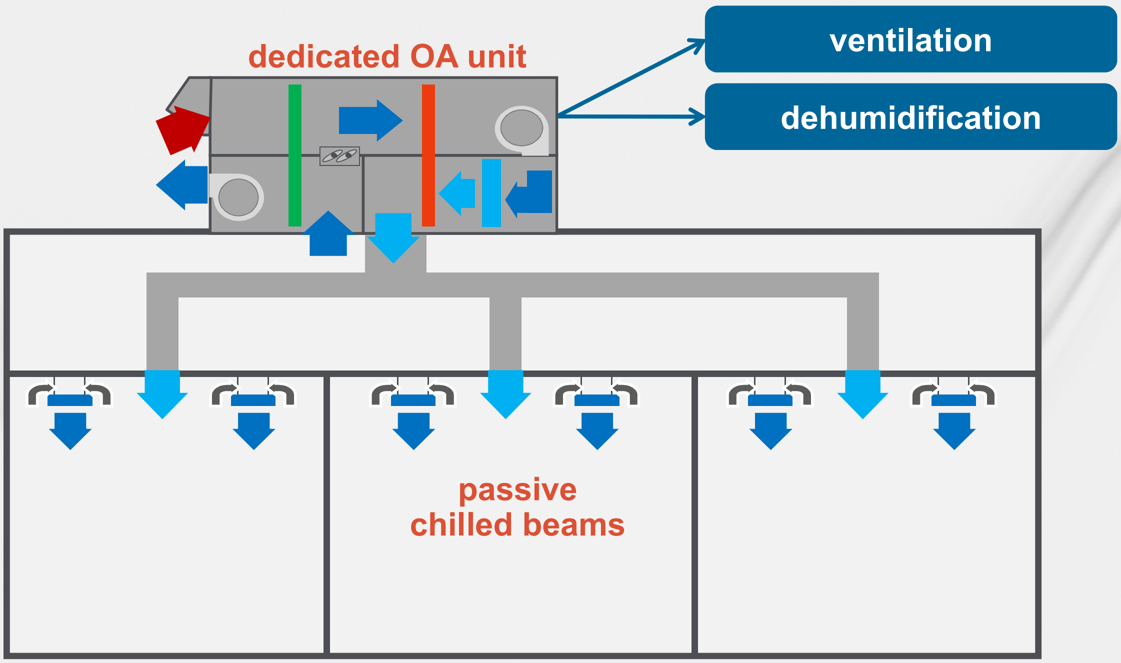


Passive Chilled Beams

Photo from Frenger Systems and FTF Group Climate
www.chilled-beams.co.uk/lancaster.htm

Passive Chilled Beam

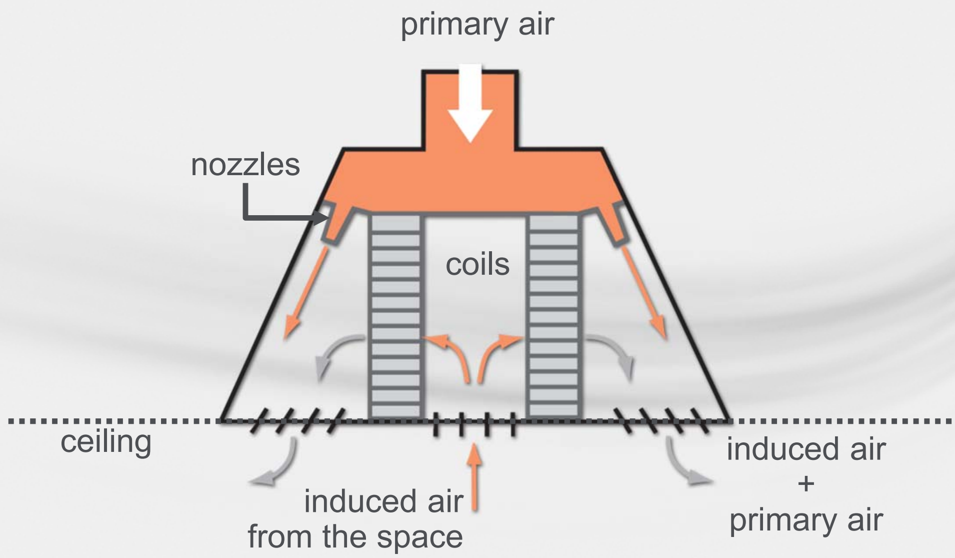


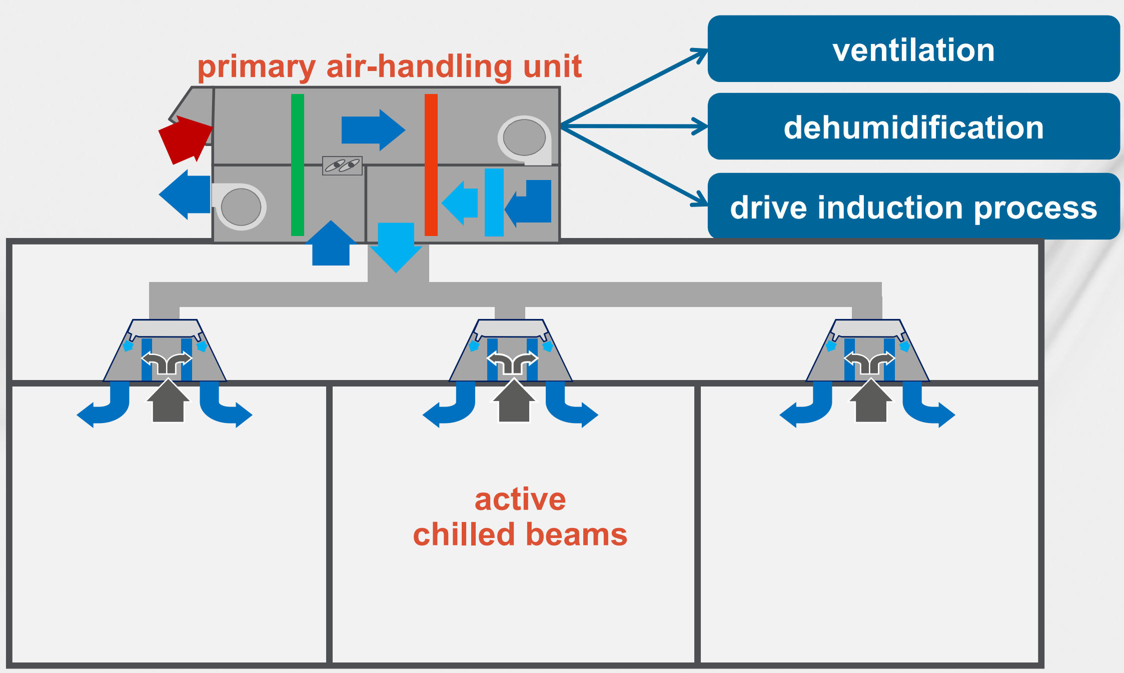


Active Chilled Beam

Photo from TROX
www.troxusa.com/usa/company/references/showcases/office_schools/Constitution_Center/index.html

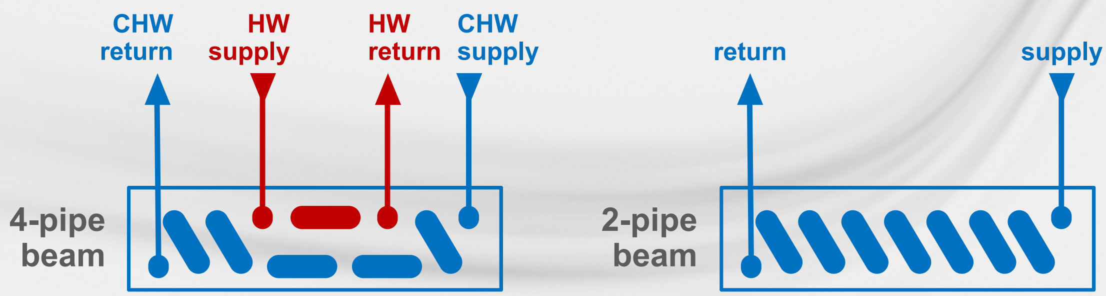
Active Chilled Beam





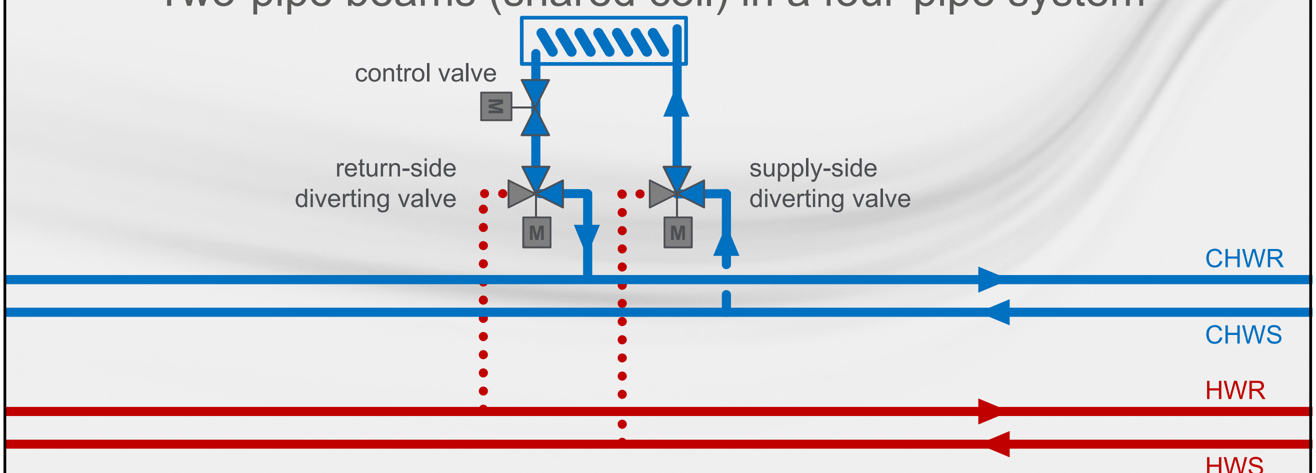
Heating with Active Chilled Beams

- Four-pipe beams



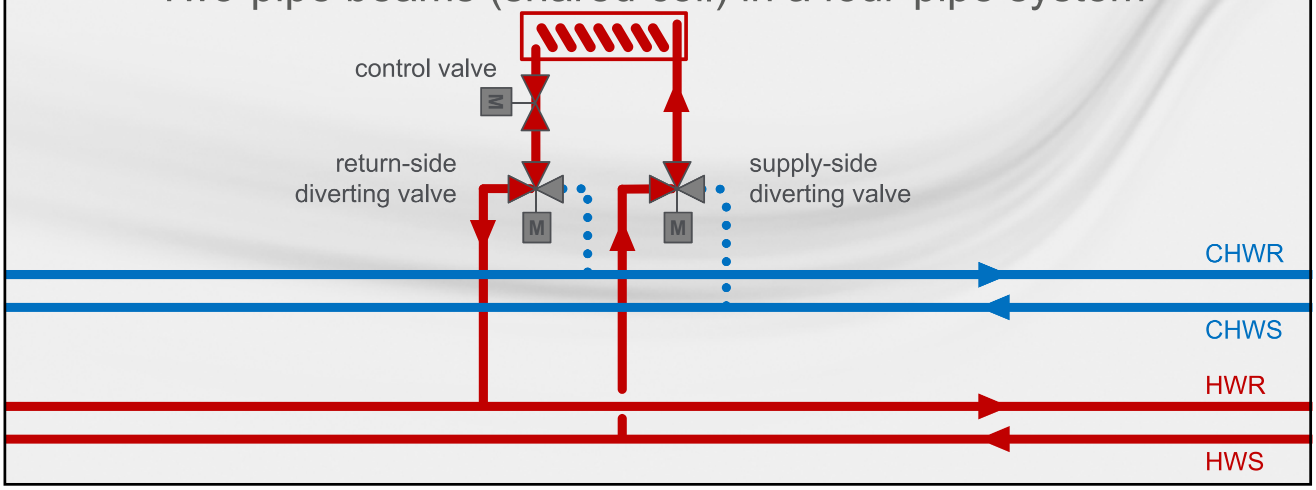
Heating with Active Chilled Beams

- Four-pipe beams
- Two-pipe beams (shared coil) in a four-pipe system



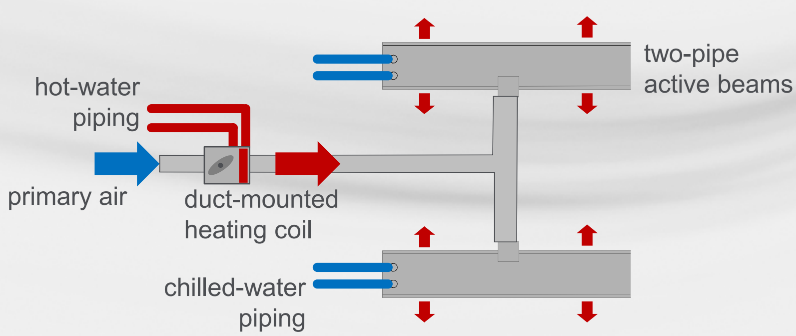
Heating with Active Chilled Beams

- Four-pipe beams
- Two-pipe beams (shared coil) in a four-pipe system



Heating with Active Chilled Beams

- Four-pipe beams
- Two-pipe beams (shared coil) in a four-pipe system
- Two-pipe beams with a heating coil in the air duct



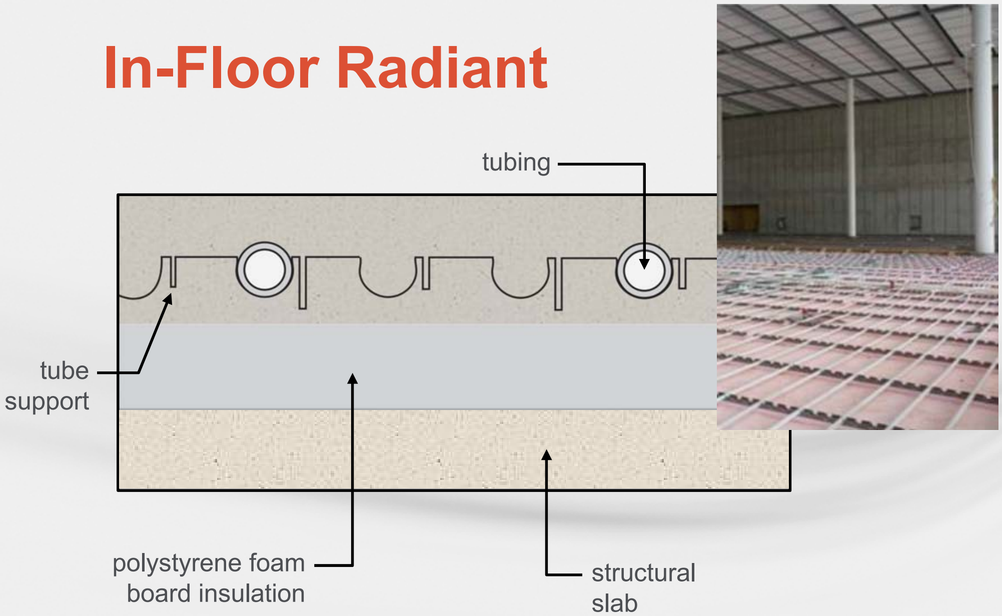
Heating with Active Chilled Beams

- Four-pipe beams
- Two-pipe beams (shared coil) in a four-pipe system
- Two-pipe beams with a heating coil in the air duct
- Separate heating system (baseboard, in-floor radiant)

Radiant Panels

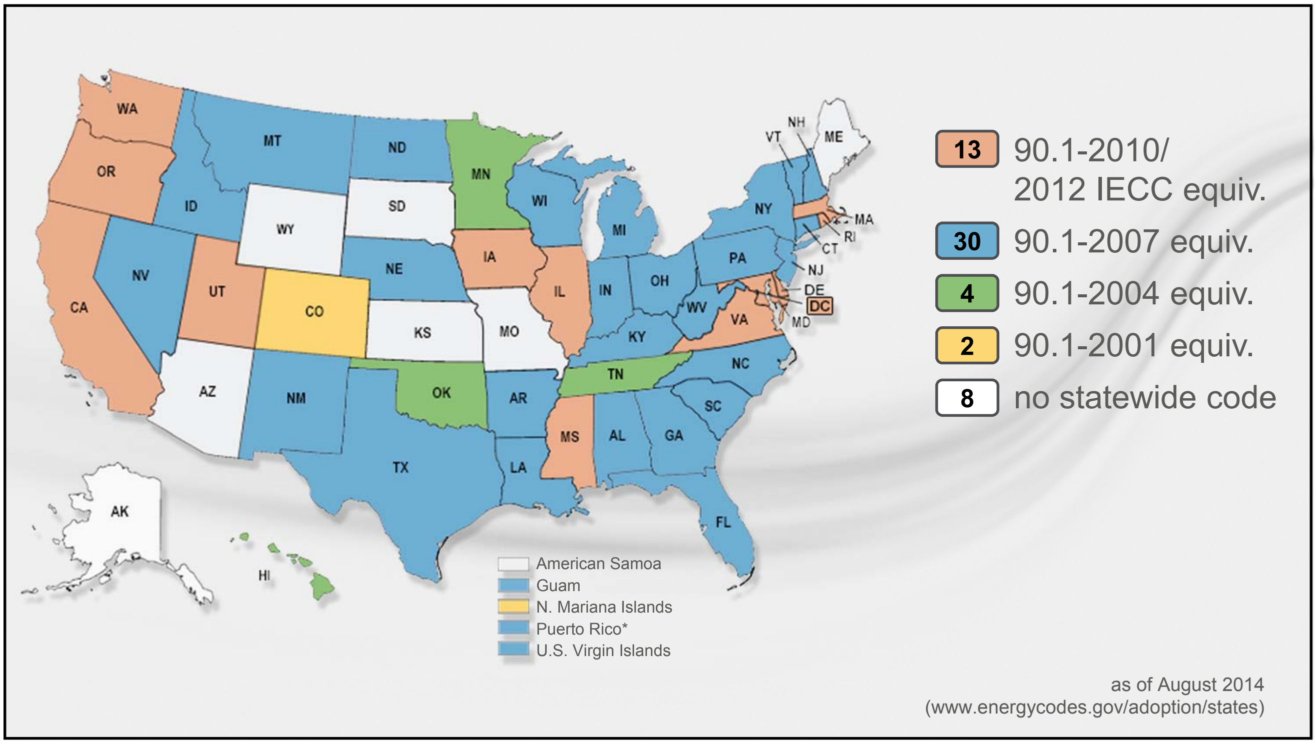
Photo from Krantz
www.krantz.de/de/Komponenten/Referenzen/Objekte/Deutschland/Seiten/John-Deere-European-Parts-Distribution-Center-Bruchsal.aspx

In-Floor Radiant



ASHRAE 90.1 Requirements

- Economizers
- Fan system power limitation


13 90.1-2010/ 2012 IECC equiv.
30 90.1-2007 equiv.
4 90.1-2004 equiv.
2 90.1-2001 equiv.
8 no statewide code
WA
OR
CA
ID
NV
MT
WY
UT
AZ
CO
NM
ND
SD
NE
KS
OK
TX
MN
IA
MO
AR
LA
WI
IL
MI
IN
OH
KY
TN
MS
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GA
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SC
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VA
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PA
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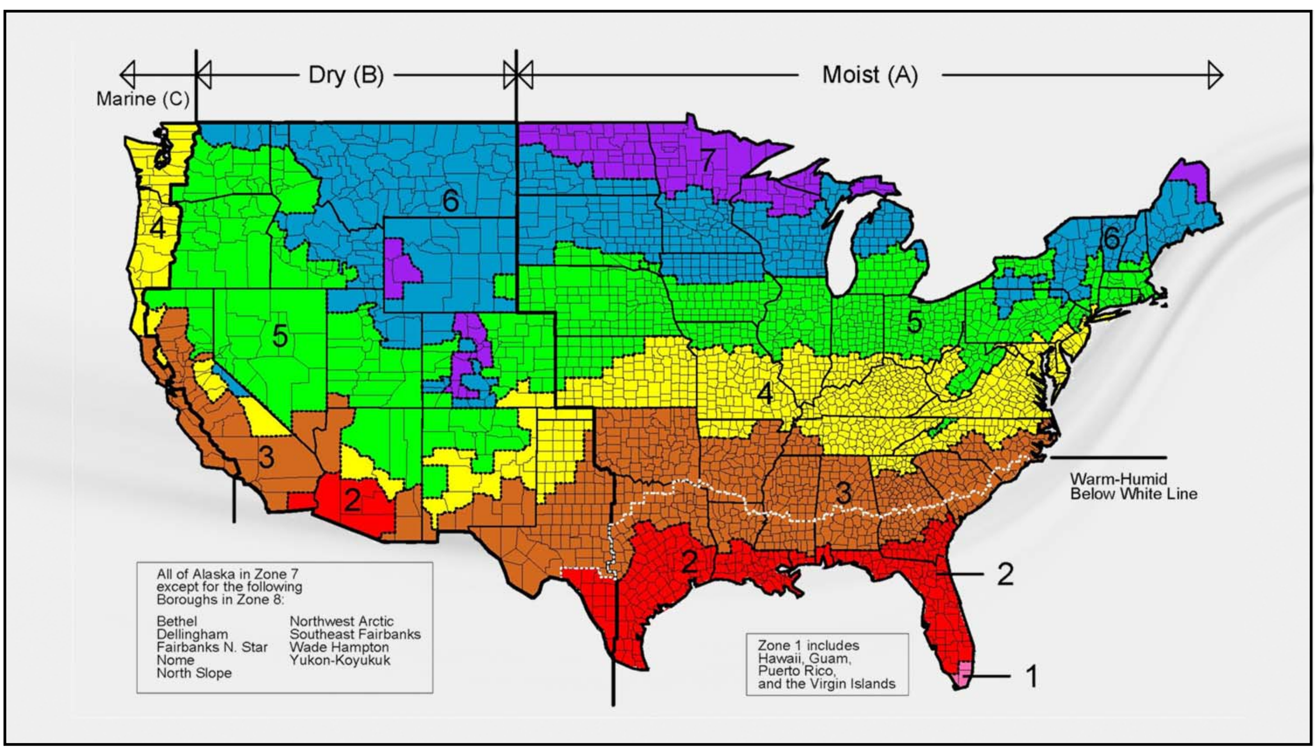
MA
RI
CT
NJ
DE
MD
DC
AK
HI
American Samoa
Guam
N. Mariana Islands
Puerto Rico*
U.S. Virgin Islands
as of August 2014
(www.energycodes.gov/adoption/states)


Marine (C)
Dry (B)
Moist (A)
7
6
5
4
3
2
1
Warm-Humid
Below White Line
All of Alaska in Zone 7 except for the following Boroughs in Zone 8:
Bethel
Dellingham
Fairbanks N. Star
Nome
North Slope
Northwest Arctic
Southeast Fairbanks
Wade Hampton
Yukon-Koyukuk
Zone 1 includes Hawaii, Guam, Puerto Rico, and the Virgin Islands

When is an Economizer Required?

6.5.1 Economizers: Each cooling system that has a fan shall include either an air or water economizer…

Exceptions: Economizers are not required for systems listed below.

a. Individual fan-cooling units with a supply capacity less than the minimum listed in the table.

90.1 Economizer Requirement

Version	2007	2010	2013
Climate zones	all except 1A - 4A and 1B	all except 1	all except 1
Cooling capacity for which an economizer is required ("system" size in Btu/h)	2b,5a,6a,7,8 ≥ 135,000 3b,3c,4b,4c,5b,5c,6b ≥ 65,000	≥ 54,000	≥ 54,000

"***Individual fan-cooling units** with a supply capacity less than the minimum listed…*"

90.1-2013 economizer example

If Performing Only Sensible Cooling…

$Q_{sensible}$ = 1.085 × CFM_{supply} × (DBT_{space} – DBT_{supply})

… assuming a 20°F ΔT (75°F DBT_{space} – 55°F DBT_{supply})

CFM_{supply} = (54,000 Btu/h) / (1.085 × 20°F) = 2,448 cfm

90.1 Fan System Power Limitation

Version	2007, 2010, 2013*	
	Constant Volume	Variable Volume
Option 1: Nameplate hp	≤ CFM_S × 0.0011	≤ CFM_S × 0.0015
Option 2: System bhp	≤ CFM_S × 0.00094 + A*	≤ CFM_S × 0.0013 + A*

* A(djustments) differ in each version of the standard.

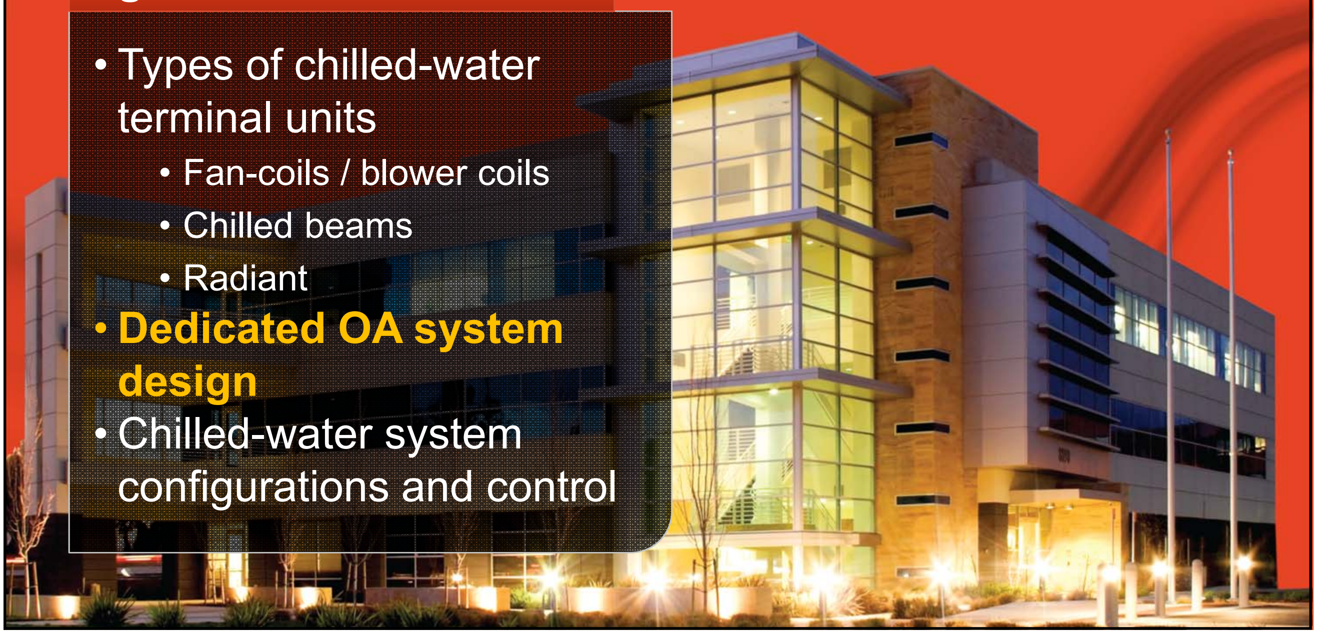
2010 and 2013 versions: *"Single-zone variable-air-volume systems shall comply with the constant-volume fan power limitation."*

Fan Power and Dedicated OA Units

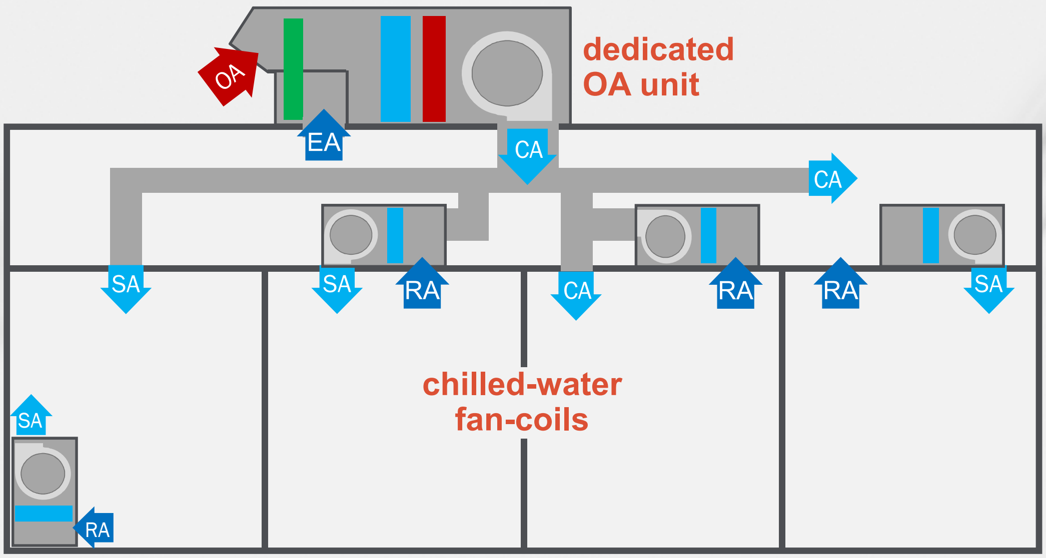
- Ventilation and return/exhaust fan power must be included
- How is fan power distributed when there are both centralized (DOAS) and terminal fans?
- Central fan power must be allocated to each terminal unit on a “CFM-weighted” basis
  - Refer to Example 6-DDD in the ASHRAE *Standard 90.1 User’s Manual*

Agenda

- Types of chilled-water terminal units
  - Fan-coils / blower coils
  - Chilled beams
  - Radiant
- **Dedicated OA system design**
- Chilled-water system configurations and control



Dedicated OA Delivery Configurations



conditioned OA delivered

Directly to Each Zone

Advantages

- Easier to ensure required outdoor airflow reaches each zone (separate diffusers)
- Opportunity to cycle off local fan because OA is not distributed through it
- Allows dedicated OA system to operate during unoccupied periods without needing to operate local fans
- Opportunity to downsize local equipment (if OA delivered cold)

Drawbacks

- Requires installation of additional ductwork and separate diffusers
- May require multiple diffusers to ensure that outdoor air is adequately dispersed throughout the zone

conditioned OA delivered

To Intake of Local HVAC Equipment

Advantages

- Helps ensure required OA reaches each zone (ducted directly to each unit)
- Avoids cost and space to install additional ductwork and separate diffusers
- Easier to ensure that OA is adequately dispersed throughout zone because it is distributed by local fan

Drawbacks

- Measurement and balancing is more difficult than if OA delivered directly to zone
- Typically requires field-fabricated plenum to connect OA duct to mix with RA
- Local fan must operate continuously to provide OA during scheduled occupancy
- Local fan must operate if dedicated OA system operates during unoccupied period

conditioned OA delivered

To Supply-Side of Local HVAC Equipment

Advantages

- Helps ensure required OA reaches each zone (ducted directly to each unit)
- Avoids cost and space to install additional ductwork and separate diffusers
- Easier to ensure that OA is adequately dispersed throughout zone because it is distributed by local fan
- Opportunity to downsize local equipment (if OA delivered cold)

Drawbacks

- Measurement and balancing is more difficult than if OA delivered directly to zone
- Local fan typically must operate continuously to provide OA during scheduled occupancy (unless pressure-independent VAV terminal)

conditioned OA delivered

To Plenum, Near Local HVAC Equipment

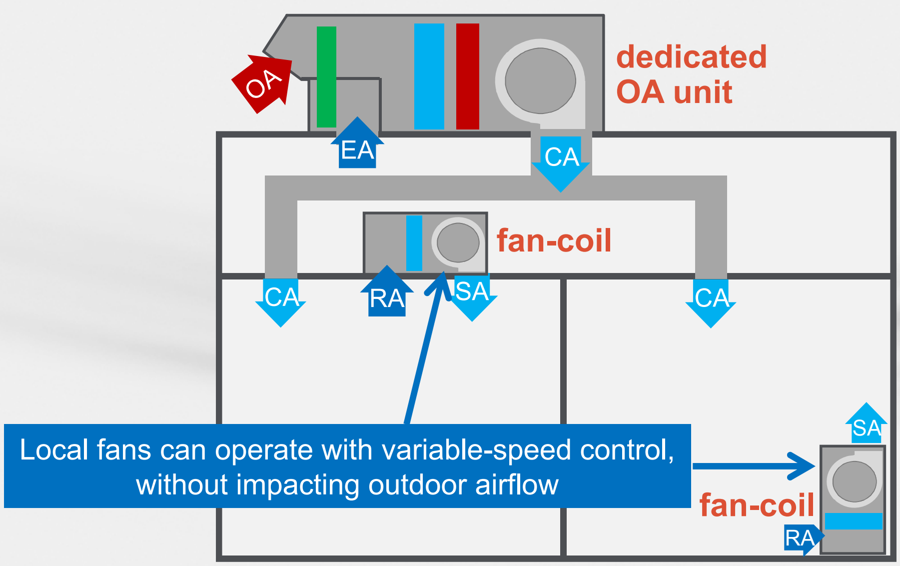
Advantages

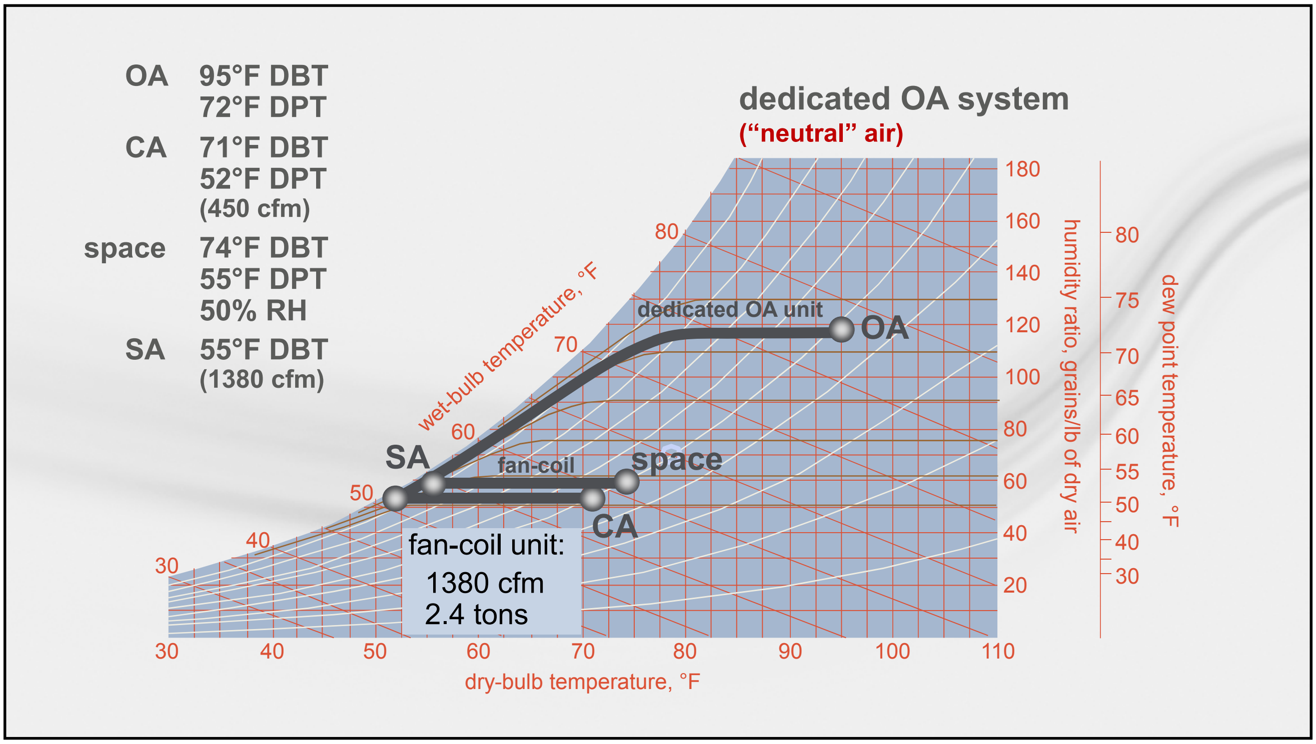
- Avoids cost and space to install additional ductwork and separate diffusers

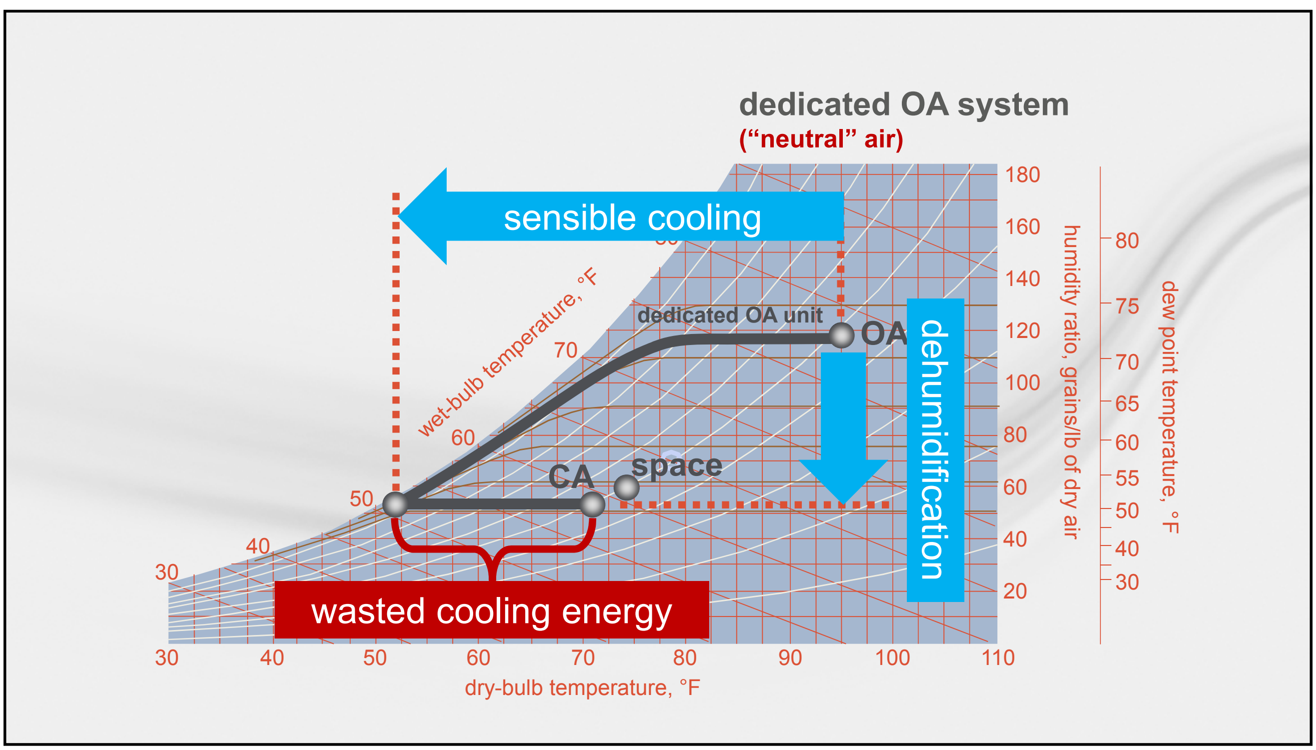
Drawbacks

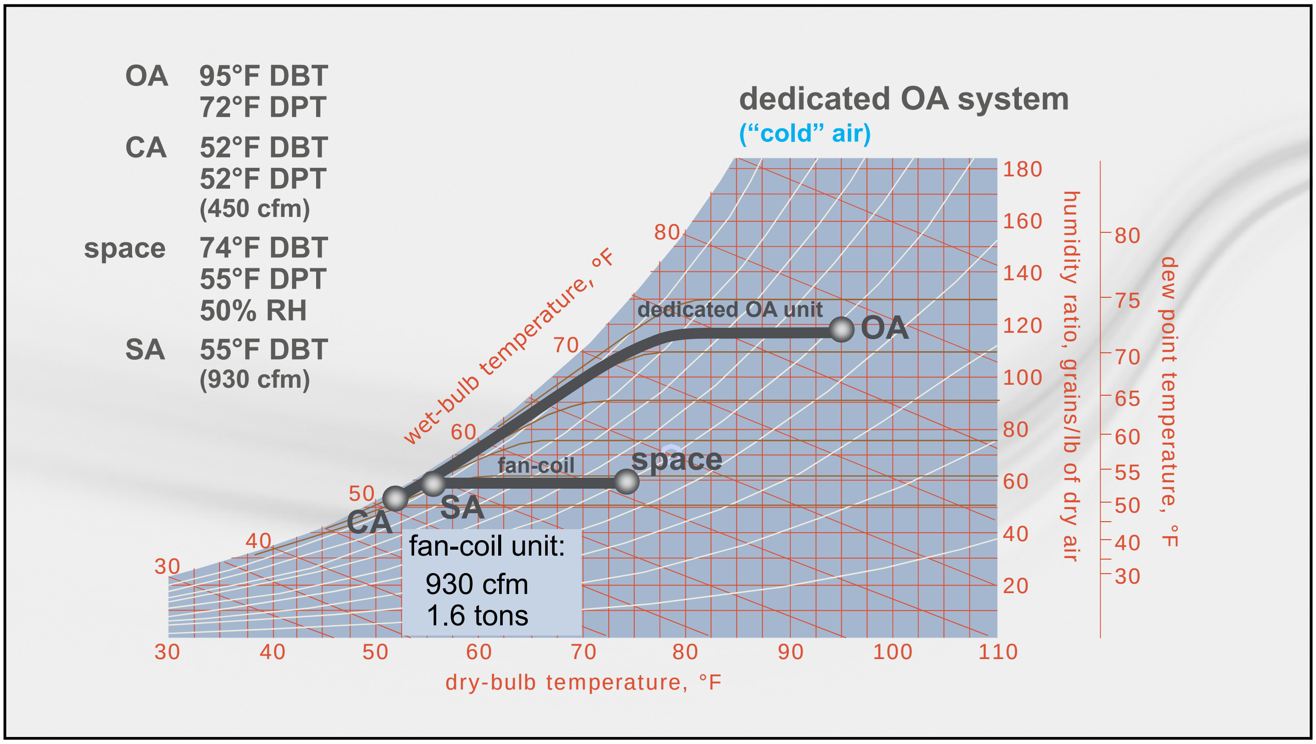
- More difficult to ensure required OA reaches each zone (not ducted directly)
  - Refer to Figure 5-E and 5-F of *ASHRAE 62.1-2010 User's Manual*
- Local fan must operate continuously to provide OA during scheduled occupancy
- Conditioned OA not able to be delivered at a cold temperature due to concerns over condensation

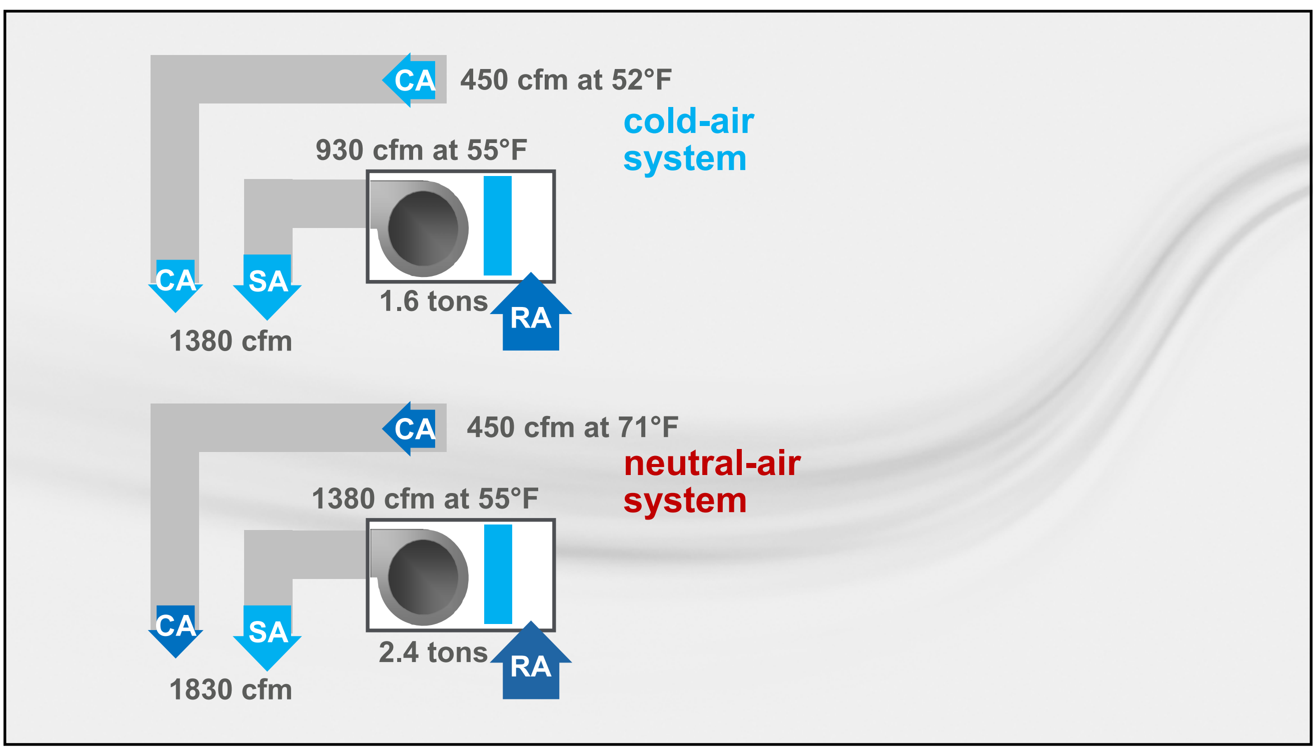
Delivered Directly to Each Zone



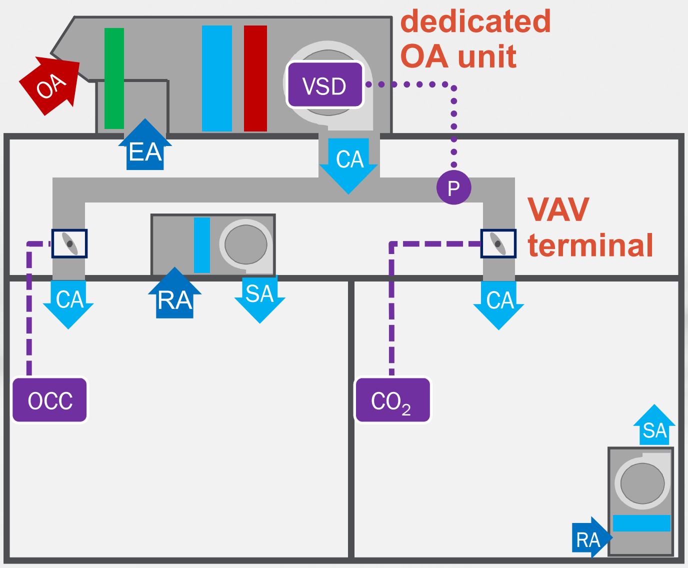

OA 95°F DBT
72°F DPT
CA 71°F DBT
52°F DPT
(450 cfm)
space 74°F DBT
55°F DPT
50% RH
SA 55°F DBT
(1380 cfm)
dedicated OA system
("neutral" air)
dedicated OA unit
OA
SA
fan-coil
space
CA
fan-coil unit:
1380 cfm
2.4 tons
wet-bulb temperature, °F
dry-bulb temperature, °F
humidity ratio, grains/lb of dry air
dew point temperature, °F


dedicated OA system
("neutral" air)
sensible cooling
dehumidification
dedicated OA unit
OA
CA
space
wasted cooling energy
wet-bulb temperature, °F
dry-bulb temperature, °F
humidity ratio, grains/lb of dry air
dew point temperature, °F

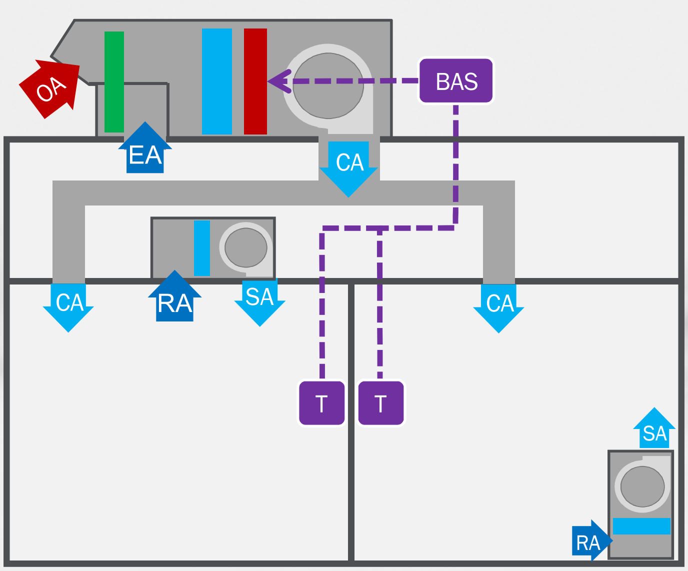

OA 95°F DBT
72°F DPT
CA 52°F DBT
52°F DPT
(450 cfm)
space 74°F DBT
55°F DPT
50% RH
SA 55°F DBT
(930 cfm)
dedicated OA system
("cold" air)
dedicated OA unit
OA
fan-coil
space
CA
SA
fan-coil unit:
930 cfm
1.6 tons
wet-bulb temperature, °F
humidity ratio, grains/lb of dry air
dew point temperature, °F
dry-bulb temperature, °F


CA 450 cfm at 52°F
cold-air system
930 cfm at 55°F
CA
SA
1.6 tons
RA
1380 cfm
CA 450 cfm at 71°F
neutral-air system
1380 cfm at 55°F
CA
SA
2.4 tons
RA
1830 cfm

What About Overcooling a Zone?



What About Overcooling a Zone?



When Should I Reheat Dehumidified OA?

- To avoid overcooling at part-load conditions
  - Implement demand-controlled ventilation
  - Activate heat in the local HVAC unit
  - Reheat dehumidified air in dedicated OA unit
- Applications where space sensible cooling loads differ greatly at any given time (e.g., hotels, dormitories)
- Applications requiring lower-than-normal dew points
- To avoid condensation when conditioned OA is delivered to the ceiling plenum

90.1 DCV Requirement

Version	2007	2010	2013
Zone size, ft^2	> 500	> 500	> 500
People/1000 ft^2	> 40	> 40	≥ 25

"… and served by systems with one or more of the following:

a. an airside economizer,

b. automatic modulating control of the outdoor air damper, or

c. a design outdoor airflow greater than 3000 cfm…"

90.1 Energy Recovery Requirement

Version	2007	2010	2013
Climate zones	1A - 6A and 1B - 4B	all	all
Lowest %OA	70%	30%	10%
Lowest airflow, cfm	5000	> 0	> 0
Hours of operation	N/A	N/A	< 8000 and ≥ 8000

90.1-2010: Energy Recovery

	% Outdoor Air at Full Design Airflow Rate					
≥	30%	40%	50%	60%	70%	80%
and <	40%	50%	60%	70%	80%	
Climate Zone	Design Supply Fan Airflow Rate, cfm					
3B, 3C, 4B, 4C, 5B	NR	NR	NR	NR	≥5000	≥5000
1B, 2B, 5C	NR	NR	≥26000	≥12000	≥5000	≥4000
6B	≥11000	≥5500	≥4500	≥3500	≥2500	≥1500
1A, 2A, 3A, 4A, 5A, 6A	≥5500	≥4500	≥3500	≥2000	≥1000	>0
7, 8	≥2500	≥1000	>0	>0	>0	>0

90.1-2013: Energy Recovery

operating < 8000 hours per year

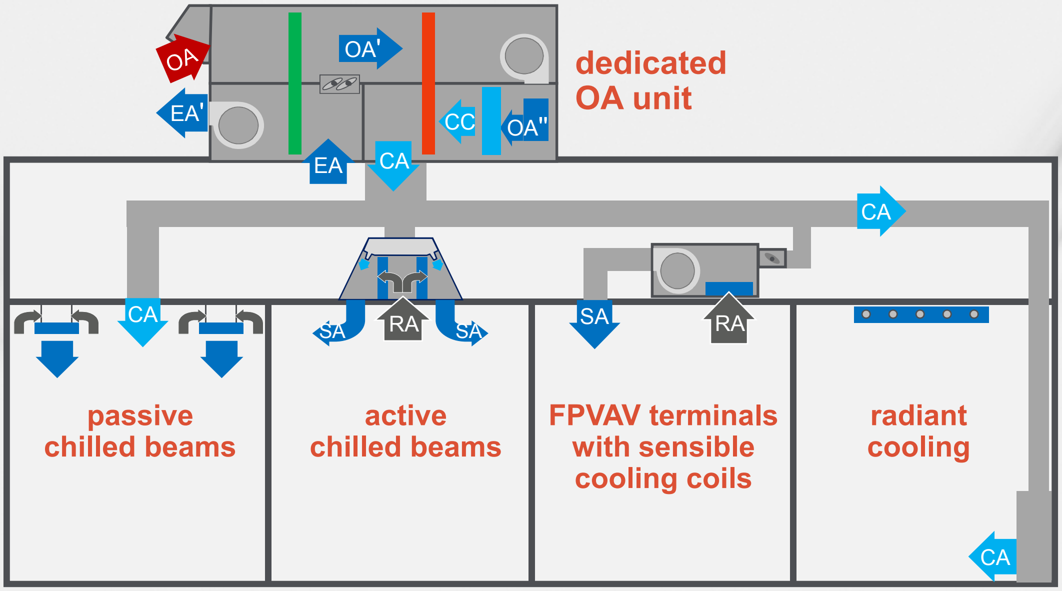
	% Outdoor Air at Full Design Airflow Rate							
≥	10%	20%	30%	40%	50%	60%	70%	80%
and <	20%	30%	40%	50%	60%	70%	80%	
Climate Zone	Design Supply Fan Airflow Rate, cfm							
3B, 3C, 4B, 4C, 5B	NR	NR	NR	NR	NR	NR	NR	NR
1B, 2B, 5C	NR	NR	NR	NR	≥26000	≥12000	≥5000	≥4000
6B	≥28000	≥26500	≥11000	≥5500	≥4500	≥3500	≥2500	≥1500
1A, 2A, 3A, 4A, 5A, 6A	≥26000	≥16000	≥5500	≥4500	≥3500	≥2000	≥1000	>0
7, 8	≥4500	≥4000	≥2500	≥1000	>0	>0	>0	>0

90.1-2013: Energy Recovery

operating ≥ 8000 hours per year

	% Outdoor Air at Full Design Airflow Rate							
≥	10%	20%	30%	40%	50%	60%	70%	80%
and <	20%	30%	40%	50%	60%	70%	80%	
Climate Zone	Design Supply Fan Airflow Rate, cfm							
3C	NR	NR	NR	NR	NR	NR	NR	NR
1B, 2B, 3B, 4C, 5C	NR	≥19500	≥9000	≥5000	≥4000	≥3000	≥1500	>0
1A, 2A, 3A, 4B, 5B	≥2500	≥2000	≥1000	≥500	>0	>0	>0	>0
4A, 5A, 6A, 6B, 7, 8	>0	>0	>0	>0	>0	>0	>0	>0

Dedicated OA Delivery Configurations



chilled beams, radiant cooling, sensible cooling coils

Avoiding Condensation

- Air system is used to control indoor dew point (typically below 55°F)
- Water is delivered to terminals at a temperature a few degrees above the indoor dew point (typically between 57°F and 60°F)

passive **chilled beams, radiant cooling, sensible cooling coils**

Air System Requirements

The air system must:

- Deliver the minimum outdoor airflow required by code to each zone (example: ASHRAE Standard 62.1)
- Deliver this air dry enough to offset the latent load in each zone and maintain indoor dew point at or below the desired limit (example: 55°F dew point)

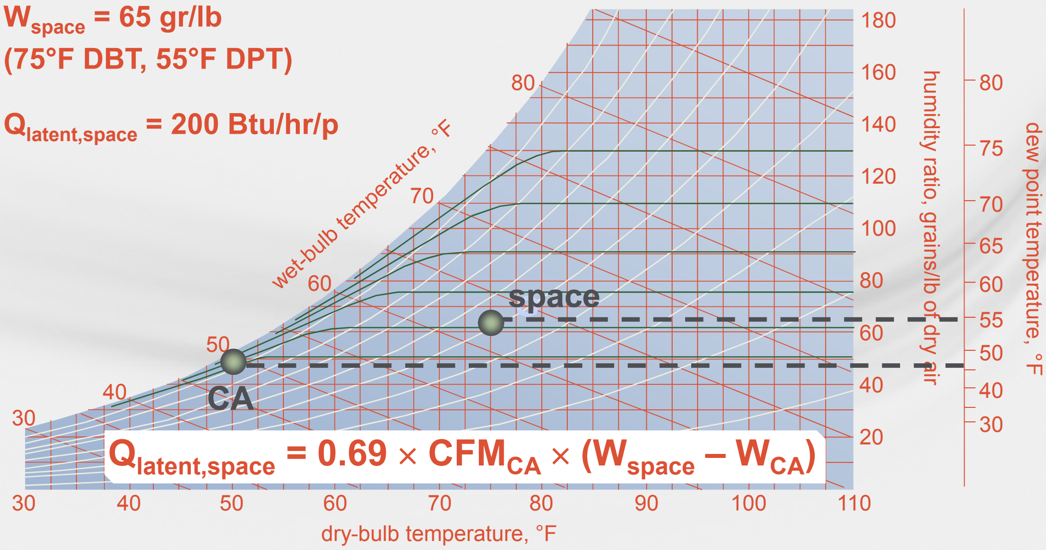
Example: Office Space

Minimum OA (ASHRAE 62.1) (to earn LEED credit)	**85 cfm** **(85 × 1.3 = 110 cfm)**
Airflow required to offset space latent load (ex: 1000 Btu/hr)	

Conditioned OA must be drier than the space…

W_{space} = 65 gr/lb
(75°F DBT, 55°F DPT)

$Q_{latent,space}$ = 200 Btu/hr/p



Example: Office Space

Minimum OA (ASHRAE 62.1) (to earn LEED credit)	**85 cfm** **(85 × 1.3 = 110 cfm)**	
Airflow required to offset space latent load (ex: 1000 Btu/hr)	**85 cfm**	**(DPT_{CA} = 47°F)**
	110 cfm	**(DPT_{CA} = 49°F)**
	360 cfm	**(DPT_{CA} = 53°F)**

Calculations: Office Space Example

Minimum OA required (ASHRAE 62.1-2010)

$V_{oz} = V_{bz} / E_z = (R_p \times P_z + R_a \times A_z) / E_z$

where,

R_p = 5 cfm/person
R_a = 0.06 cfm/ft²
P_z = 5 people
A_z = 1000 ft²
E_z = 1.0

$V_{oz} = (5 \times 5 + 0.06 \times 1000) / 1.0$
$= 85$ cfm

LEED "Increased Ventilation" credit

$V_{oz} = 85$ cfm × 1.3 = 110 cfm

Airflow required to offset space latent load

$Q_{space,latent} = 0.69 \times CFM_{CA} \times (W_{space} - W_{CA})$

where,

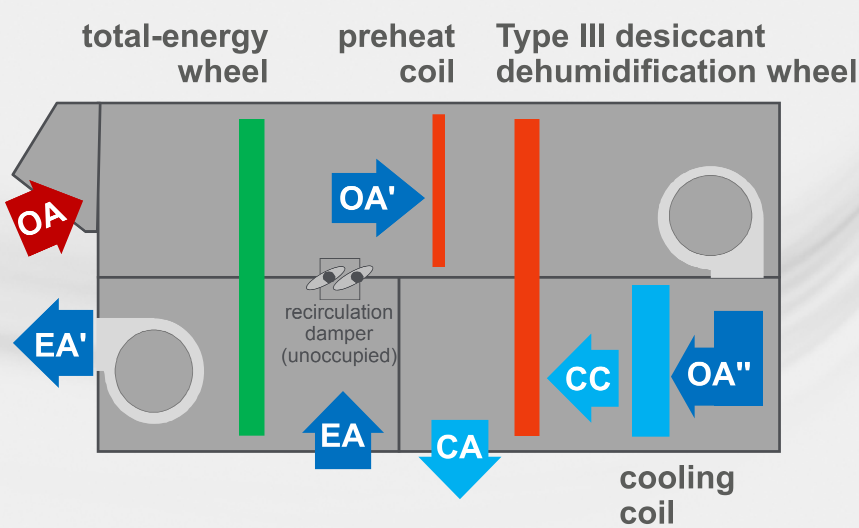
$Q_{space,latent}$ = 200 Btu/h/person × 5 people
W_{space} = 65 gr/lb (75°F DBT, 55°F DPT)

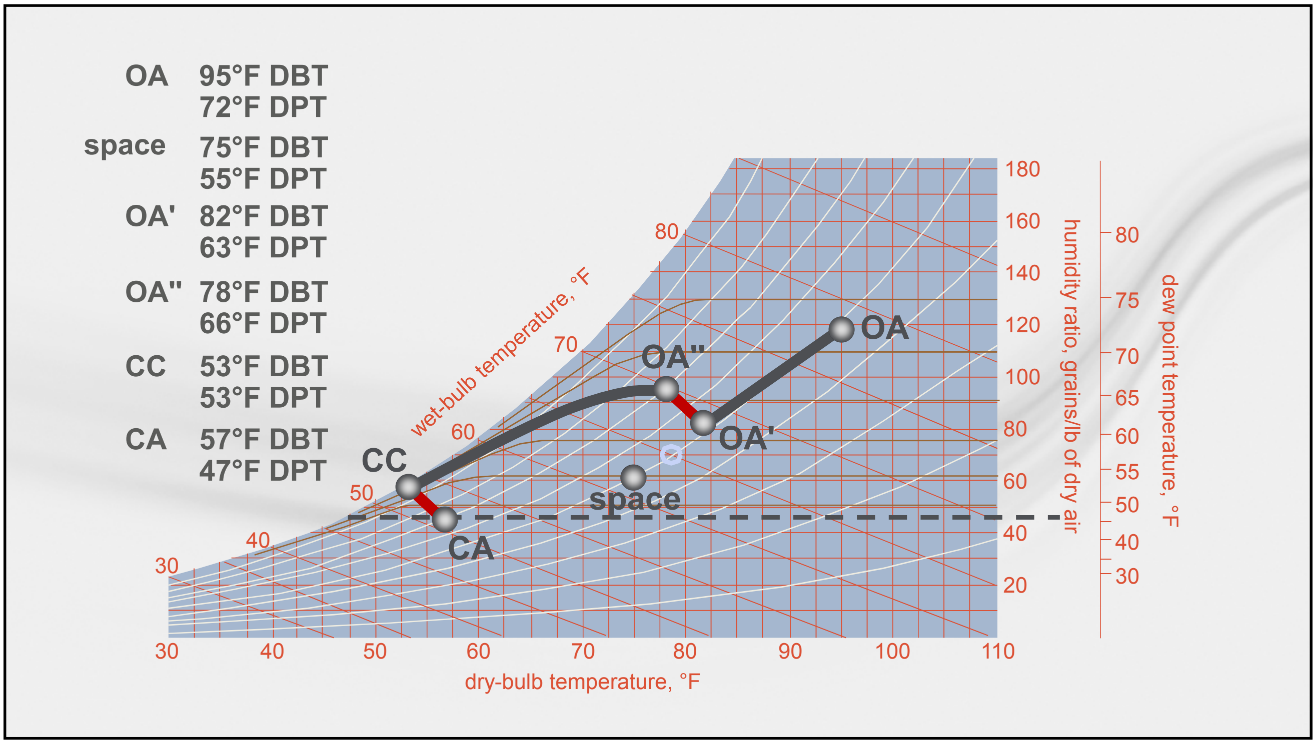
1000 Btu/h = 0.69 × 85 cfm × (65 gr/lb – W_{CA})
W_{CA} = 48 gr/lb (DPT$_{CA}$ = 47°F)

1000 Btu/h = 0.69 × 110 cfm × (65 gr/lb – W_{CA})
W_{CA} = 52 gr/lb (DPT$_{CA}$ = 49°F)

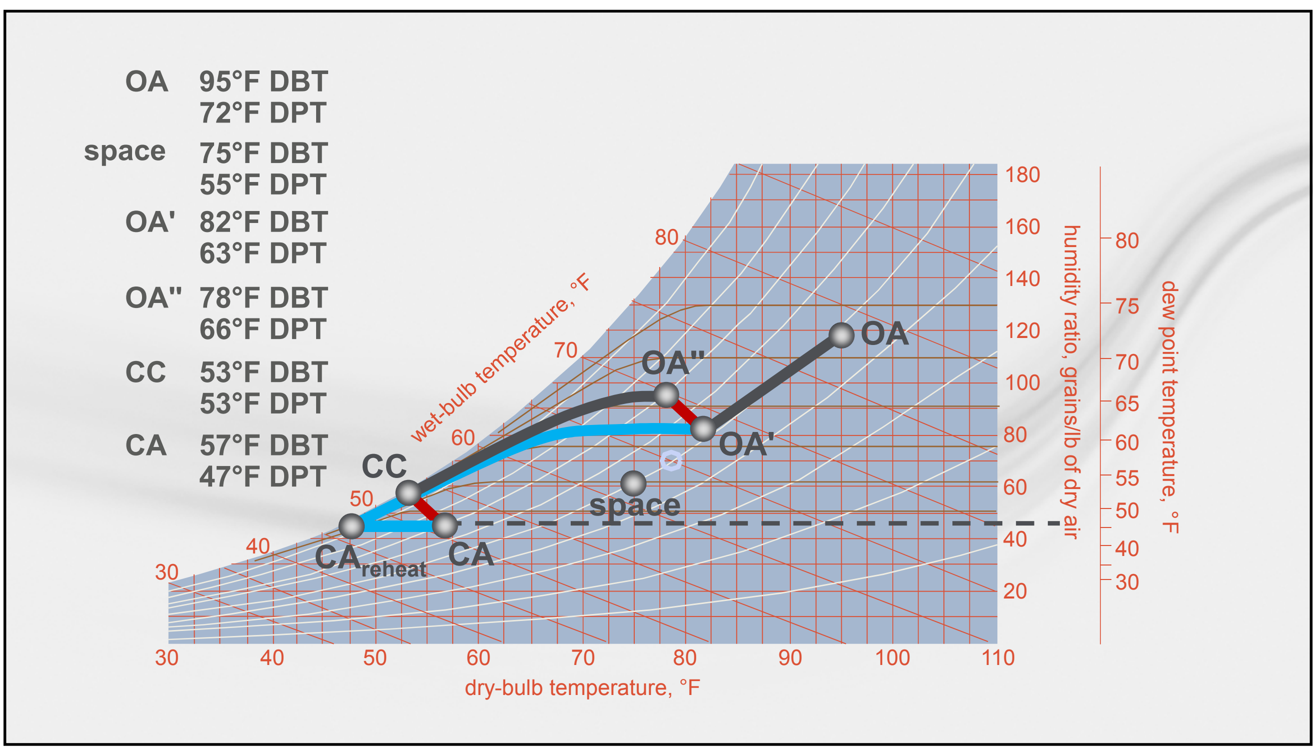
1000 Btu/h = 0.69 × 360 cfm × (65 gr/lb – W_{CA})
W_{CA} = 61 gr/lb (DPT$_{CA}$ = 53°F)

Dual-Wheel Dedicated OA Unit

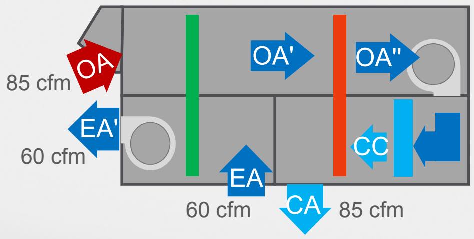
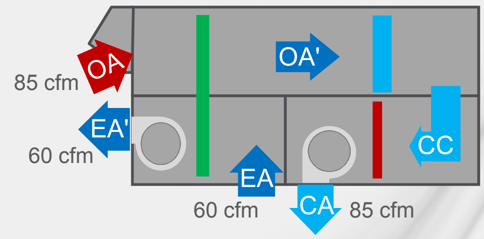


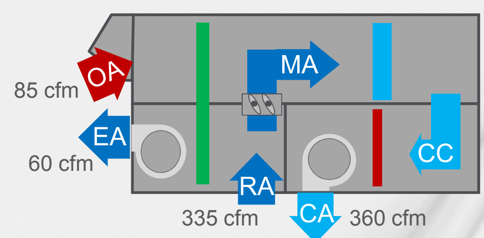


OA 95°F DBT 72°F DPT
space 75°F DBT 55°F DPT
OA' 82°F DBT 63°F DPT
OA" 78°F DBT 66°F DPT
CC 53°F DBT 53°F DPT
CA 57°F DBT 47°F DPT
wet-bulb temperature, °F
humidity ratio, grains/lb of dry air
dew point temperature, °F
dry-bulb temperature, °F
OA
OA"
OA'
CC
CA
space

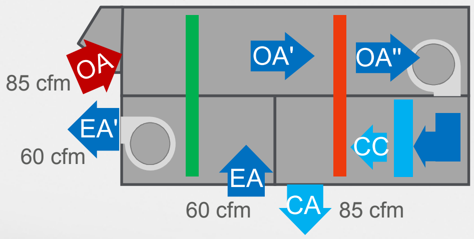


OA 95°F DBT 72°F DPT
space 75°F DBT 55°F DPT
OA' 82°F DBT 63°F DPT
OA" 78°F DBT 66°F DPT
CC 53°F DBT 53°F DPT
CA 57°F DBT 47°F DPT
wet-bulb temperature, °F
humidity ratio, grains/lb of dry air
dew point temperature, °F
dry-bulb temperature, °F
OA
OA"
OA'
CC
CA
CAreheat
space

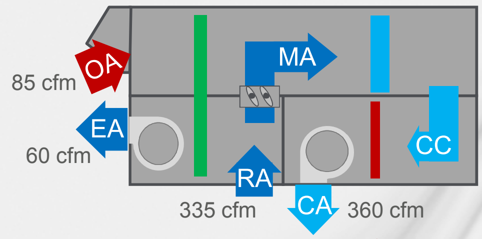
	dedicated OA unit w/ total-energy wheel and Type III desiccant wheel		dedicated OA unit w/ total-energy wheel
			
CFM_{CA}	85 cfm		85 cfm
CFM_{OA}	85 cfm		85 cfm
DPT_{CA}	47°F		47°F
DBT_{CA}	57°F		57°F
DBT_{CC} (affects CHW temp)	53°F	← warmer coil temp	47°F
AHU cooling coil	0.4 tons	← fewer tons	0.5 tons
Terminal unit coil	1.5 tons		1.5 tons
AHU supply fan	0.11 bhp	← more fan power	0.08 bhp
AHU reheat coil	0 MBh	← less or no reheat	0.73 MBh

			mixed-air unit w/ total-energy wheel
			
CFM_{CA}	85 cfm	larger ducts →	360 cfm
CFM_{OA}	85 cfm		85 cfm
DPT_{CA}	47°F	higher DPT →	53°F
DBT_{CA}	57°F		57°F
DBT_{CC} (affects CHW temp)	47°F	warmer coil temp →	55°F
AHU cooling coil	0.5 tons	more AHU tons →	0.9 tons
Terminal unit coil	1.5 tons	smaller terminal →	1.1 tons
AHU supply fan	0.08 bhp	more fan power →	0.31 bhp
AHU reheat coil	0.73 MBh	less or no reheat →	0 MBh

dedicated OA unit
w/ total-energy wheel
and Type III desiccant wheel



mixed-air unit
w/ total-energy wheel



CFM_{CA}	85 cfm		85 cfm	larger ducts →	360 cfm
CFM_{OA}	85 cfm		85 cfm		85 cfm
DPT_{CA}	47°F		47°F	higher DPT →	53°F
DBT_{CA}	57°F		57°F		57°F
DBT_{CC} (affects CHW temp)	53°F	← warmer coil temp	47°F	warmer coil temp →	55°F
AHU cooling coil	0.4 tons	← fewer tons	0.5 tons	more AHU tons →	0.9 tons
Terminal unit coil	1.5 tons		1.5 tons	smaller terminal →	1.1 tons
AHU supply fan	0.11 bhp	← more fan power	0.08 bhp	more fan power →	0.31 bhp
AHU reheat coil	0 MBh	← less or no reheat	0.73 MBh	less or no reheat →	0 MBh

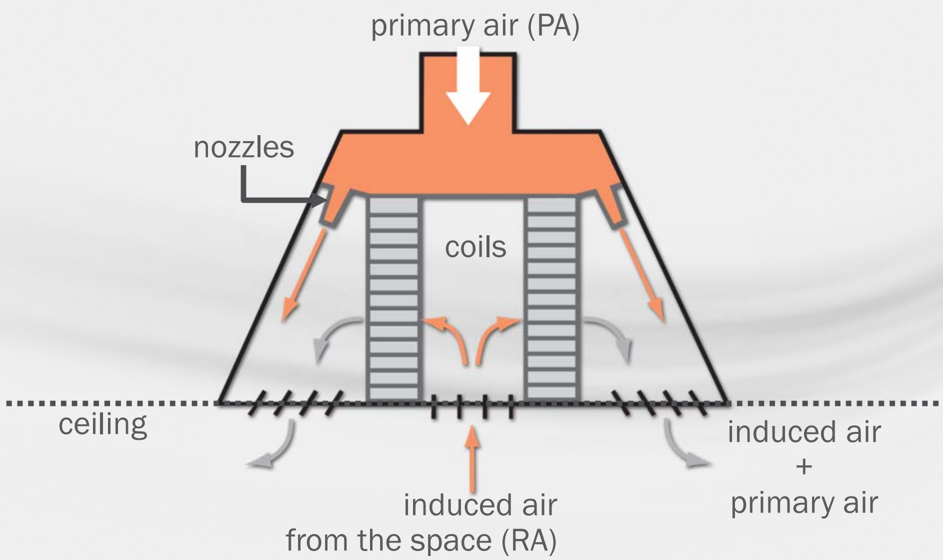
active **chilled beams**

Air System Requirements

The air system must:

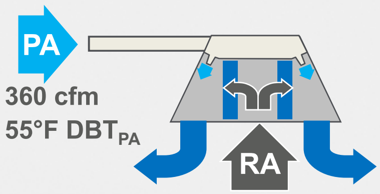
- Deliver the minimum outdoor airflow required by code to each zone (example: ASHRAE Standard 62.1)
- Deliver this air dry enough to offset the latent load in each zone and maintain indoor dew point at or below the desired limit (example: 55°F dew point)
- Deliver primary airflow (PA) needed to induce sufficient room air (RA) to offset space sensible cooling load

Active Chilled Beam



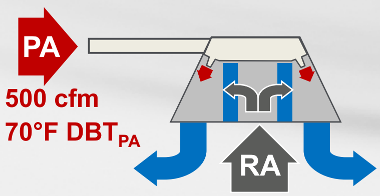
Example: Office Space

Minimum OA (ASHRAE 62.1) (to earn LEED credit)	**85 cfm** **(85 × 1.3 = 110 cfm)**	
Airflow required to offset space latent load (ex: 1000 Btu/hr)	**85 cfm** **110 cfm** **360 cfm**	**(DPT_{PA} = 47°F)** **(DPT_{PA} = 49°F)** **(DPT_{PA} = 53°F)**
Airflow needed to induce sufficient room air to offset space sensible cooling load (ex: 19,500 Btu/hr)	**360 cfm** **500 cfm**	**(DBT_{PA} = 55°F)** **(DBT_{PA} = 70°F)**



"Cold" (55°F) primary-air temperature

- primary air offsets 40% of sensible cooling load
- four (4) beams, each 6-ft long x 2-ft wide
- total primary airflow = 360 cfm
- total water flow = 6.0 gpm



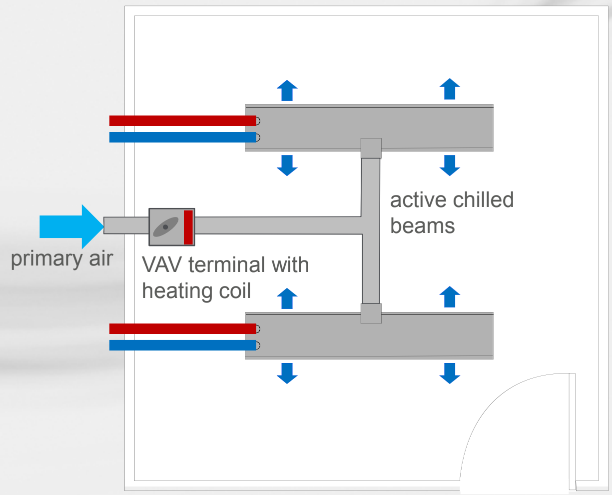
"Neutral" (70°F) primary-air temperature

- primary air offsets 14% of sensible cooling load
- six (6) beams, each 6-ft long x 2-ft wide
- total primary airflow = 500 cfm
- total water flow = 9.0 gpm

	"cold" primary air		"neutral" primary air
CFM_{PA}	360 cfm	larger ducts →	500 cfm
CFM_{OA}	85 cfm		85 cfm
DPT_{PA}	53°F		54°F
DBT_{PA}	55°F		70°F
DBT_{CC} (affects CHW temp)	55°F		56°F
AHU cooling coil	0.9 tons	more AHU tons →	1.5 tons
Terminal unit coil	1.0 tons	more beams →	1.4 tons
AHU supply fan	0.3 bhp	more fan power →	0.6 bhp
GPM_{AHU}	1.8 gpm	more pump power →	2.9 gpm
$GPM_{terminal}$	6.0 gpm		9.0 gpm

Cold PA: Preventing Overcooling

- Reset primary air dry-bulb temperature to avoid overcooling worst-case (coldest) zone
- Install a duct heating coil for each zone (or group of similar zones)



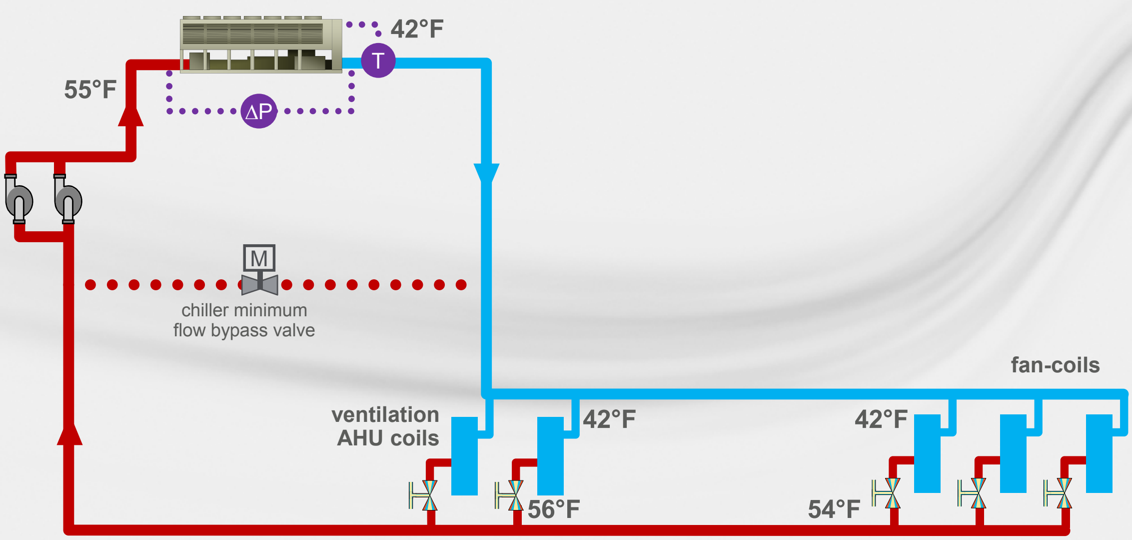
Agenda

- Types of chilled-water terminal units
  - Fan-coils / blower coils
  - Chilled beams
  - Radiant
- Dedicated OA system design
- **Chilled-water system configurations and control**

Chilled-Water System

- Single-chiller system
- Dual-temperature system
- Dual-temperature system with redundancy

Single-Chiller, Single-Temperature System

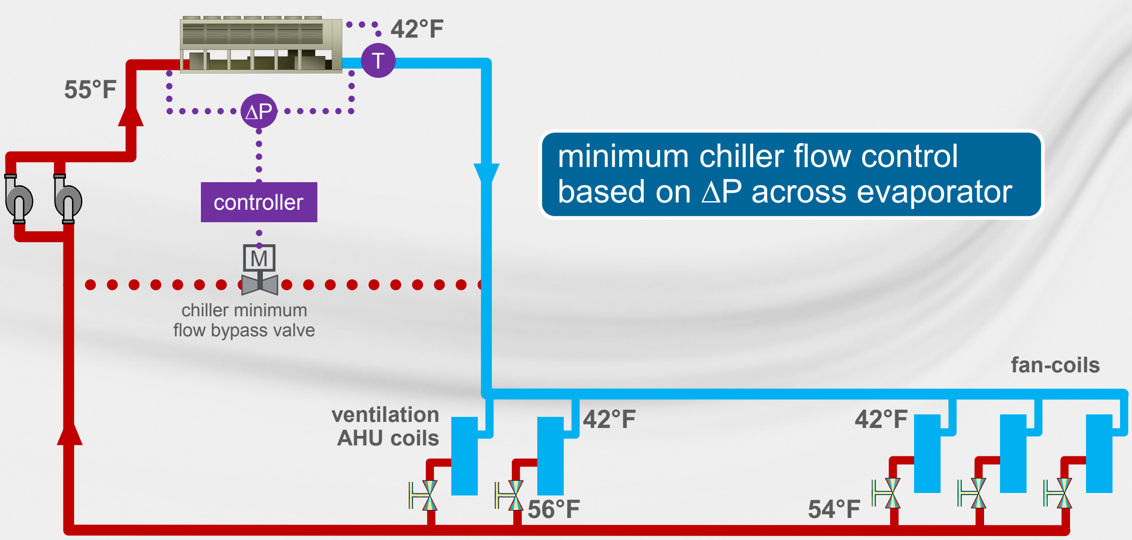


Single-Chiller System Example

- 180 tons
- 13°F ΔT (1.85 gpm/ton), 330 gpm
- 75 feet of head
- 70% pump efficiency
- Pump power
  = (330 gpm x 75 ft) / (3960 x 0.70)
  = 8.9 hp

- ASHRAE 90.1 requirements
- Variable flow not required since power is below 10 hp
- But customer was "sold" on variable flow, so variable primary flow (VPF) is used

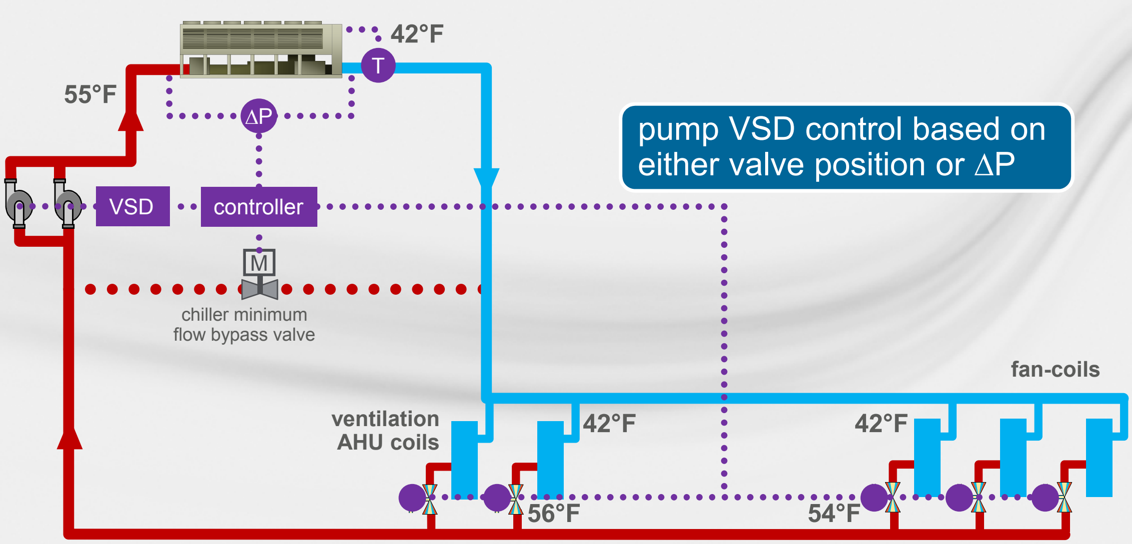
Single-Chiller, Single-Temperature System

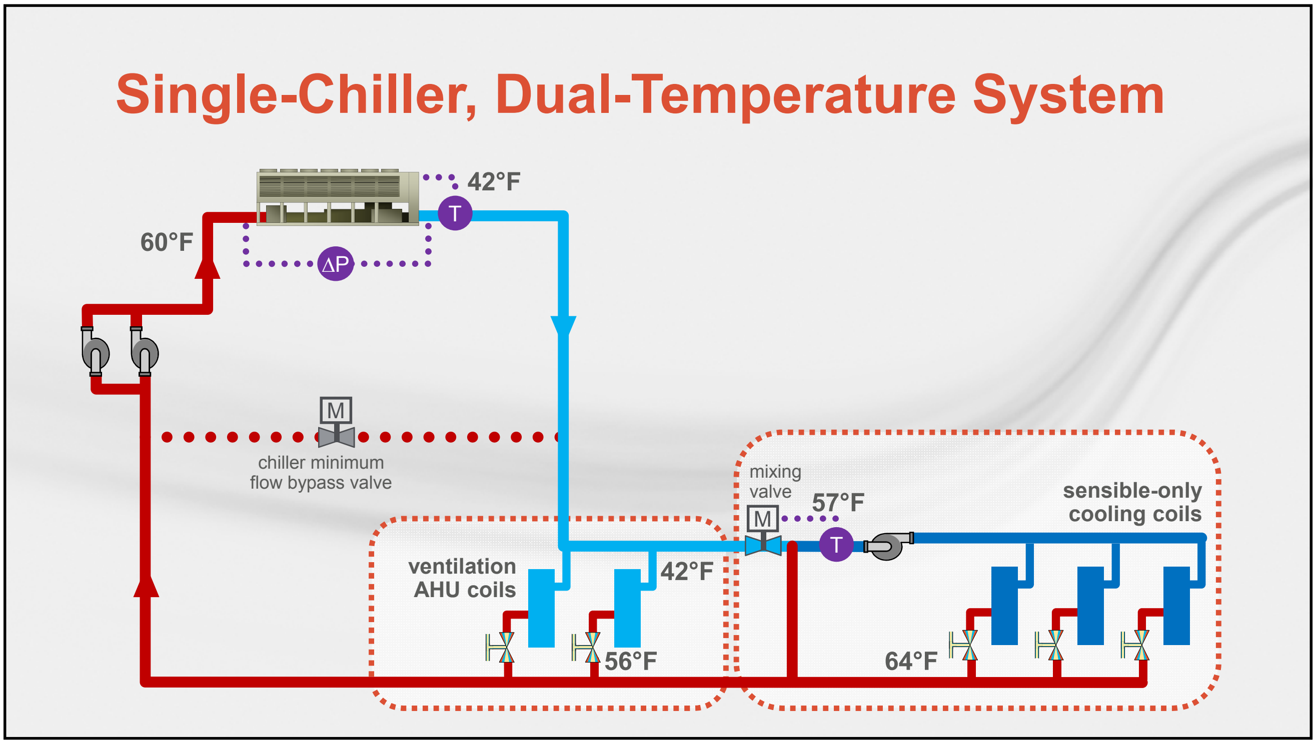


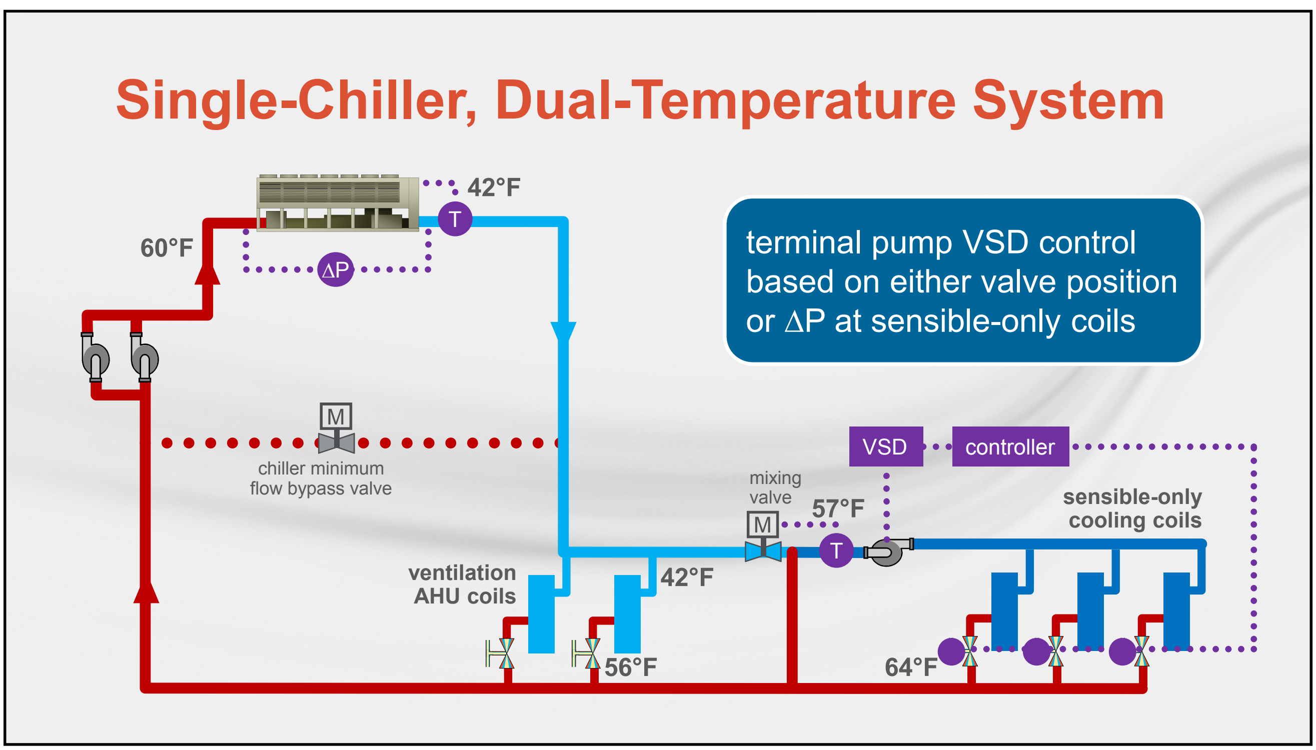
Chilled-Water Pump Control

- Modulating valves on terminal units
  - May use valve position ("pump-pressure optimization")
- Two-position valves on terminal units
  - Use a ΔP sensor at "most remote" coil

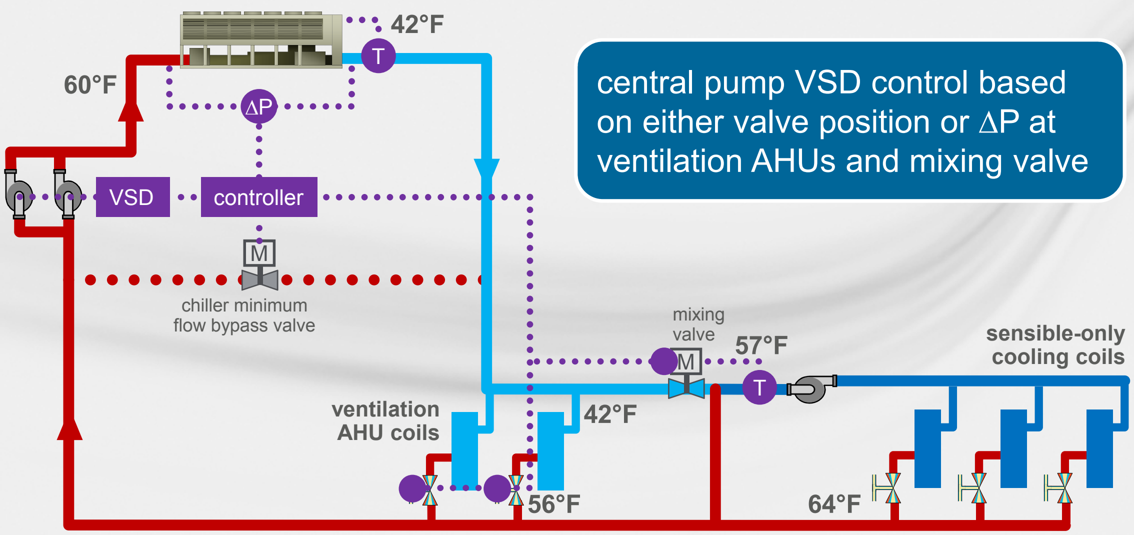
Single-Chiller, Single-Temperature System




Single-Chiller, Dual-Temperature System
42°F
60°F
T
ΔP
M
chiller minimum
flow bypass valve
mixing
valve
57°F
sensible-only
cooling coils
M
T
ventilation
AHU coils
42°F
56°F
64°F


Single-Chiller, Dual-Temperature System
42°F
60°F
T
ΔP
terminal pump VSD control
based on either valve position
or ΔP at sensible-only coils
M
chiller minimum
flow bypass valve
VSD
controller
mixing
valve
57°F
sensible-only
cooling coils
M
T
ventilation
AHU coils
42°F
56°F
64°F

Single-Chiller, Dual-Temperature System

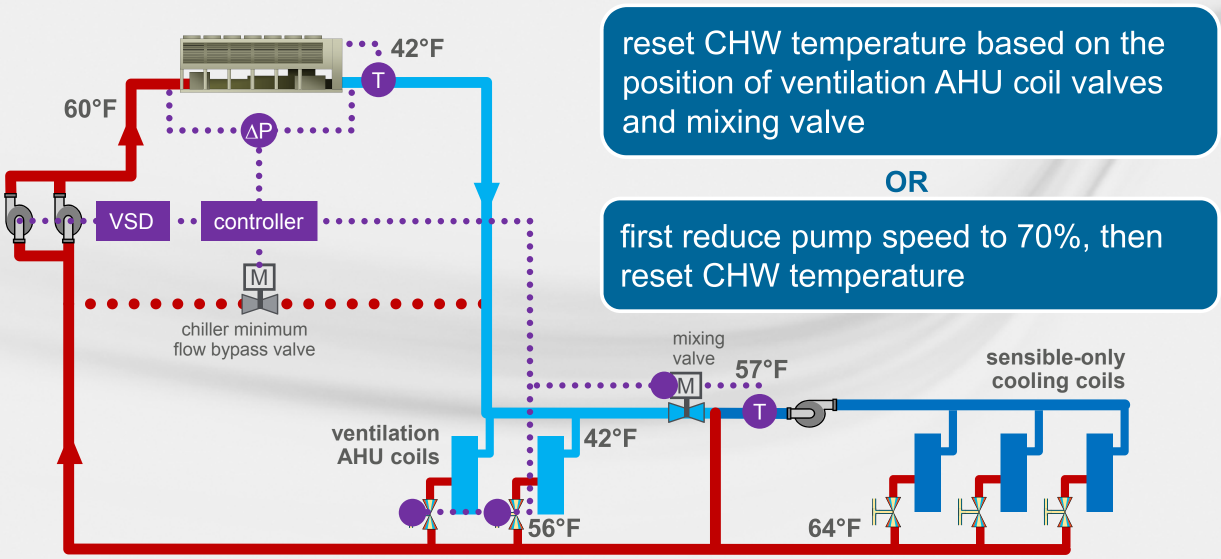


Example Chiller Selections

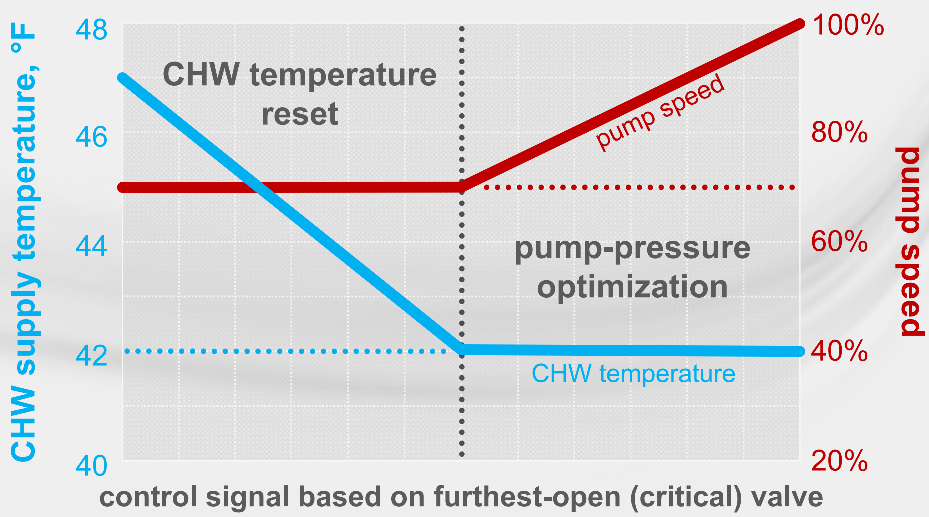
	Design					Minimum	
Number of Passes	Capacity (tons)	Full Load EER	NPLV (EER)	Flow Rate (gpm)	ΔP (ft. H20)	Flow Rate (gpm)	ΔP (ft. H20)
2	193	9.6	13.2	256	3.8	241	3.4
3	197	9.7	13.4	262	13.5	161	5.4

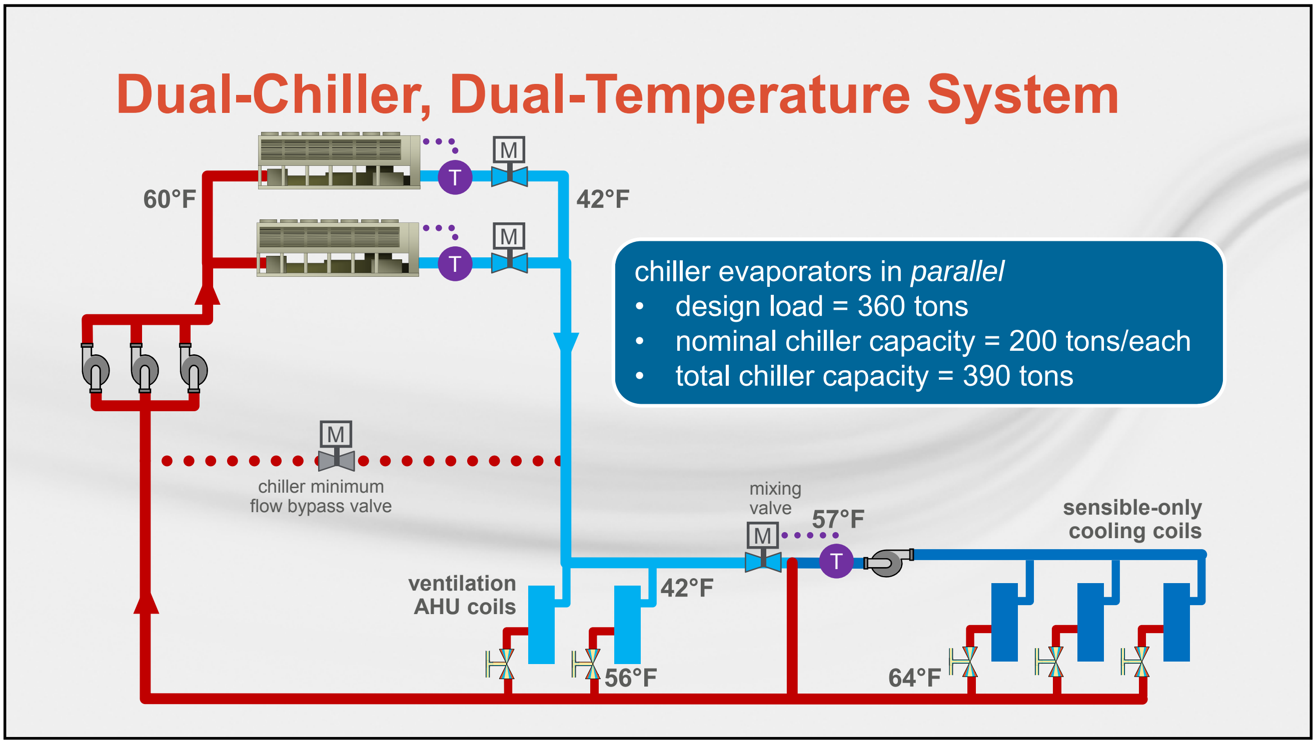
Flow rate cannot be reduced much for the two-pass evaporator

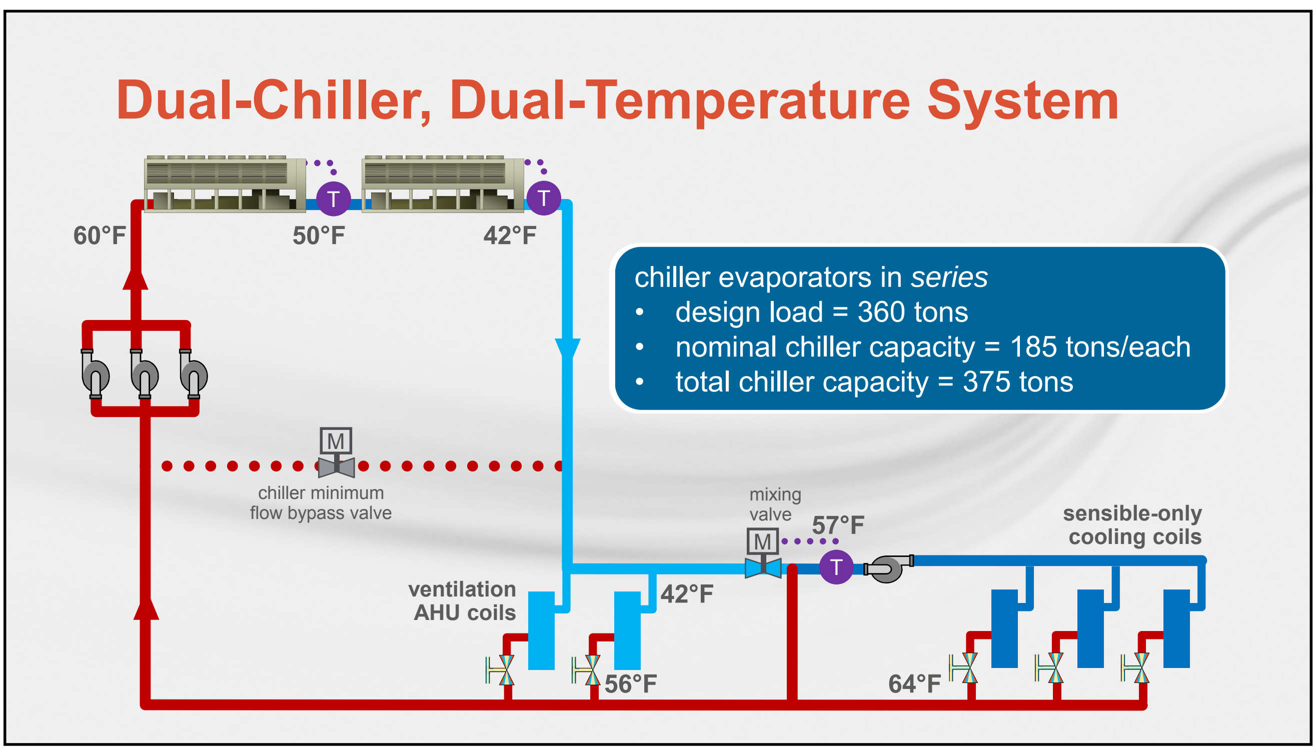
Pump-Pressure Optimization or Chilled-Water Reset?

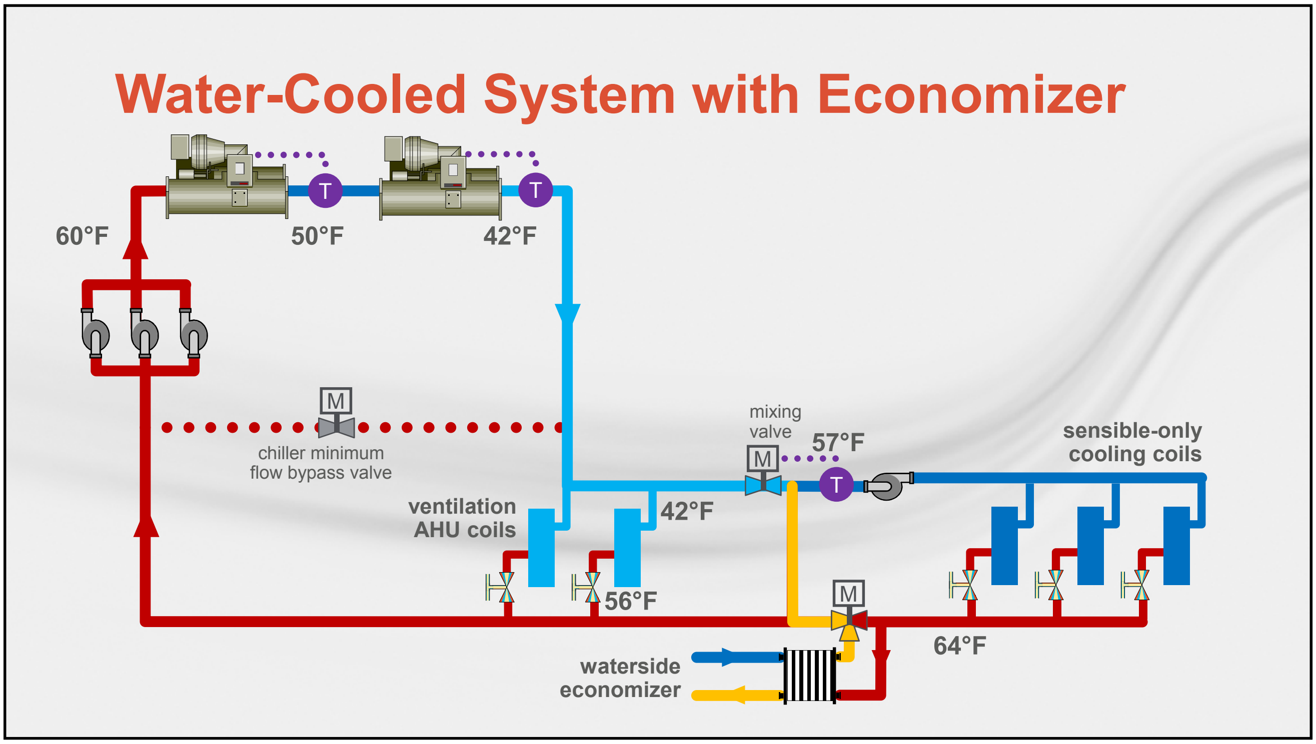


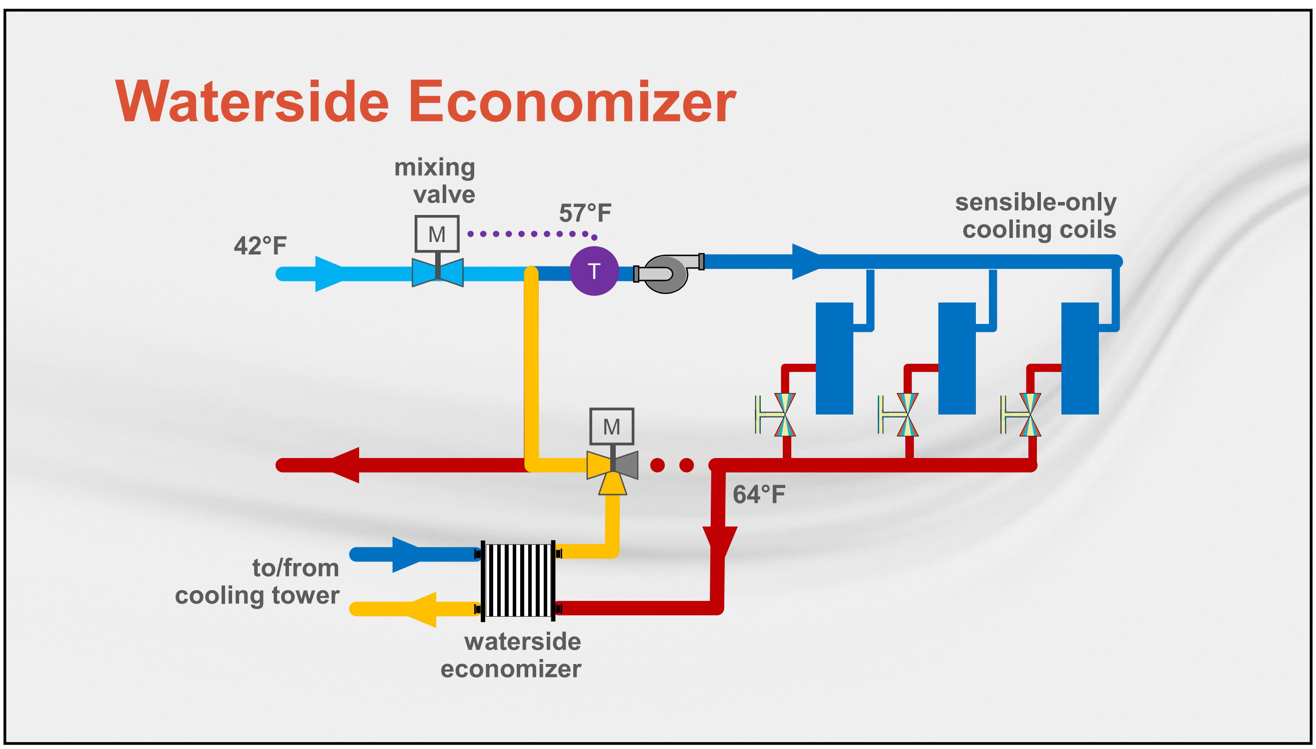
Example of Hybrid Control



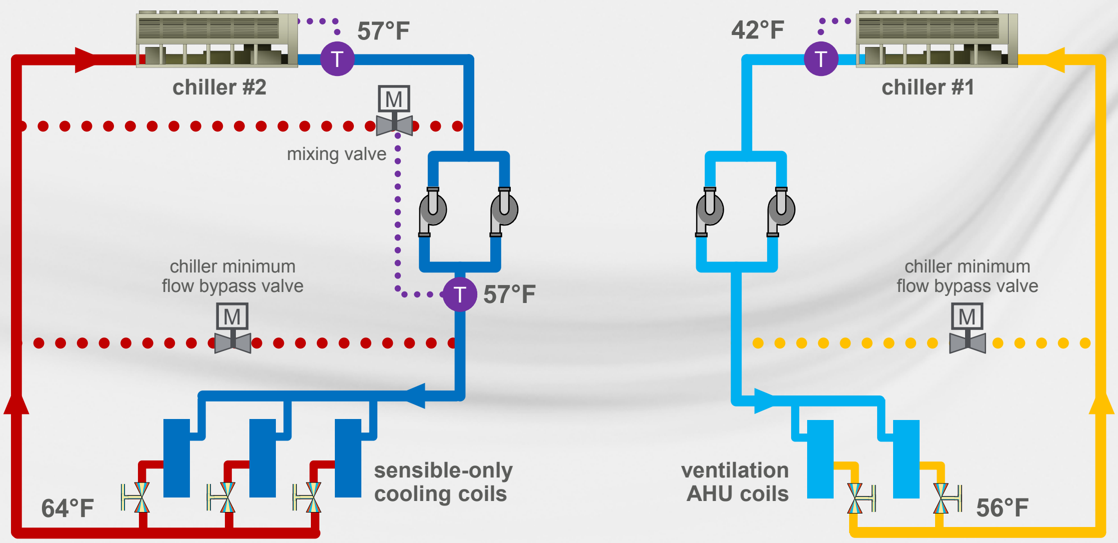

Dual-Chiller, Dual-Temperature System
60°F
42°F
chiller evaporators in *parallel*
• design load = 360 tons
• nominal chiller capacity = 200 tons/each
• total chiller capacity = 390 tons
M
T
chiller minimum
flow bypass valve
mixing
valve
57°F
sensible-only
cooling coils
ventilation
AHU coils
42°F
56°F
64°F


Dual-Chiller, Dual-Temperature System
60°F
50°F
42°F
chiller evaporators in *series*
• design load = 360 tons
• nominal chiller capacity = 185 tons/each
• total chiller capacity = 375 tons
M
T
chiller minimum
flow bypass valve
mixing
valve
57°F
sensible-only
cooling coils
ventilation
AHU coils
42°F
56°F
64°F


Water-Cooled System with Economizer
60°F
50°F
42°F
chiller minimum
flow bypass valve
mixing
valve
57°F
sensible-only
cooling coils
ventilation
AHU coils
42°F
56°F
64°F
waterside
economizer

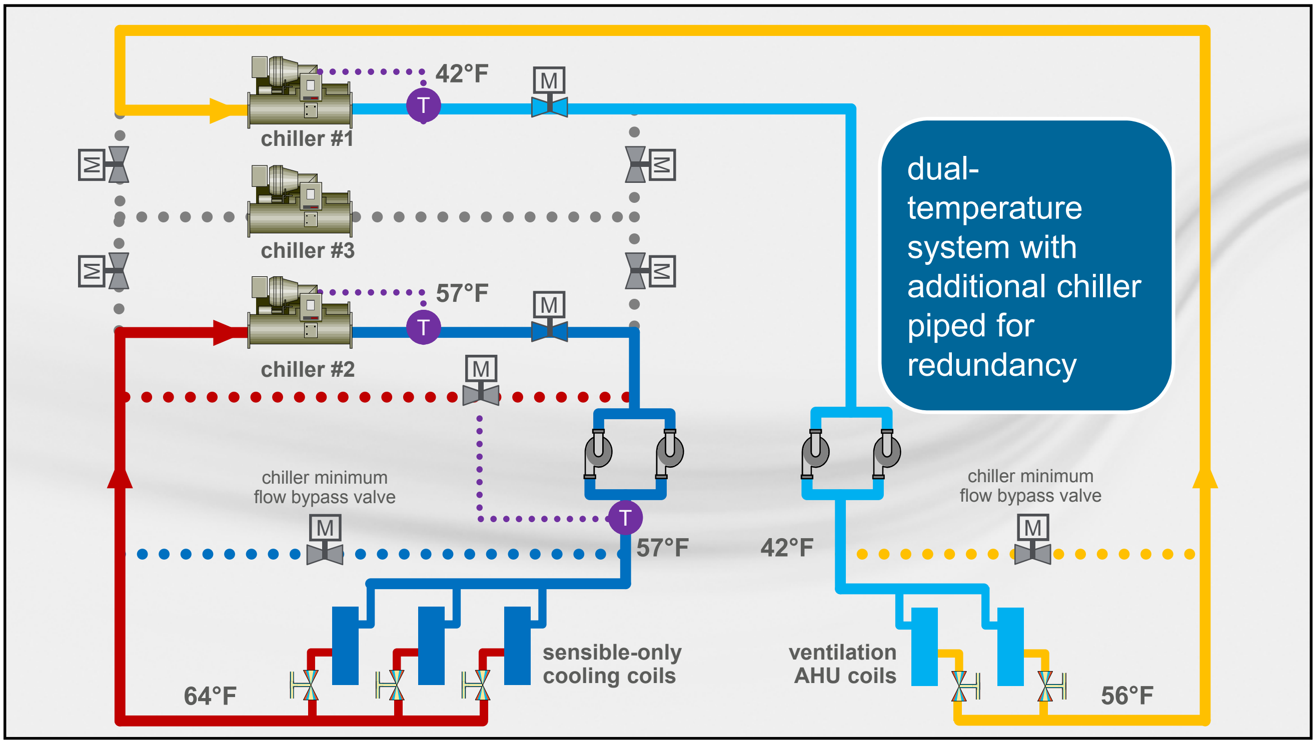

Waterside Economizer
mixing
valve
57°F
42°F
sensible-only
cooling coils
64°F
to/from
cooling tower
waterside
economizer

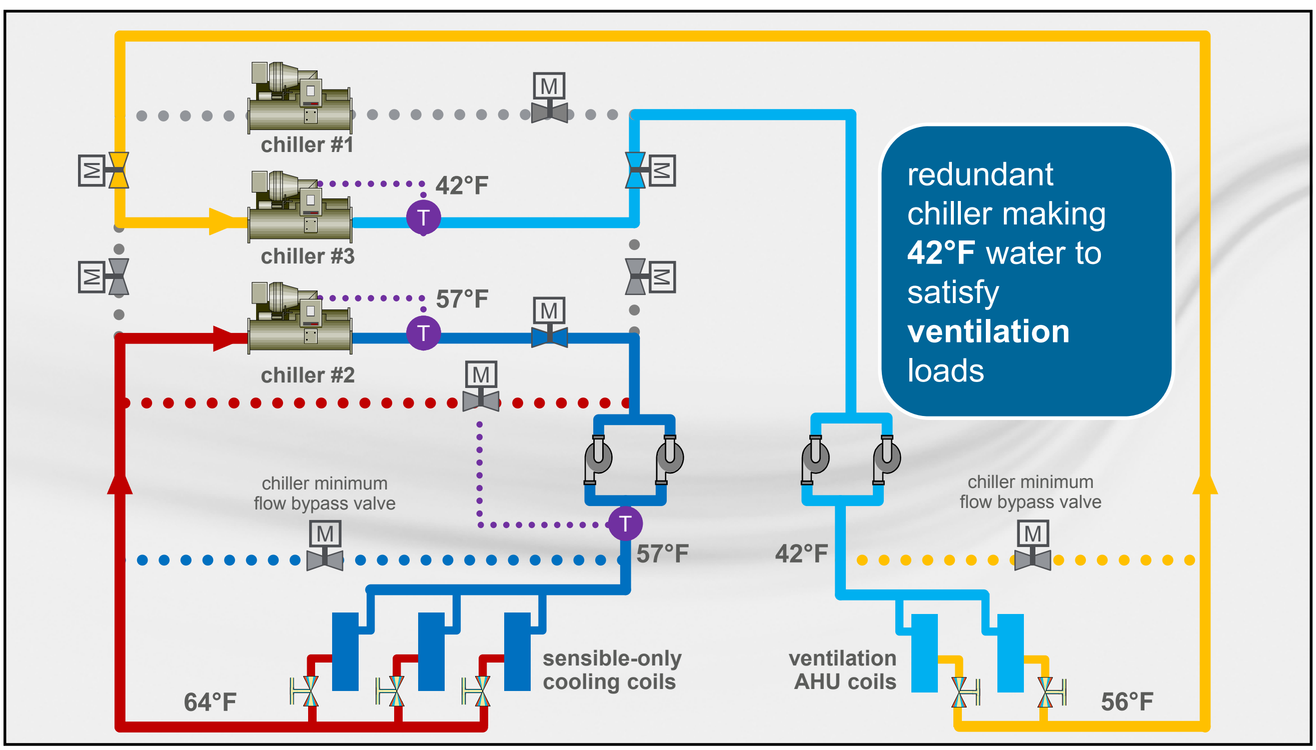
Dual-Temperature System

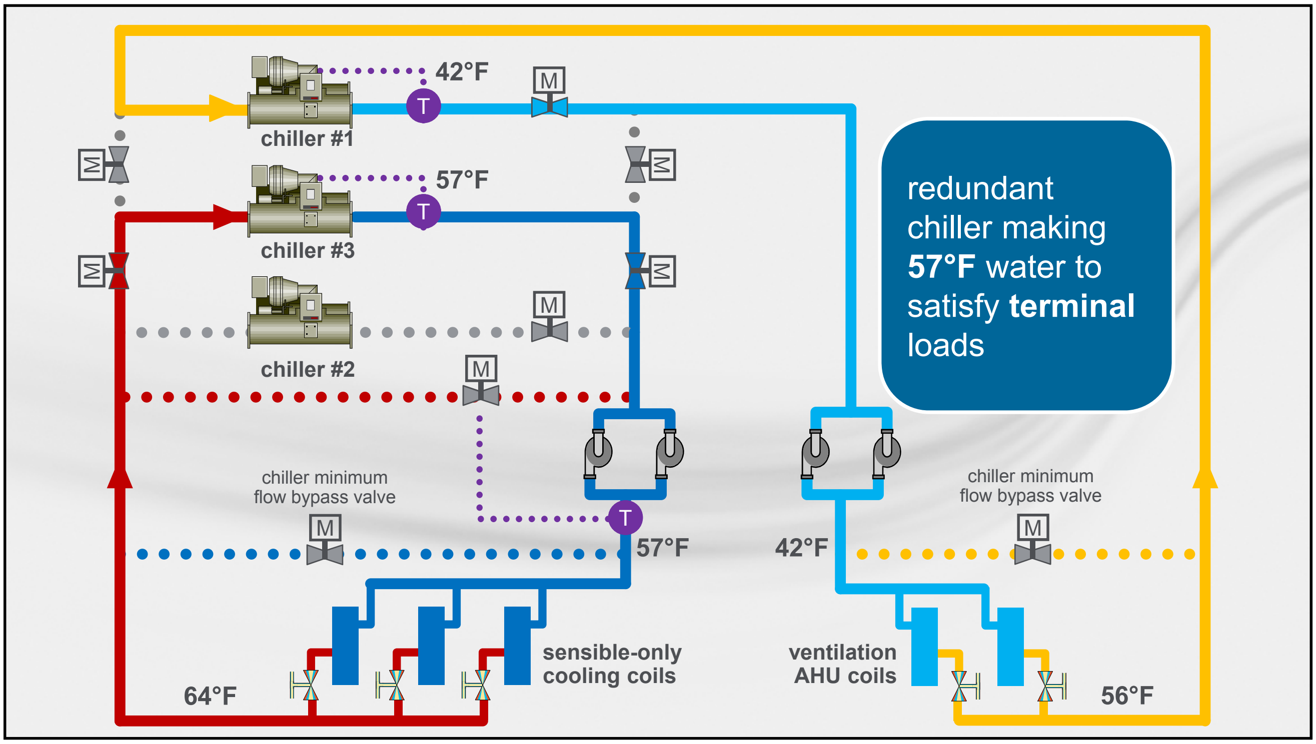


Example Chiller Selections

Supply Water Temperature (°F)	Nominal Size (tons)	Number of Passes	Capacity (tons)	Full Load EER	NPLV (EER)	Flow Rate (gpm)	ΔP (ft. H20)
42	200	3	197	9.7	13.4	308	13.5
57	155	2	189	10.9	16.6	647	31.1


42°F
M
chiller #1
M
M
chiller #3
M
M
57°F
M
chiller #2
M
dual-temperature system with additional chiller piped for redundancy
chiller minimum flow bypass valve
M
T
57°F
42°F
chiller minimum flow bypass valve
M
sensible-only cooling coils
ventilation AHU coils
64°F
56°F


M
chiller #1
M
M
42°F
chiller #3
M
M
57°F
M
chiller #2
M
redundant chiller making **42°F** water to satisfy **ventilation** loads
chiller minimum flow bypass valve
M
T
57°F
42°F
chiller minimum flow bypass valve
M
sensible-only cooling coils
ventilation AHU coils
64°F
56°F



Chilled-Water Systems

- Single-temperature system can be used for fan-coils
- Dual-temperature systems are applicable for terminal units providing sensible cooling only
- In two-chiller systems, configuring the chillers in series offers installed and operating cost benefits
- In a dual-temperature system, one additional chiller can provide redundancy, if piped properly

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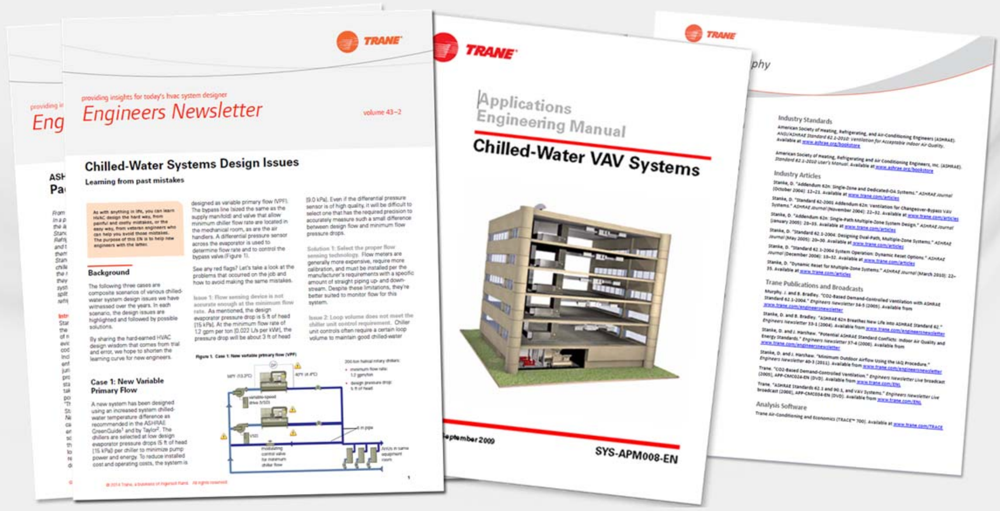

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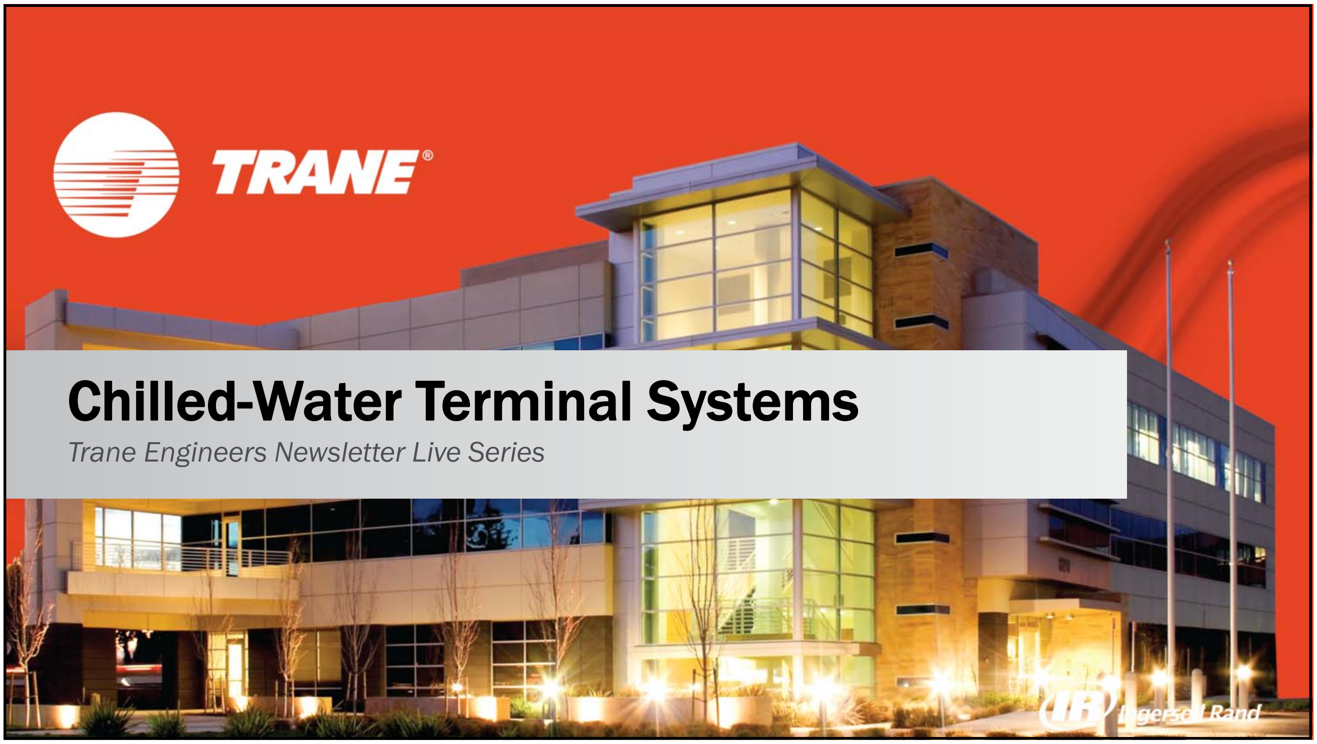
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