Trane Engineers Newsletter Live

HVAC Myths and Realities

Presenters: Systems and Applications Engineers Lee Cline, Dustin Meredith and Mick Schwedler with Jeanne Harshaw (host)

Trane program number: APP-CMC062-EN

Agenda

Trane *Engineers Newsletter Live* Series

HVAC Myths and Realities

Abstract

This program addresses various "myths," claims, and misunderstandings in the HVAC & R market place. Topics will include energy efficiency claims, system performance, acoustics, technologies, and others. Each myth will be explored with respect to why it "seems correct on the surface." This will be followed by technically correct details, examples and situations so building owners, operators and project teams can evaluate the likelihood of actually realizing claimed effectiveness, performance and savings.

Presenters: Trane applications engineers Lee Cline, Dustin Meredith, Mick Schwedler and Jeanne Harshaw (host)

After viewing attendees will be able to:

1. Apply several solutions to avoid low delta T.
2. Summarize the impact pressure changes have on fan curves and airflow.
3. Understand that to maintain comfortable humidity levels, discharge air condition and its impact on the space must be considered along with discharge air temperature.
4. Explain how ASHRAE Standards 15 and 34 differ and how they work together.

Agenda

- Myth 1: Low delta T Is unavoidable
- Myth 2: 55°supply air temperature is adequate for today's load
- Myth 3: ASHRAE Standard 15 has to be updated before the new refrigerants can be used
- Myth 4: Single-zone VAV units don't need hot gas reheat
- Myth 5: VFDs and affinity laws
- Myth 6: Small changes in pressure can have a huge impact on airflow for flat fan curves and may cause the system to surge
- Myth 7: New chilled-water systems need to be variable-primary flow
- Myth 8: System airflow issues are the fans fault
- Myth 9: Claims to energy savings

Bonus Features

- Myth 10: Anti-freeze doesn't have much affect on chilled water systems
- Myth 11: If refrigerant volume is too high for an occupied space to satisfy ASHRAE Standard 15 requirements, putting a refrigerant monitor in that occupied space meets the Standard 15 requirements

Presenter biographies

HVAC Myths and Realities

Lee Cline | systems engineer | Trane

Lee is a staff engineer in the Systems Engineering department with over 36 years of experience at Trane. His career at Trane started as a factory service engineer for heavy refrigeration, helping to introduce the CVHE centrifugal chiller with the first generation of electronic controls to the industry. Lee went on to join the team that kicked off the microelectronic building automation and Integrated Comfort Systems (ICS) controls offering at Trane.
In his current role, he continues to push new unit and system control and optimization concepts into the industry, many of which are integrated in Trane EarthWise™ Systems. As a Systems Engineer Lee also has the opportunity to discuss HVAC system application and control with owners, engineers and contractors on a daily basis.

Lee earned his Bachelors degree in Mechanical Engineering from Michigan Technological University. He is a member of ASHRAE and a Registered Professional Engineer in the State of Wisconsin.

Dustin Meredith | applications engineer | Trane

Dustin joined Trane in 2000 as a marketing engineer. In his current role as an applications engineer, he specializes in airside products and systems. His expertise includes sound & vibration analysis, fan application, and air system design. He holds multiple patents and has been instrumental in advancing cutting-edge direct-drive fan and motor applications to industry. Dustin authors technical engineering bulletins, presents technical seminars, and analyzes systems for optimum performance.

Dustin is a registered professional engineer and earned his mechanical engineering, computer science, and MBA degrees from the University of Kentucky. He is an ASHRAE Section Head and former Chair of ASHRAE Technical Committee TC 2.6—Sound & Vibration Control. He is a corresponding member of ASHRAE Technical Committee 5.1—Fans—and is Trane's voting representative for the Air Movement and Control Association.

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.

Mick is an ASHRAE Fellow and member of the Board of Directors. He is a recipient of ASHRAE's Exceptional Service, Distinguished Service and Standards Achievement Awards. He is past Chair of SSPC 90.1 and contributed to the ASHRAE GreenGuide. He is also active with the U.S. Green Building Council, having served on technical and education committees and is currently the LEED Technical Committee Chair. Mick earned his BSME degree from Northwestern University and his MSME from the University of Wisconsin Solar Energy Lab.

HVAC Myths and Realities

Trane Engineers Newsletter Live Series

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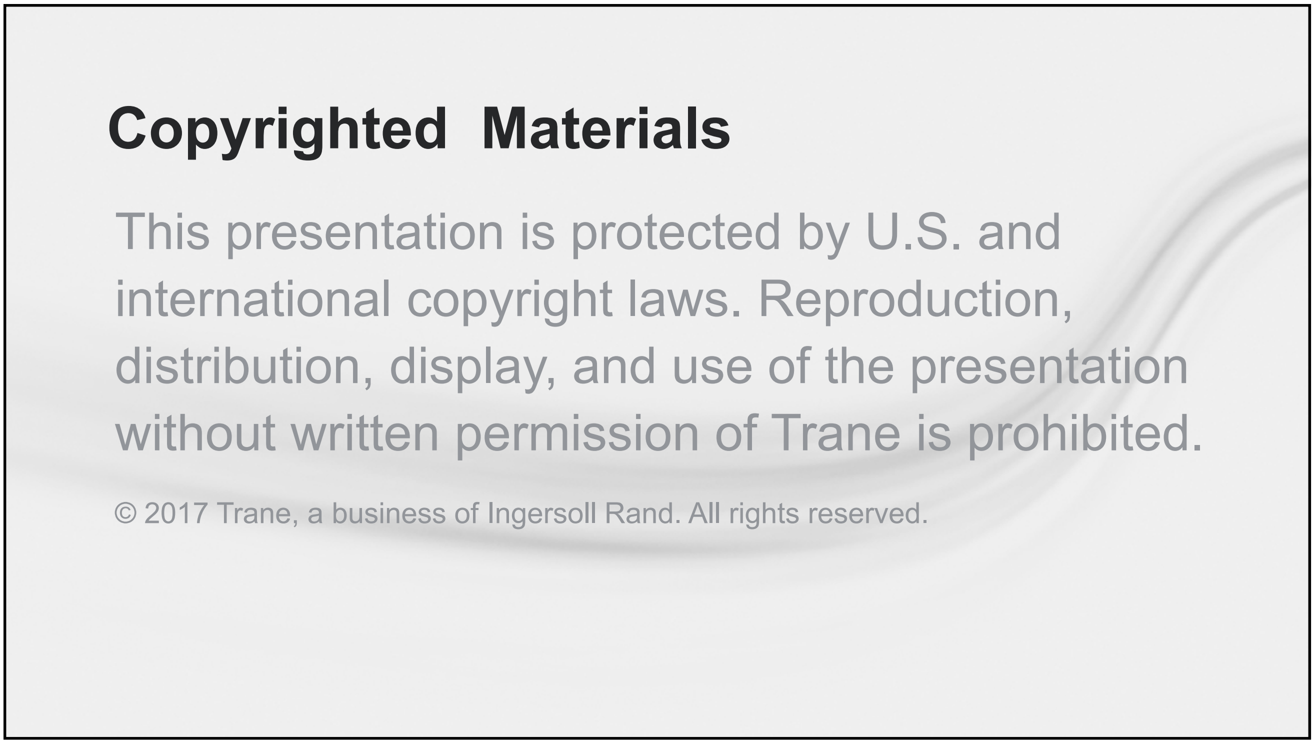

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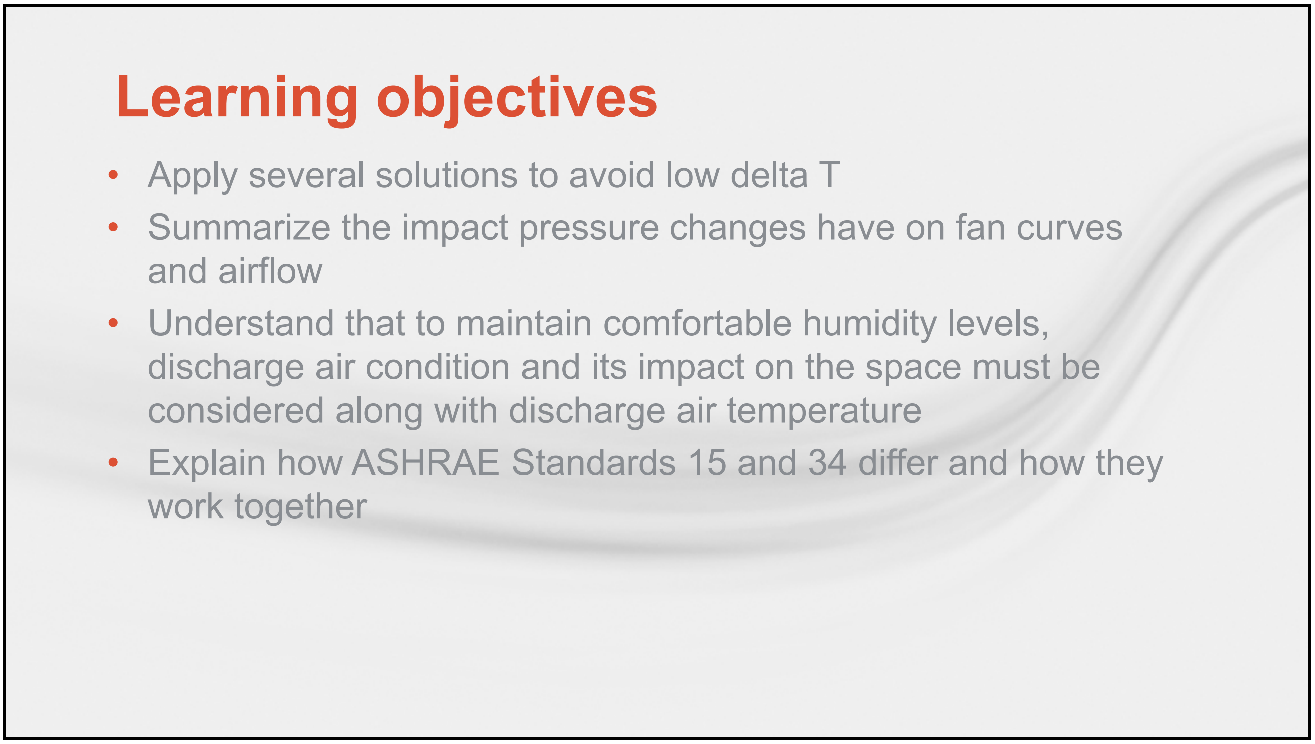
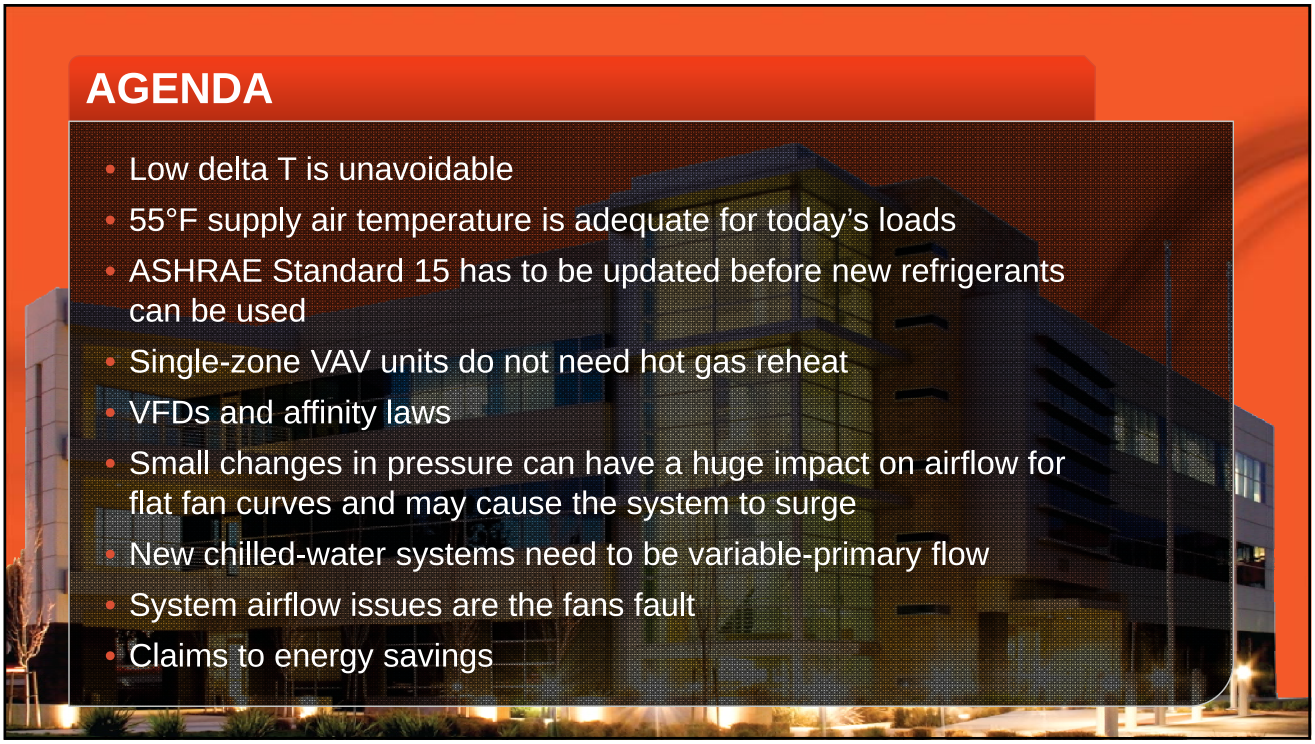


Learning objectives

- Apply several solutions to avoid low delta T
- Summarize the impact pressure changes have on fan curves and airflow
- Understand that to maintain comfortable humidity levels, discharge air condition and its impact on the space must be considered along with discharge air temperature
- Explain how ASHRAE Standards 15 and 34 differ and how they work together



AGENDA

- Low delta T is unavoidable
- 55°F supply air temperature is adequate for today's loads
- ASHRAE Standard 15 has to be updated before new refrigerants can be used
- Single-zone VAV units do not need hot gas reheat
- VFDs and affinity laws
- Small changes in pressure can have a huge impact on airflow for flat fan curves and may cause the system to surge
- New chilled-water systems need to be variable-primary flow
- System airflow issues are the fans fault
- Claims to energy savings

Today's Presenters

Dustin Meredith
Applications Engineer

Lee Cline
Applications Engineer

Mick Schwedler
Manager, Applications Engineer

Myth Number 1

Low delta T is unavoidable.

Transport Energy

is low delta T unavoidable?

- Tons= $\frac{(\Delta T \times GPM)}{24}$

Solving for gpm…

- GPM= $\frac{(Tons \times 24)}{\Delta T}$

Pumping power…

- Frictional Head $\propto$ Flow^2
- Water HP (bhp)= $\frac{(GPM \times \text{head (ft)})}{3960}$
- Water HP $\propto$ Flow ^{3} $\propto$ Delta T ^{3}

Coil Delta T

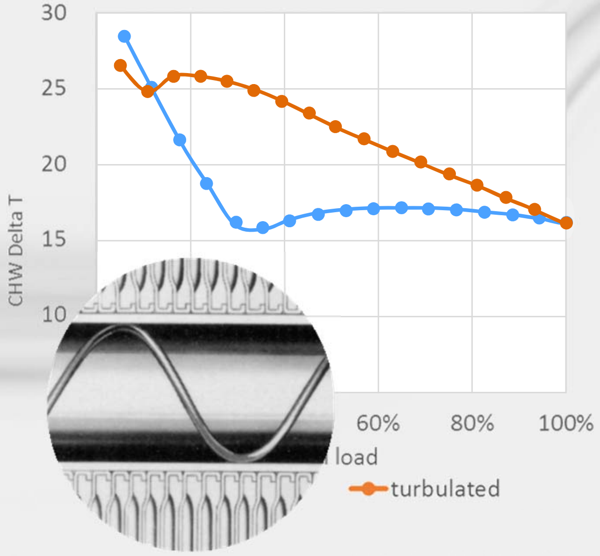
is low delta T unavoidable?

- AHRI Certified Coil
- Air Flow (VAV) unloading

ASHRAE 90.1-2016

6.5.4.7 Chilled-Water Coil Selection
Chilled-water cooling coils shall be selected to provide a 15°F or higher temperature difference between leaving and entering water temperatures and a minimum of 57°F leaving water temperature at *design conditions*.

2015 Engineer's Newsletter Live
Coil Selection and Optimization

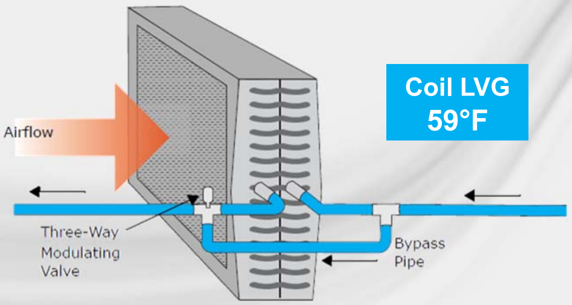


Reason 1: 3-Way Control Valves

undesirable mixing in variable flow systems

1. 3-way control valves
    - ***Eliminate them!***

50% Coil Load



Coil LVG 59°F

Coil Delta T = 17°F

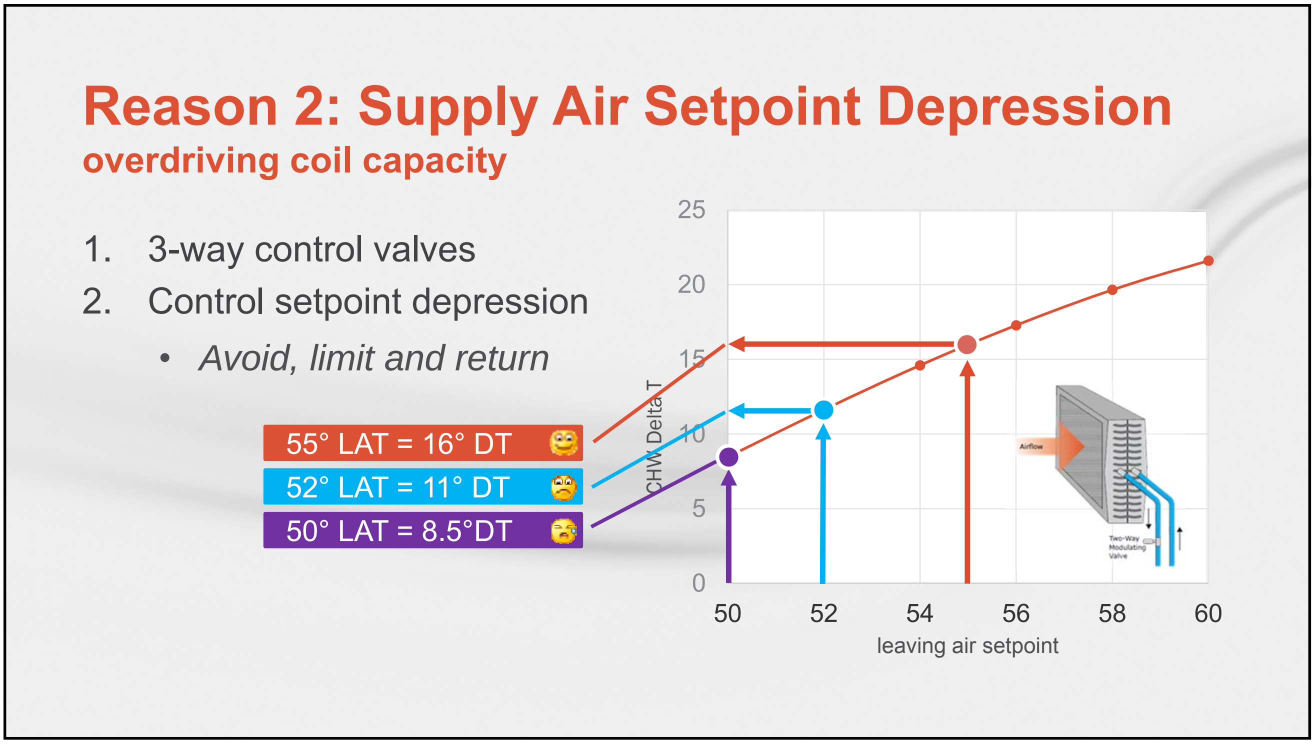
System Delta T = 8.5°F

CHWR 50.5°F

Bypass 42°F

CHWS 42°F

CHWR = [(42° x 50) + (59° x 50)] / 100 = 50.5°



Reason 2: Supply Air Setpoint Depression

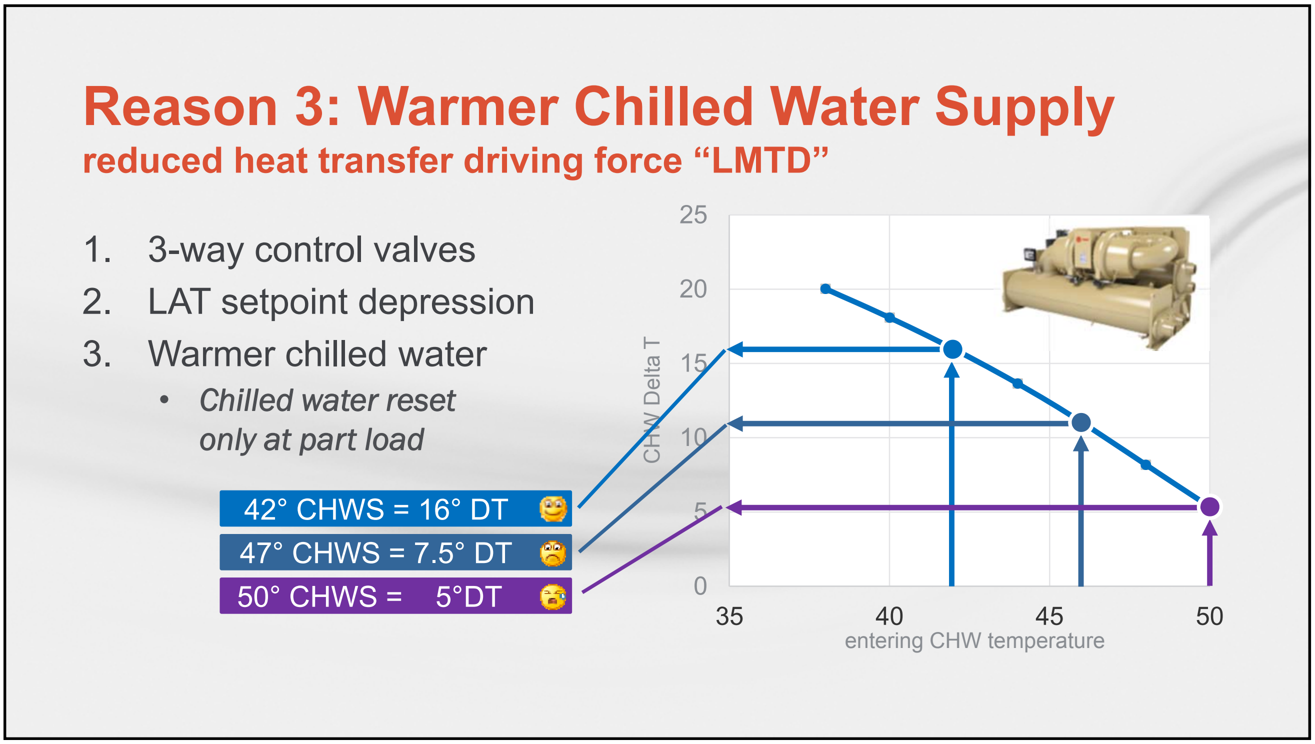
overdriving coil capacity

1. 3-way control valves
2. Control setpoint depression
    - *Avoid, limit and return*

55° LAT = 16° DT 😊

52° LAT = 11° DT 😟

50° LAT = 8.5°DT 😢



Reason 3: Warmer Chilled Water Supply

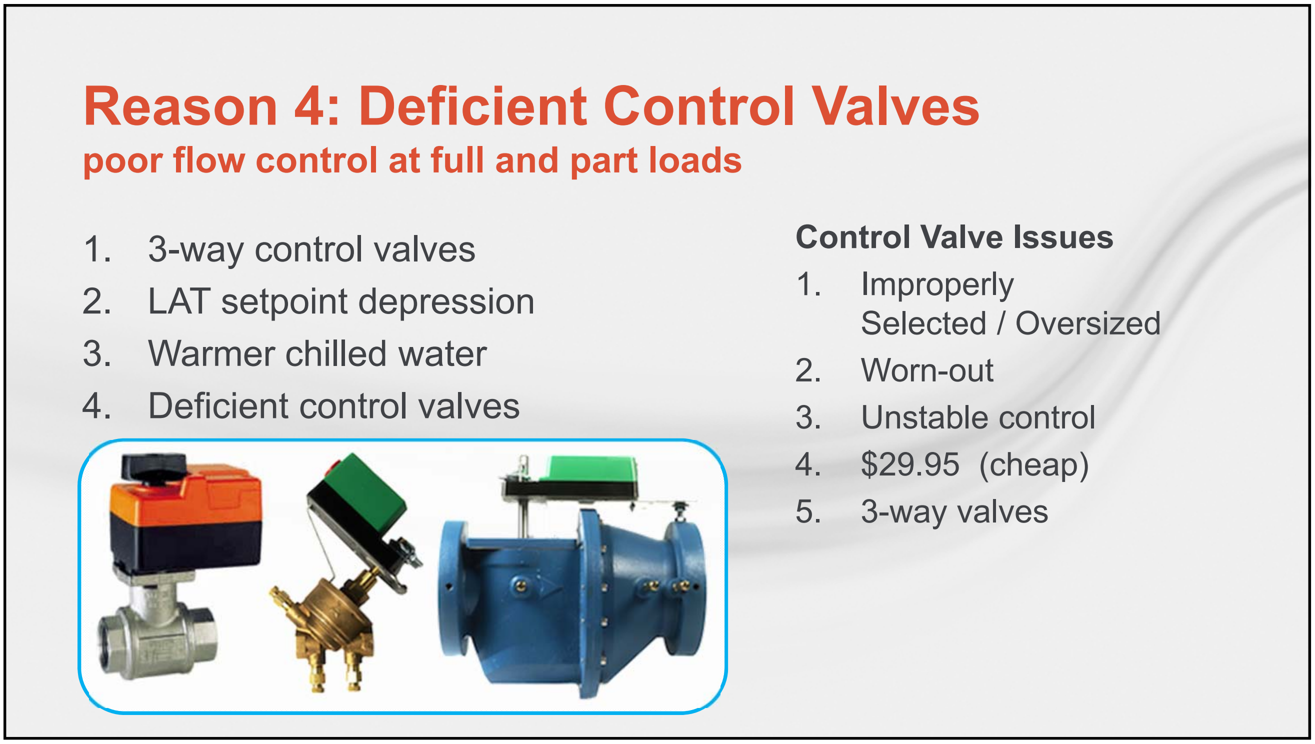
reduced heat transfer driving force “LMTD”

1. 3-way control valves
2. LAT setpoint depression
3. Warmer chilled water
    - *Chilled water reset only at part load*

42° CHWS = 16° DT 😊

47° CHWS = 7.5° DT 😟

50° CHWS = 5°DT 😢



Reason 4: Deficient Control Valves

poor flow control at full and part loads

1. 3-way control valves
2. LAT setpoint depression
3. Warmer chilled water
4. Deficient control valves

Control Valve Issues

1. Improperly Selected / Oversized
2. Worn-out
3. Unstable control
4. $29.95 (cheap)
5. 3-way valves

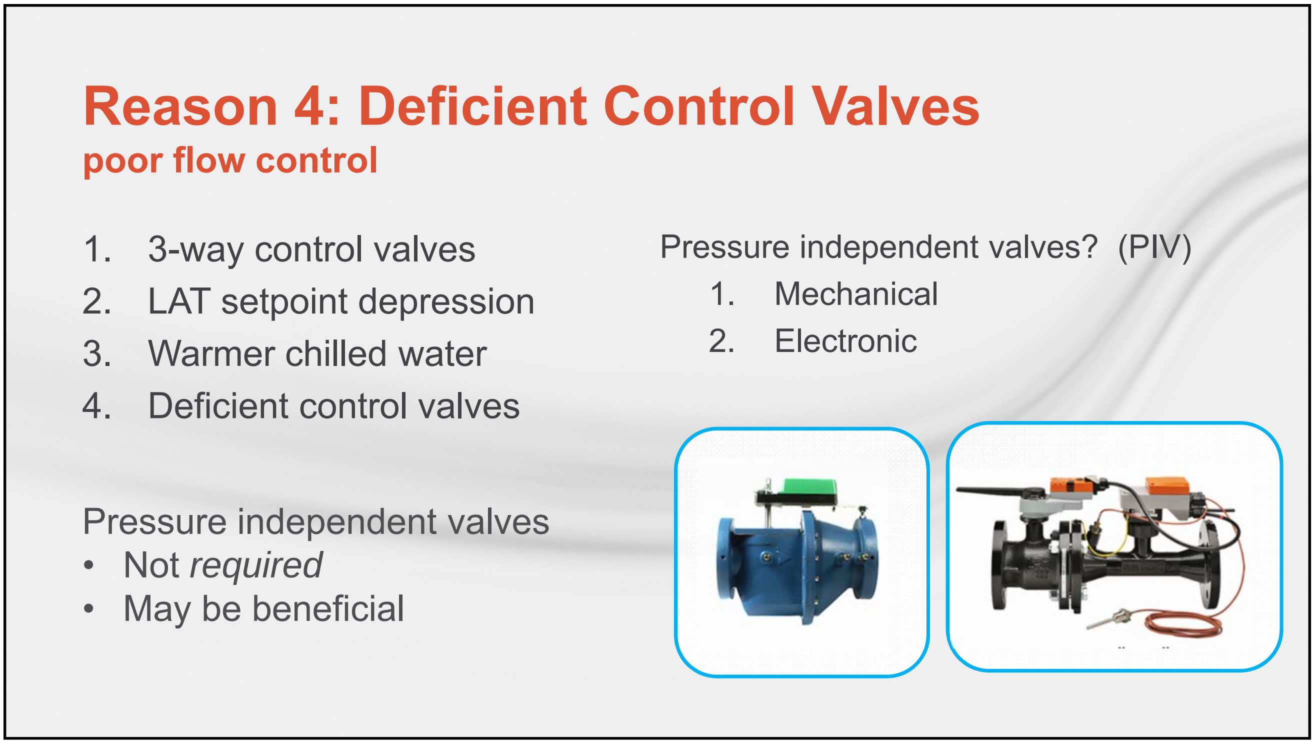
Reason 4: Deficient Control Valves

poor flow control

1. 3-way control valves
2. LAT setpoint depression
3. Warmer chilled water
4. Deficient control valves
   - *Specify quality valves specific to use*

8th floor control point 20 ft pd

2nd floor pressure 90 ft pd



Reason 4: Deficient Control Valves

poor flow control

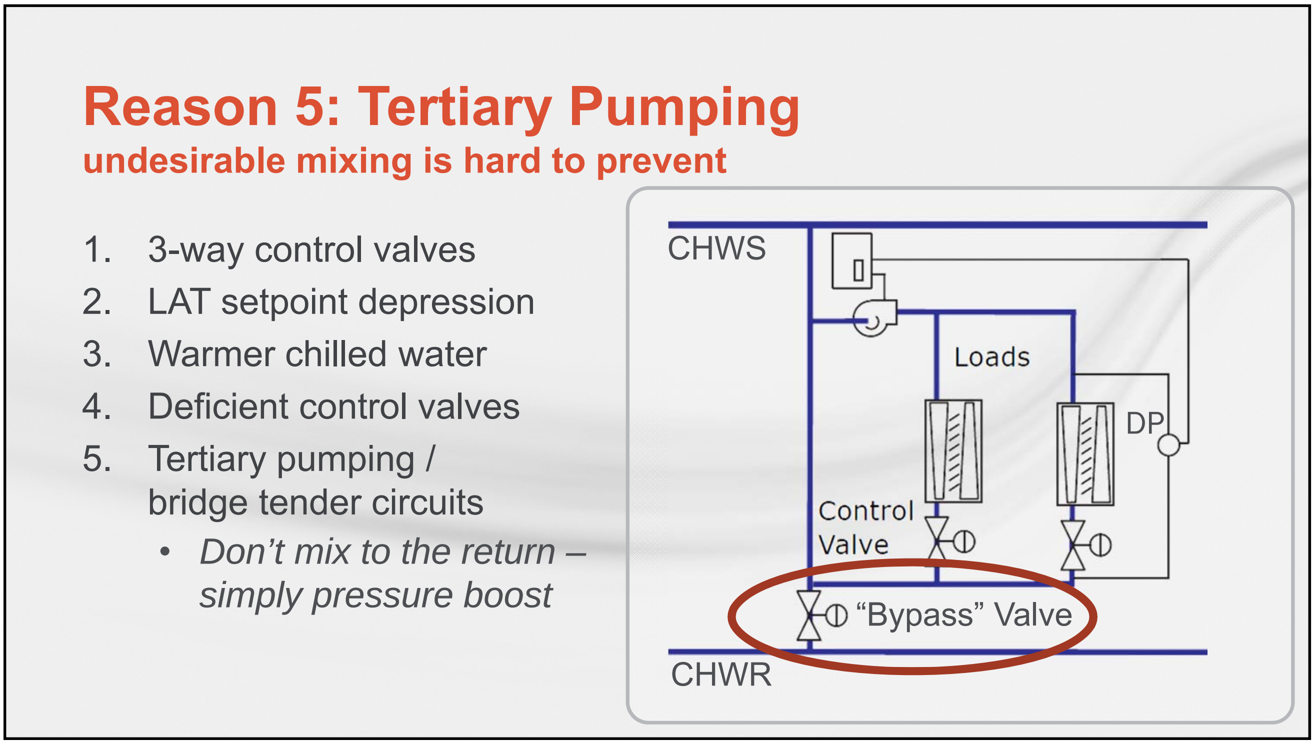
1. 3-way control valves
2. LAT setpoint depression
3. Warmer chilled water
4. Deficient control valves

Pressure independent valves? (PIV)

1. Mechanical
2. Electronic

Pressure independent valves

- Not *required*
- May be beneficial



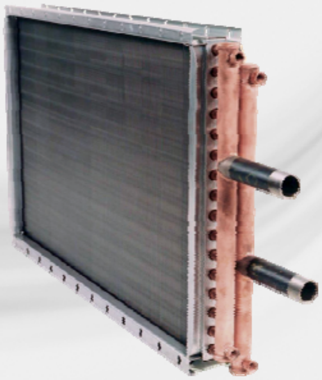
Reason 5: Tertiary Pumping

undesirable mixing is hard to prevent

1. 3-way control valves
2. LAT setpoint depression
3. Warmer chilled water
4. Deficient control valves
5. Tertiary pumping / bridge tender circuits
    - *Don't mix to the return – simply pressure boost*

Design Delta T and Greater is Achievable

1. AHRI certified coil selections
2. AHU set point limits
3. Chilled water reset only at part load
4. Properly selected / high quality valves
5. Pressure boosting – no tertiary “mixing”



Myth Number

Low Delta T s u a le.

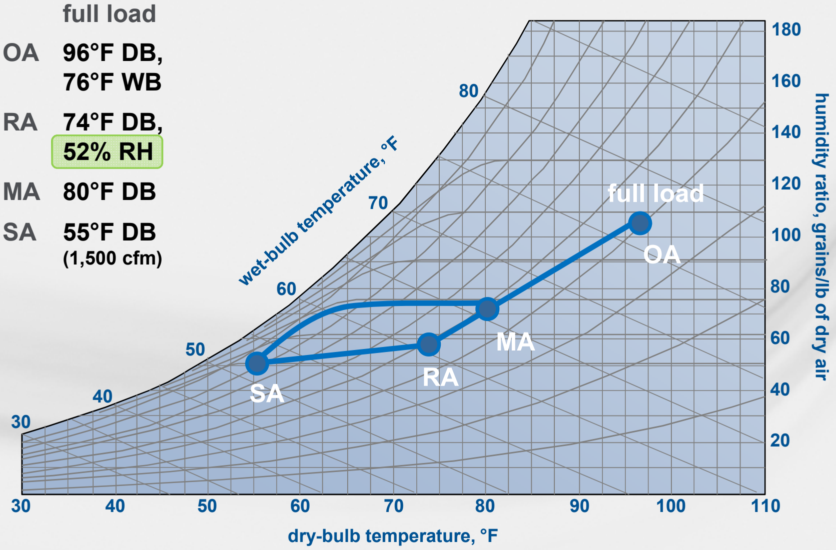
BUSTED

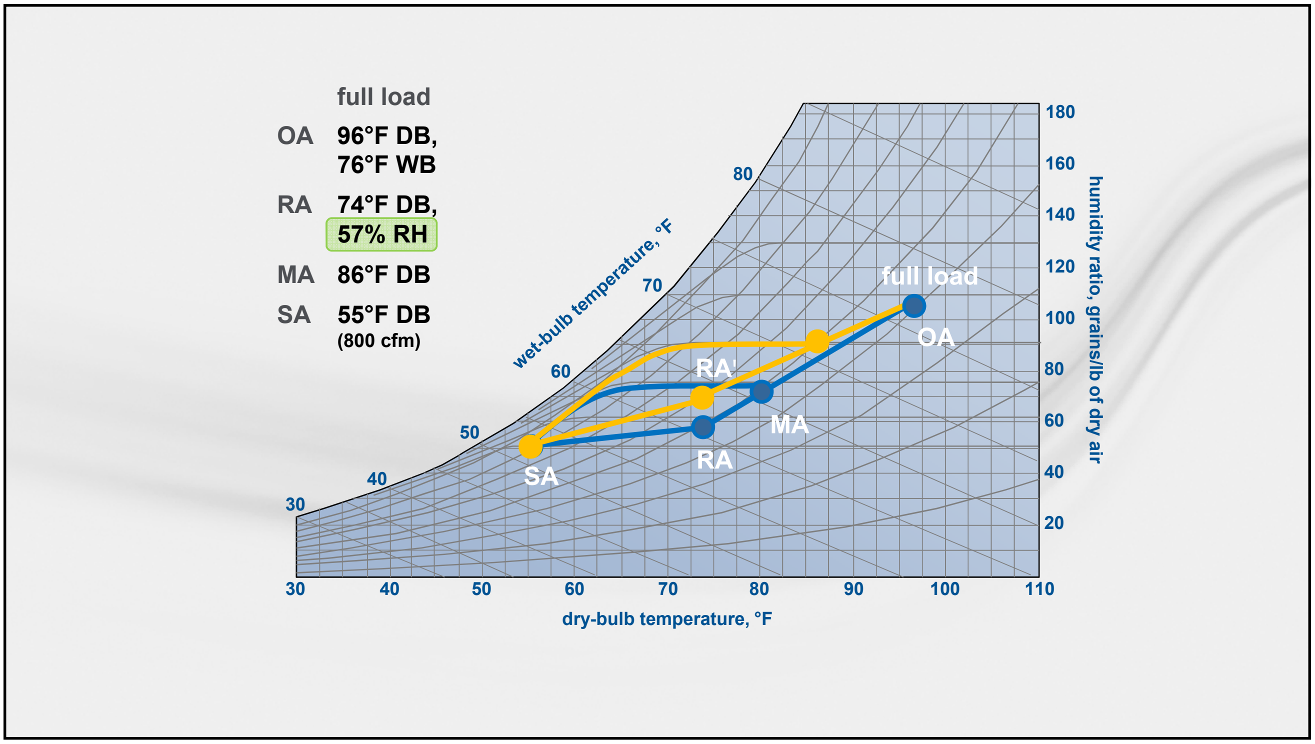
Myth Number 2

55°F supply air temperature is adequate for today’s loads.

---

	full load	
OA	96°F DB, 76°F WB	
RA	74°F DB, 52% RH	
MA	80°F DB	
SA	55°F DB (1,500 cfm)	





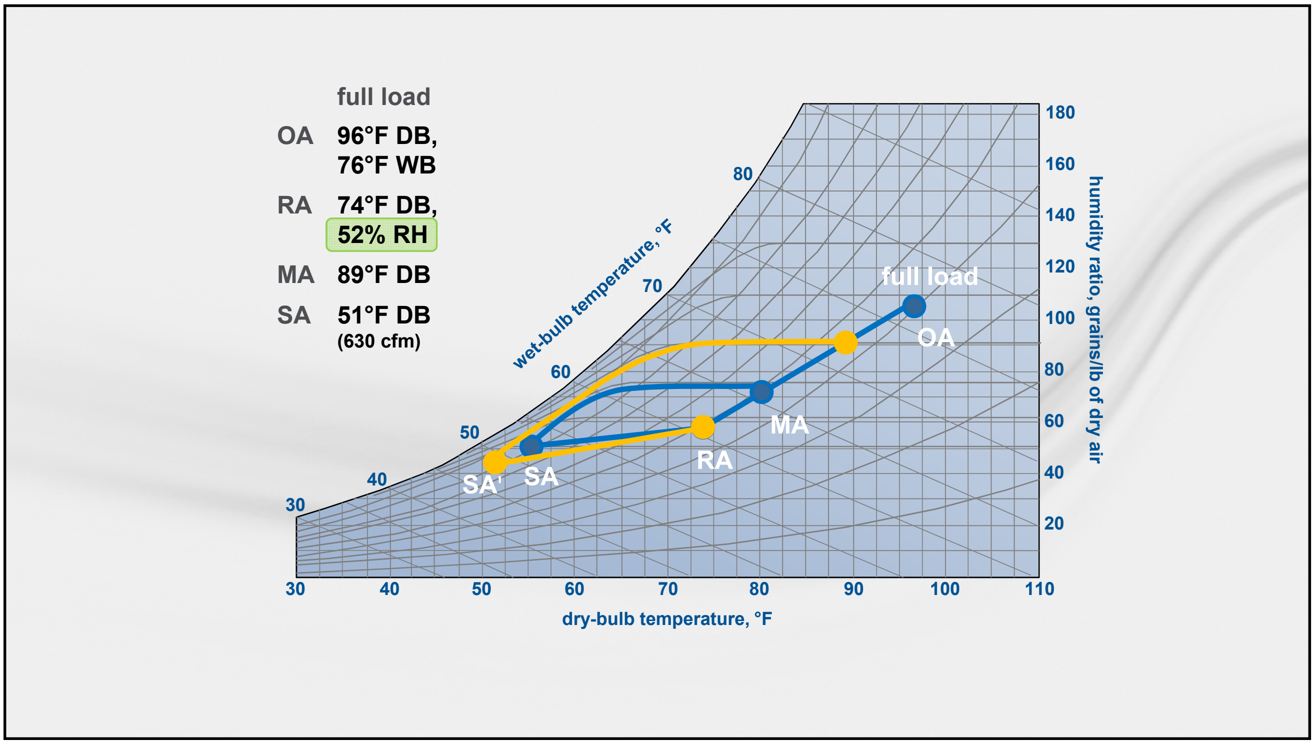
full load

OA 96°F DB,
76°F WB

RA 74°F DB,
57% RH

MA 86°F DB

SA 55°F DB
(800 cfm)



full load

OA 96°F DB,
76°F WB

RA 74°F DB,
52% RH

MA 89°F DB

SA 51°F DB
(630 cfm)

Improving Dehumidification

- Cool and reheat
- Face-and-bypass dampers
- Reduce airflow
- Dual paths
- Desiccants

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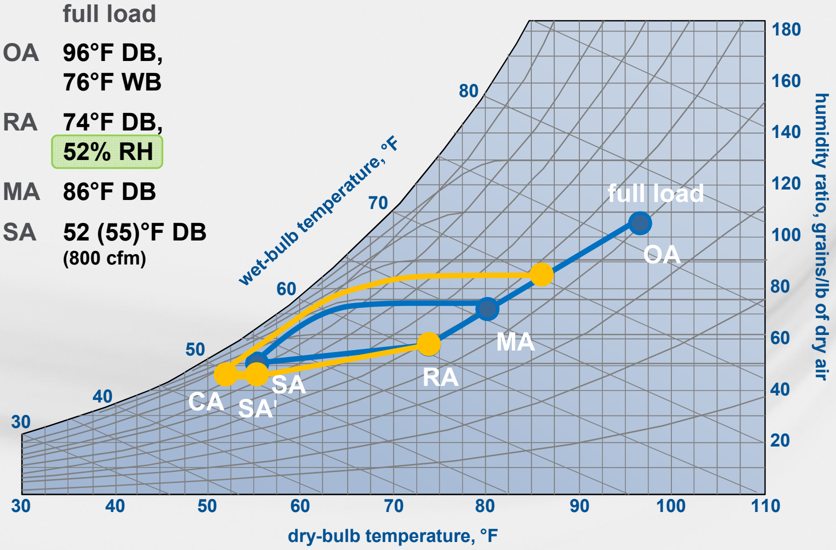
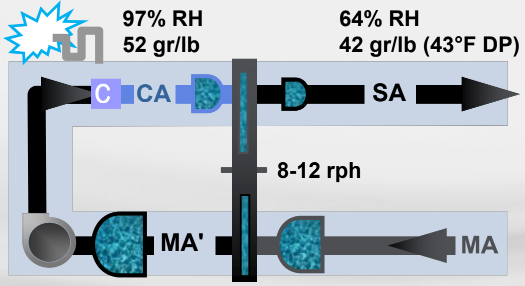
	full load
OA	96°F DB, 76°F WB
RA	74°F DB, 52% RH
MA	86°F DB
SA	52 (55)°F DB (800 cfm)



Type III Series Desiccant (CDQ)



50°F DB
97% RH
52 gr/lb

55°F DB
64% RH
42 gr/lb (43°F DP)

75°F DB
67% RH
87 gr/lb (64°F DP)

80°F DB
50% RH
77 gr/lb (60°F DP)
(10,000 cfm)

---

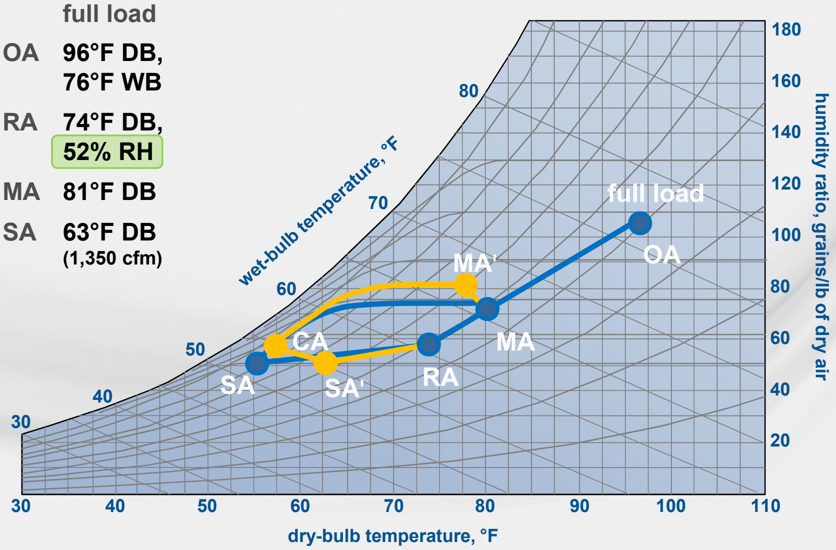
full load

OA 96°F DB,
76°F WB

RA 74°F DB,
52% RH

MA 81°F DB

SA 63°F DB
(1,350 cfm)



Myth Number 2

55°F supply air temperature is adequate for today's a

Myth Number 3

ASHRAE Standard 15 has to be updated before new refrigerants can be used.

ASHRAE

STANDARD

ANSI/ASHRAE Standard 34-2013
(Supersedes ANSI/ASHRAE Standard 34-2010)
Includes ANSI/ASHRAE addenda listed in Appendix H

Designation and Safety Classification of Refrigerants

1. PURPOSE

This standard is intended to establish a simple means of referring to common refrigerants instead of using the chemical name, formula, or trade name. It establishes a uniform system for assigning reference numbers, safety classifications, and refrigerant concentration limits to refrigerants. The standard also identifies requirements to apply for designations and safety classifications for refrigerants and to determine refrigerant concentration limits.

ASHRAE

STANDARD

ANSI/ASHRAE Standard 15-2013
(Supersedes ANSI/ASHRAE Standard 15-2010)
Includes ANSI/ASHRAE addenda listed in Appendix F

Safety Standard for Refrigeration Systems

1. PURPOSE

This standard specifies safe design, construction, installation, and operation of refrigeration systems.

See Appendix F for approval dates by the ASHRAE Standards Committee, the ASHRAE Board of Directors, and the American National Standards Institute.

This standard is under continuous maintenance by a Standing Standard Project Committee (SSPC) for which the Standards Committee has established a documented program for regular publication of addenda or revisions, including procedures for timely, documented, consensus action on requests for change to any part of the standard. The change submittal form, instructions, and deadlines may be obtained in electronic form from the ASHRAE website (www.ashrae.org) or in paper form from the Manager of Standards. The latest edition of an ASHRAE Standard may be purchased from the ASHRAE website (www.ashrae.org) or from ASHRAE Customer Service, 1791 Tullie Circle, NE, Atlanta, GA 30329-2305. E-mail: orders@ashrae.org. Fax: 678-539-2129. Telephone: 404-636-8400 (worldwide), or toll free 1-800-527-4723 (for orders in US and Canada). For reprint permission, go to www.ashrae.org/permissions.

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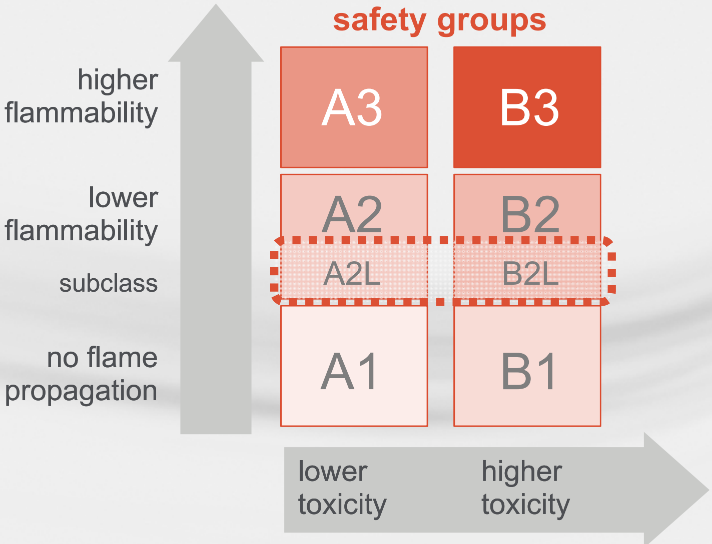
Refrigerant safety groups from ANSI/ASHRAE Standard 34-2013



safety groups

higher flammability	A3	B3
lower flammability	A2	B2
subclass	A2L	B2L
no flame propagation	A1	B1
	lower toxicity	higher toxicity

A2L and B2L are lower flammability refrigerants with a maximum burning velocity of ≤ 10 cm/s (3.9 in./s.).

Standard 34 Addenda on www.ashrae.org

- 35 Addenda
- 3 New refrigerants, 27 blends. Examples:
  - 1233zd(E)
  - 513A
  - 514A
  - 451B

Myth Number [illegible]

ASHRAE Standard[illegible] be updated before new refrigerants can be used.

BUSTED

Myth Number 4

Single-zone VAV units do not need hot gas reheat.

Classroom Example

basic CV system

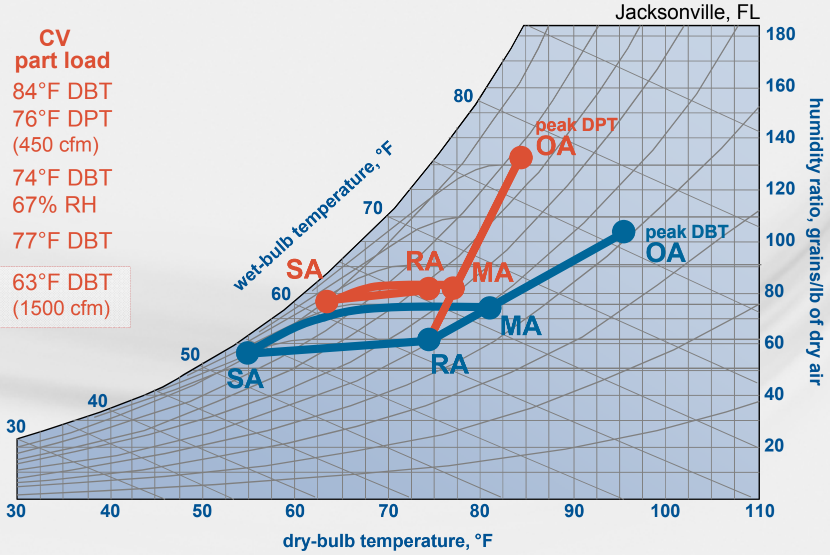
Jacksonville, Florida

	peak DB
outdoor condition	96°F DB, 76°F WB
sensible load	29,750 Btu/h
latent load	5,250 Btu/h
space SHR	0.85
supply airflow	1,500 cfm
outdoor airflow	450 cfm
space temp	74°F
supply air temp	55.7°F

$$1{,}500 \text{ cfm} = \frac{29{,}750 \text{ Btu/h}}{1.085 \times (74°\text{F} - T_{\text{supply}})}$$

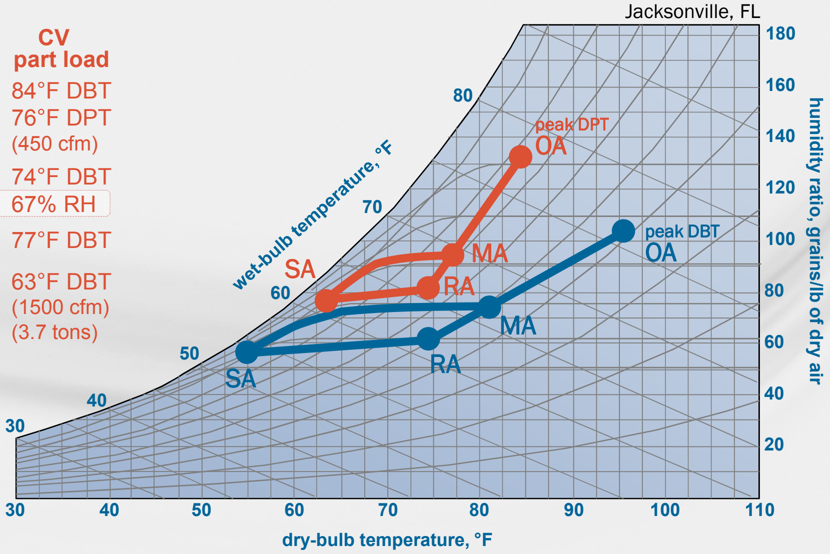
Example: K-12 Classroom

	CV full load	CV part load
OA	96°F DBT	84°F DBT
	68°F DPT	76°F DPT
	(450 cfm)	(450 cfm)
RA	74°F DBT	74°F DBT
	52% RH	67% RH
MA	81°F DBT	77°F DBT
SA	55°F DBT	63°F DBT
	(1500 cfm)	(1500 cfm)
	(4.8 tons)	



Example: K-12 Classroom

	CV full load	CV part load
OA	96°F DBT	84°F DBT
	68°F DPT	76°F DPT
	(450 cfm)	(450 cfm)
RA	74°F DBT	74°F DBT
	52% RH	67% RH
MA	81°F DBT	77°F DBT
SA	55°F DBT	63°F DBT
	(1500 cfm)	(1500 cfm)
	(4.8 tons)	(3.7 tons)

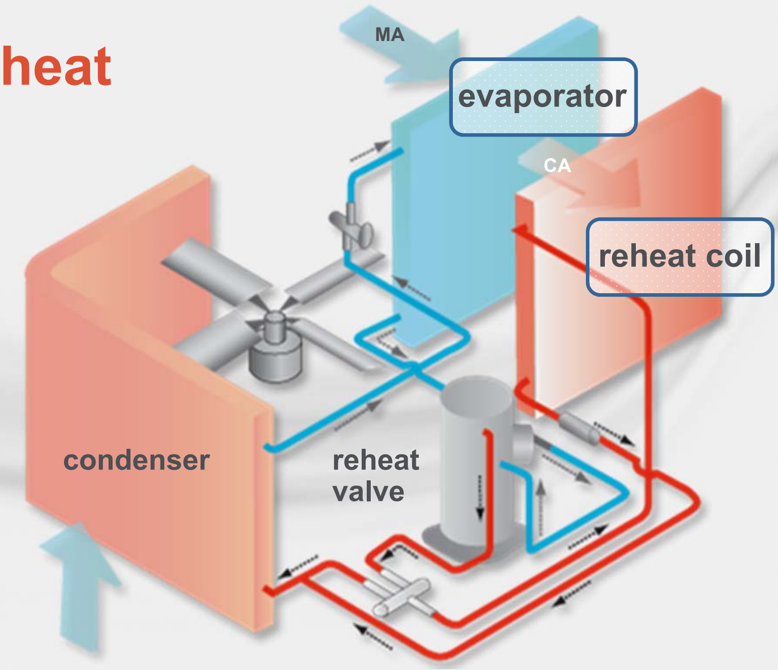


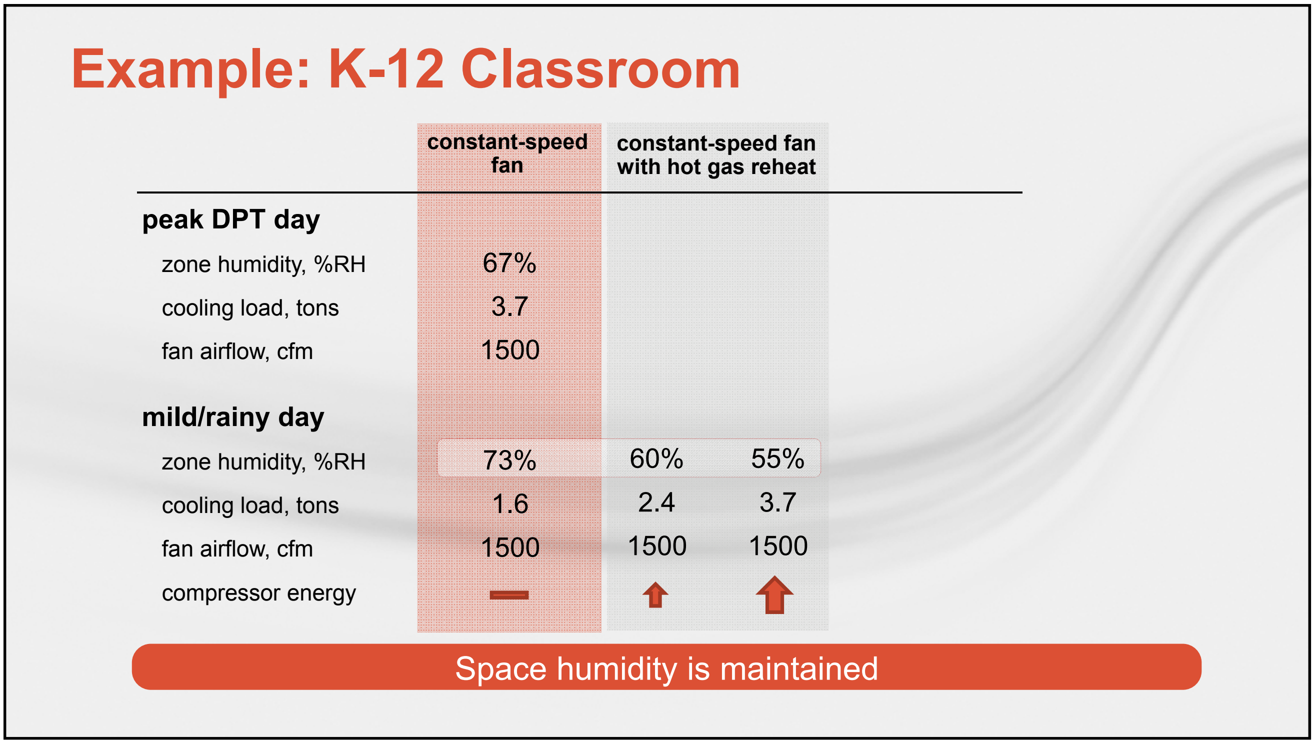
Example: K-12 Classroom

	constant-speed fan
peak DPT day	
zone humidity, %RH	67%
cooling load, tons	3.7
fan airflow, cfm	1500
mild/rainy day	
zone humidity, %RH	73%
cooling load, tons	1.6
fan airflow, cfm	1500

Hot Gas Reheat

packaged DX units

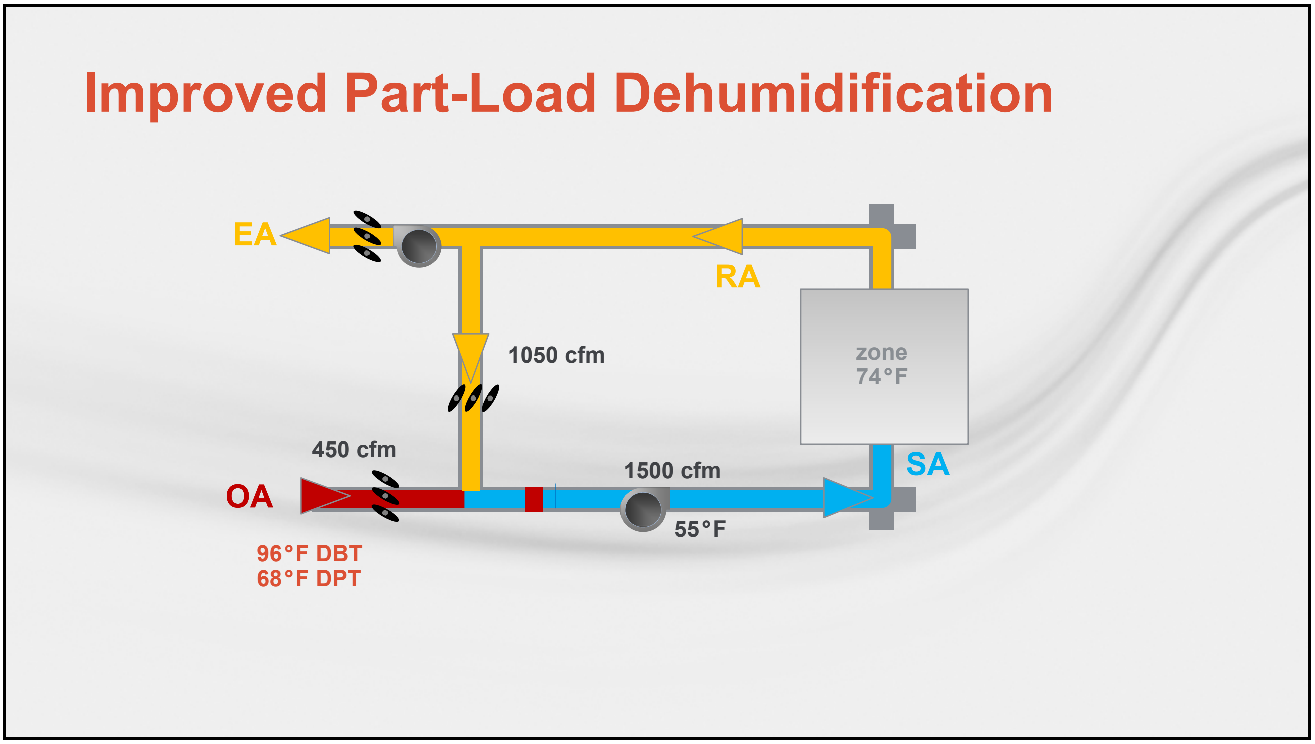




Example: K-12 Classroom

	constant-speed fan	constant-speed fan with hot gas reheat	
peak DPT day			
zone humidity, %RH	67%		
cooling load, tons	3.7		
fan airflow, cfm	1500		
mild/rainy day			
zone humidity, %RH	73%	60%	55%
cooling load, tons	1.6	2.4	3.7
fan airflow, cfm	1500	1500	1500
compressor energy	—	↑	⬆

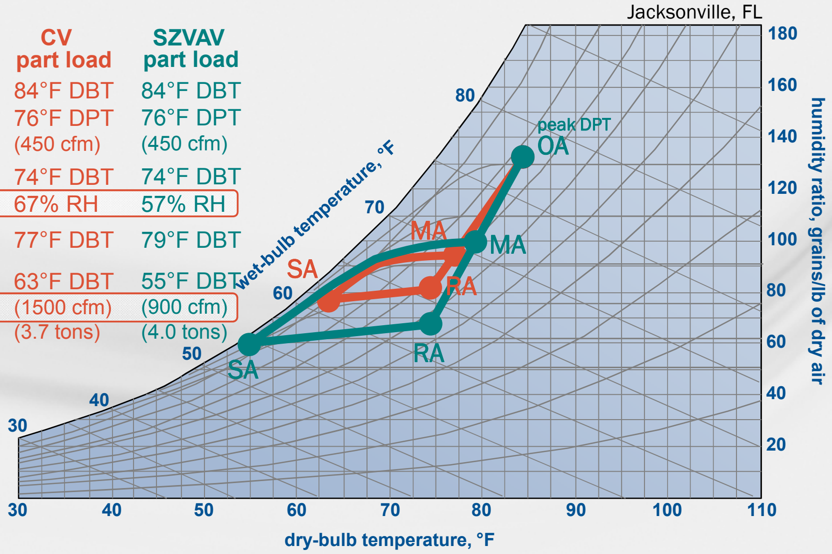
Space humidity is maintained



Improved Part-Load Dehumidification

Example: K-12 Classroom

	CV full load	CV part load	SZVAV part load
OA	96°F DBT	84°F DBT	84°F DBT
	68°F DPT	76°F DPT	76°F DPT
	(450 cfm)	(450 cfm)	(450 cfm)
RA	74°F DBT	74°F DBT	74°F DBT
	52% RH	67% RH	57% RH
MA	81°F DBT	77°F DBT	79°F DBT
SA	55°F DBT	63°F DBT	55°F DBT
	(1500 cfm)	(1500 cfm)	(900 cfm)
	(4.8 tons)	(3.7 tons)	(4.0 tons)



Example: K-12 Classroom

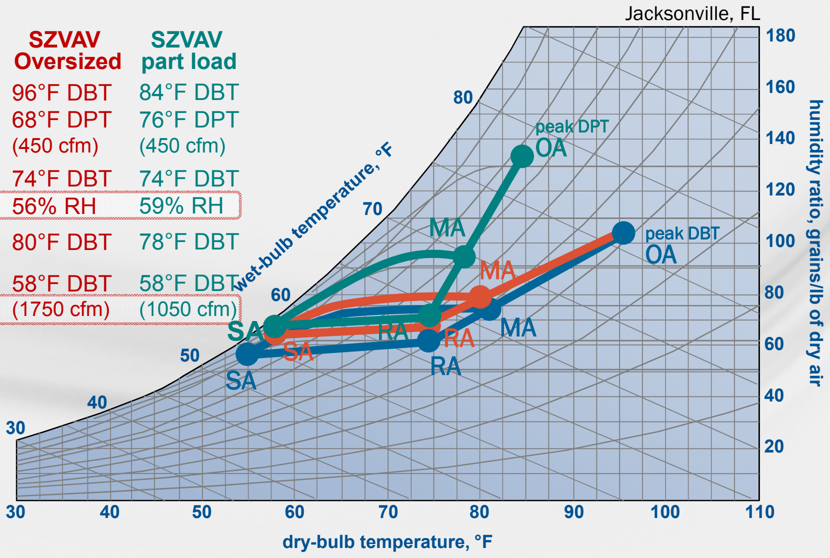
	constant-speed fan	constant-speed fan with hot gas reheat		variable-speed fan
peak DPT day				
zone humidity, %RH	67%			57%
cooling load, tons	3.7			4.0
fan airflow, cfm	1500			900
mild/rainy day				
zone humidity, %RH	73%	60%	55%	60%
cooling load, tons	1.6	2.4	3.7	1.9
fan airflow, cfm	1500	1500	1500	750

SZVAV Dehumidification Performance

- VAV may be enough
- Consider hot gas reheat for:
  - Even lower space humidity levels
  - Widely varying loads
  - Oversized units

Example: K-12 Classroom

	SZVAV full load	SZVAV Oversized	SZVAV part load
OA	96°F DBT	96°F DBT	84°F DBT
	68°F DPT	68°F DPT	76°F DPT
	(450 cfm)	(450 cfm)	(450 cfm)
RA	74°F DBT	74°F DBT	74°F DBT
	52% RH	56% RH	59% RH
MA	81°F DBT	80°F DBT	78°F DBT
SA	55°F DBT	58°F DBT	58°F DBT
	(1500 cfm)	(1750 cfm)	(1050 cfm)



Avoid Oversizing!

- Oversizing supply airflow leads to:
  - Warmer supply-air temperature
  - Less dehumidification (in non-arid climates)
  - Elevated indoor humidity
- Examples include:
  - Auditoriums
  - Gymnasiums
  - Church sanctuaries
  - Etc.

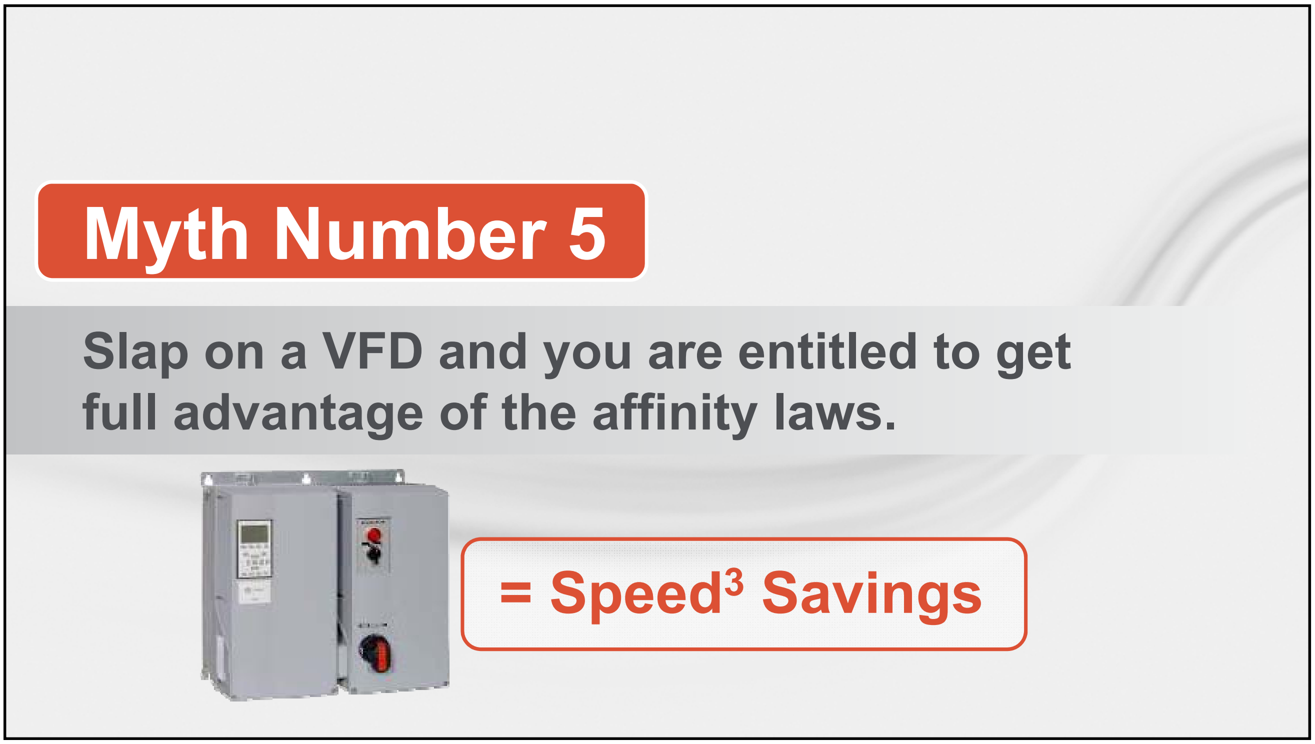
Humidity Control with SZVAV

- Avoid oversizing equipment
- Verify proper fan speed and discharge air temperature setpoints
- Equip the unit with hot gas reheat, if necessary

Myth Number 4

BUSTED

Single-zon[illegible] VA[illegible] u[illegible] n[illegible] need
hot ga[illegible]t



Myth Number 5

Slap on a VFD and you are entitled to get full advantage of the affinity laws.

= Speed^3 Savings

The Affinity Laws

dynamic compression fans/impellers

Background:

1. Fans, pump impellers and other “dynamic compression” devices.
2. Application limited to systems with only frictional flow losses.
3. Ignoring changes in device efficiency at different conditions.

If and only if the above are true then:

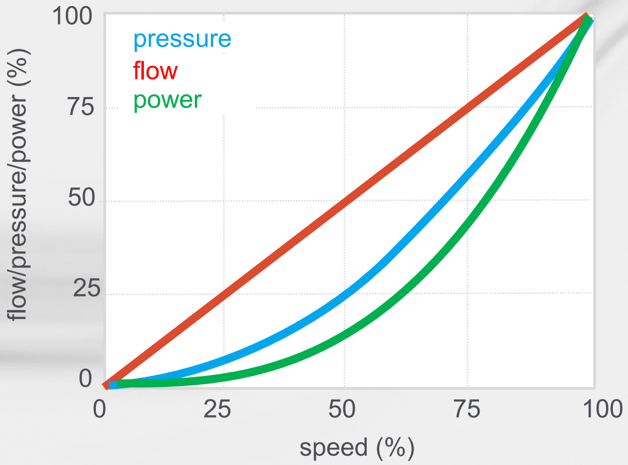
1. Pressure varies proportionally to the square of the impeller speed.
2. Flow produced varies proportionally to the impeller speed.
3. Power (BHP) required varies in a cubic proportion to the impeller speed.

The Affinity Laws – Graphically

dynamic compression fans/impellers

Device performance in frictional pressure loss systems

- Pressure is proportional to the speed squared
- Flow is proportional to the speed
- Power is proportional to the speed cubed

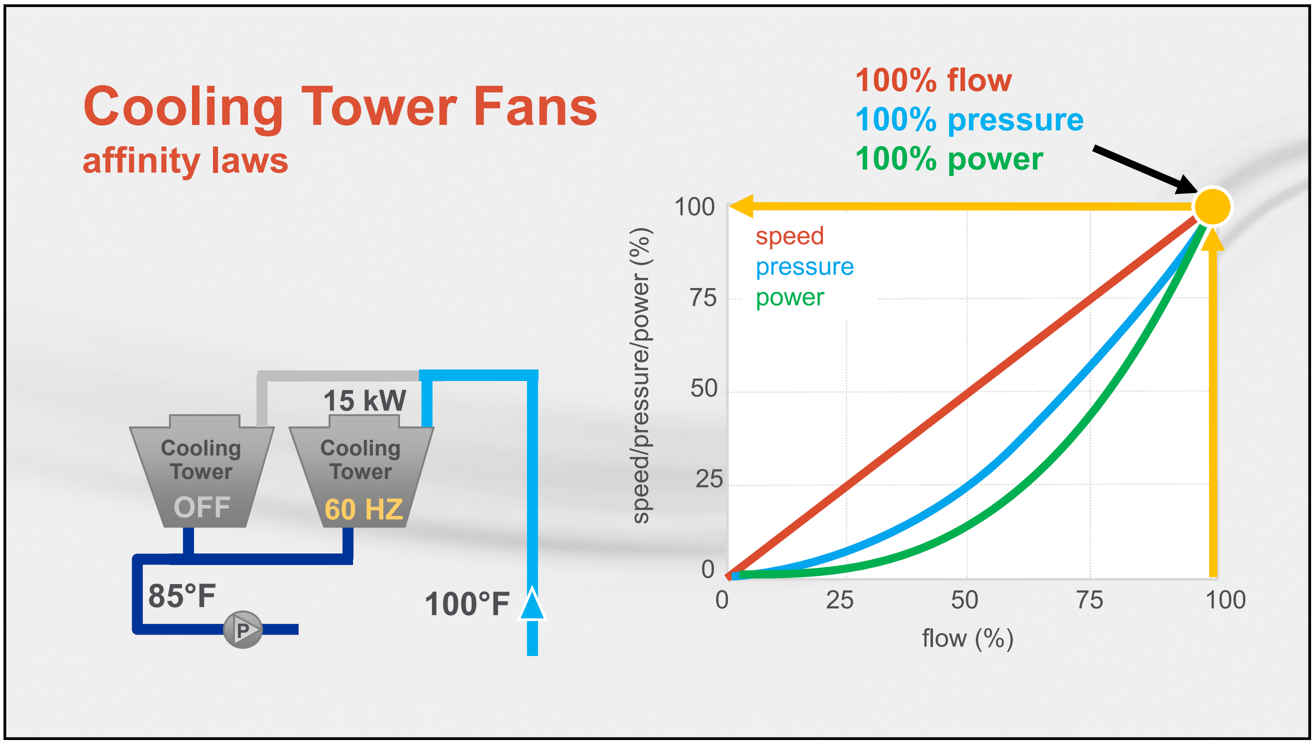


Systems and the Affinity Laws

compliant systems

Systems that comply

- Cooling towers
- Single zone VAV air systems.



Cooling Tower Fans

affinity laws

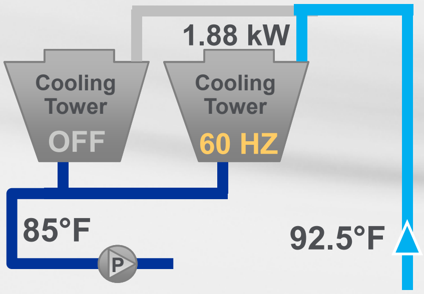
100% flow
100% pressure
100% power

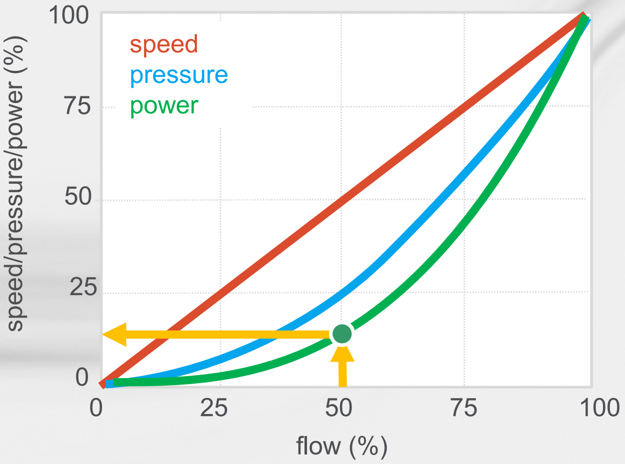
Cooling Tower Fans

affinity laws

For “Free Discharge” Fans
$W2 = W1 \times (S2 / S1)^3$

$W2 = 15\ kW \times (30 / 60)^3$
$W2 = 1.88\ kW$





Systems and the Affinity Laws

non-compliant systems

Systems that don’t comply:

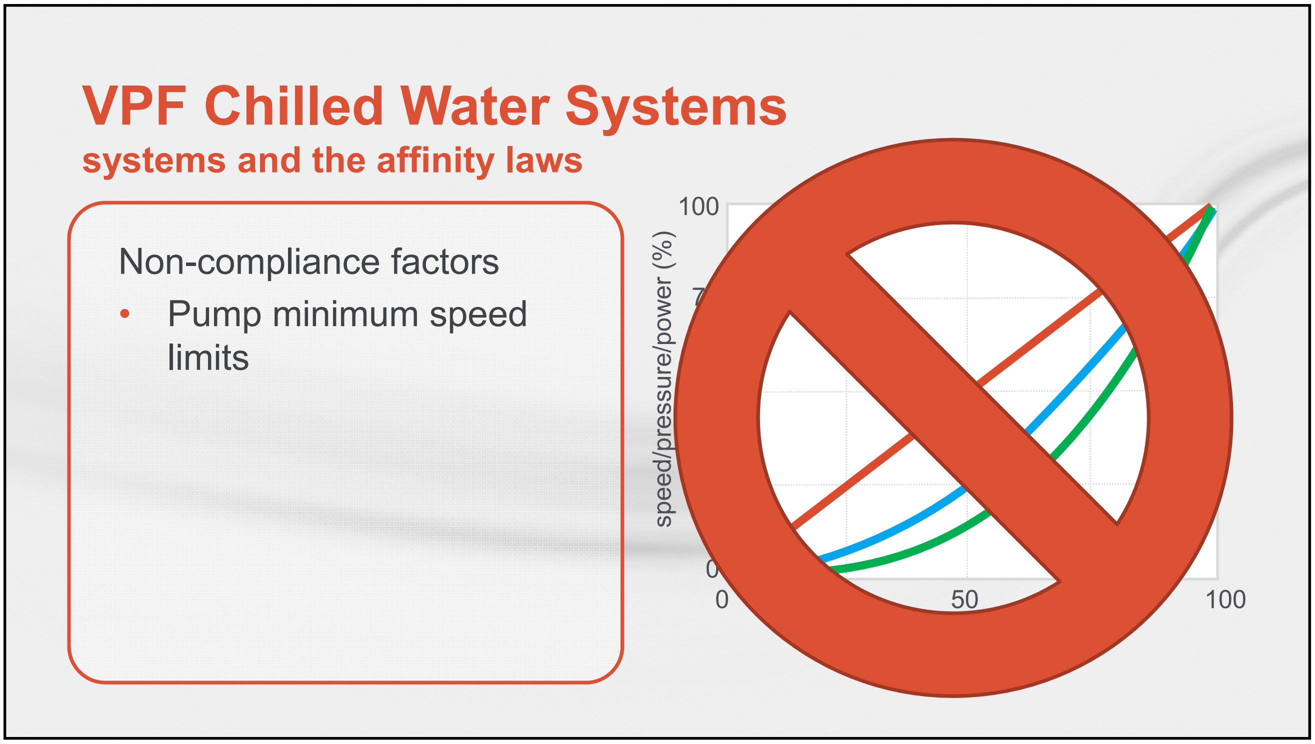
- Chilled water
- Hot water
- MultiZone VAV
- Condenser water
- HVAC cooling units
- HVAC heating units (HP)

Non-compliant characteristics:

Control valves and setpoints

Fixed lift

Refrigeration lift / heat exchangers / minimum flows

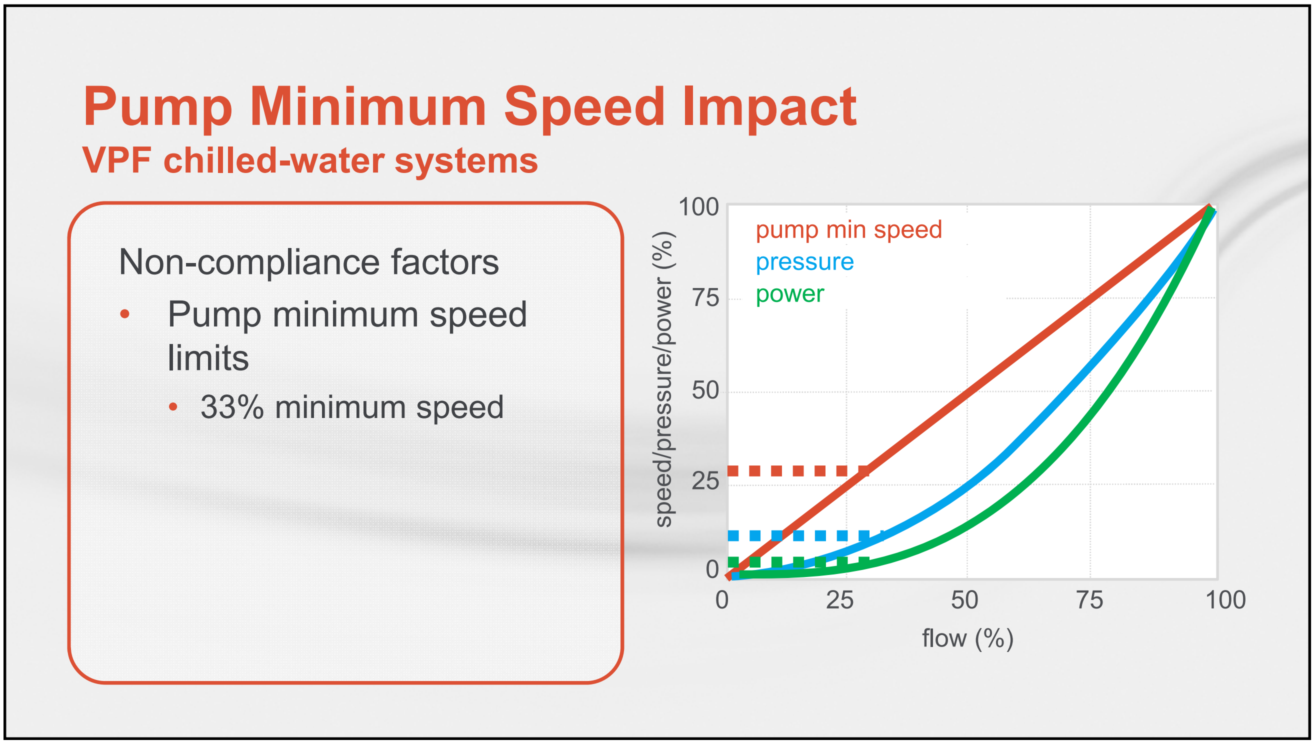


VPF Chilled Water Systems

systems and the affinity laws

Non-compliance factors

- Pump minimum speed limits

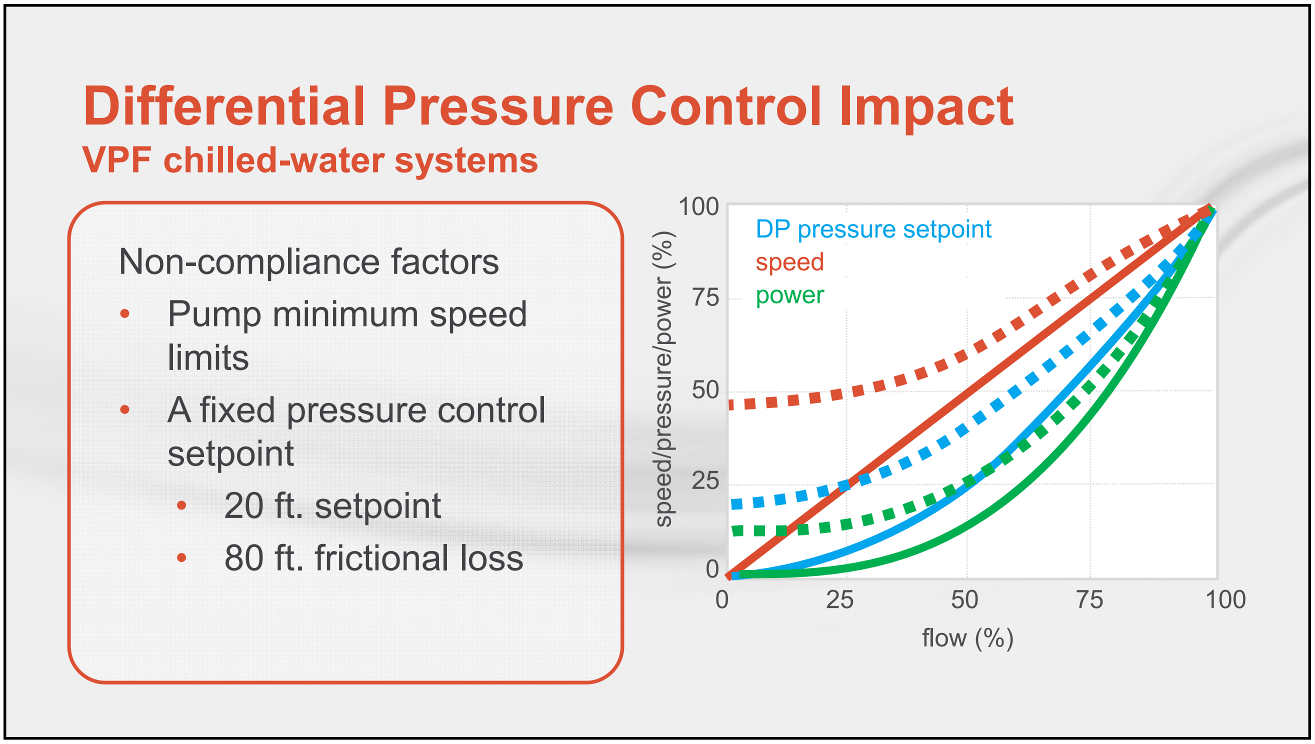


Pump Minimum Speed Impact

VPF chilled-water systems

Non-compliance factors

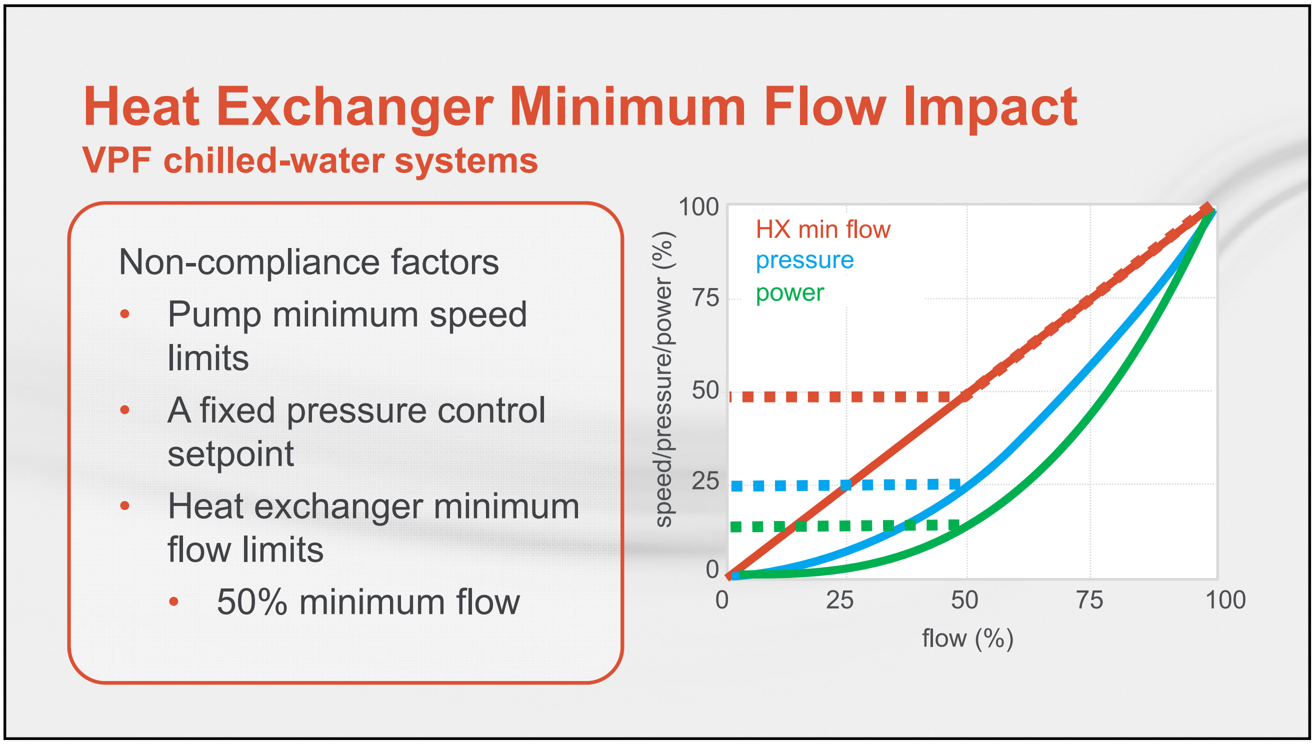
- Pump minimum speed limits
  - 33% minimum speed



Differential Pressure Control Impact

VPF chilled-water systems

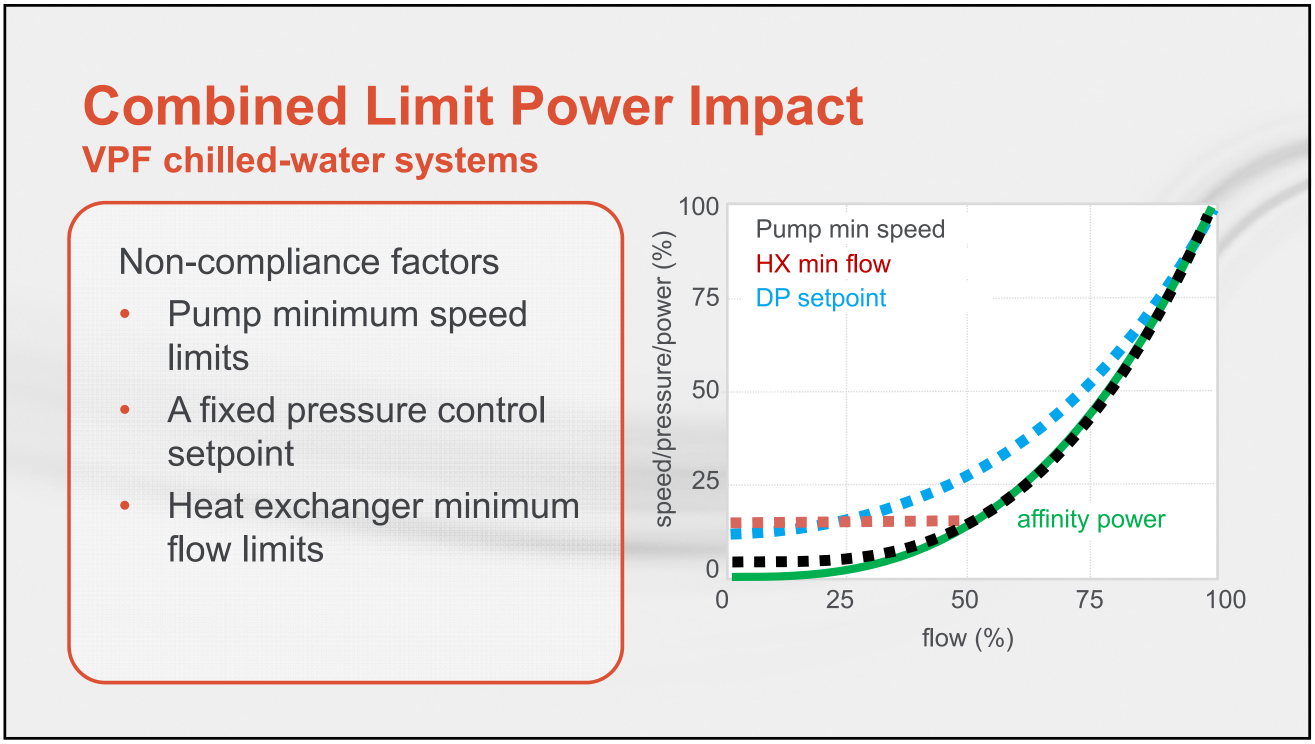
Non-compliance factors
- Pump minimum speed limits
- A fixed pressure control setpoint
  - 20 ft. setpoint
  - 80 ft. frictional loss



Heat Exchanger Minimum Flow Impact

VPF chilled-water systems

Non-compliance factors
- Pump minimum speed limits
- A fixed pressure control setpoint
- Heat exchanger minimum flow limits
  - 50% minimum flow

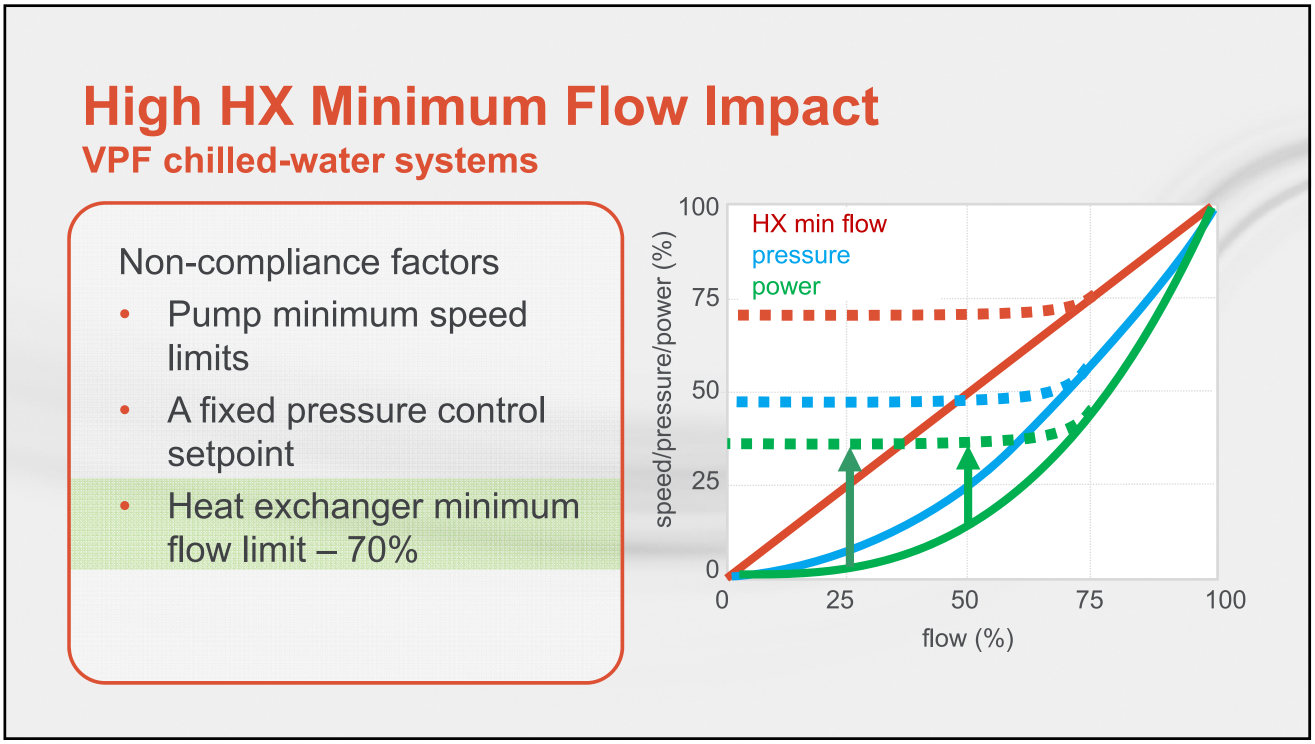


Combined Limit Power Impact

VPF chilled-water systems

Non-compliance factors

- Pump minimum speed limits
- A fixed pressure control setpoint
- Heat exchanger minimum flow limits



High HX Minimum Flow Impact

VPF chilled-water systems

Non-compliance factors

- Pump minimum speed limits
- A fixed pressure control setpoint
- Heat exchanger minimum flow limit – 70%

Myth Number 5

Slap on a VFD and you are entitled to get full advantage of the affinity laws.

Systems that comply

- Cooling towers
- Single zone VAV HVAC systems.

Systems that don't comply

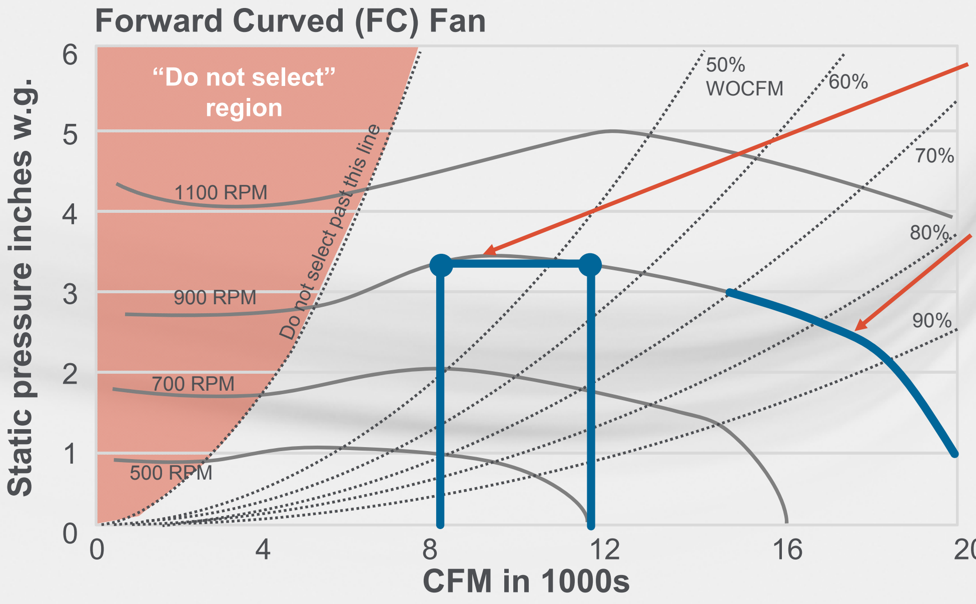
- Chilled water
- Hot water
- Condenser water
- Multi-zone VAV
- HVAC cooling units
- HVAC heating units (HP)

BUSTED

Myth Number 6

Small changes in pressure can have a huge impact on airflow for flat fan curves and may cause a fan system to surge.

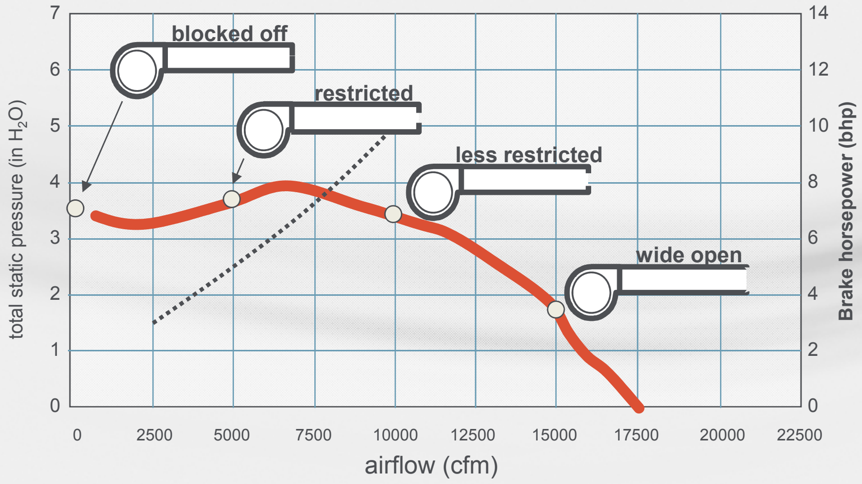
Small Changes in Pressure



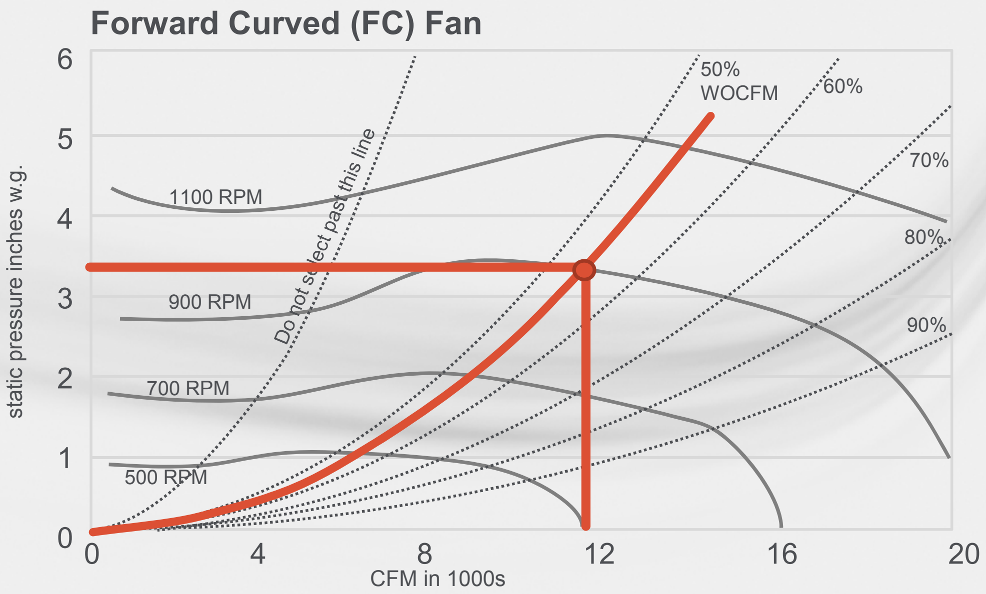
peak pressure

Far from peak – some may select here

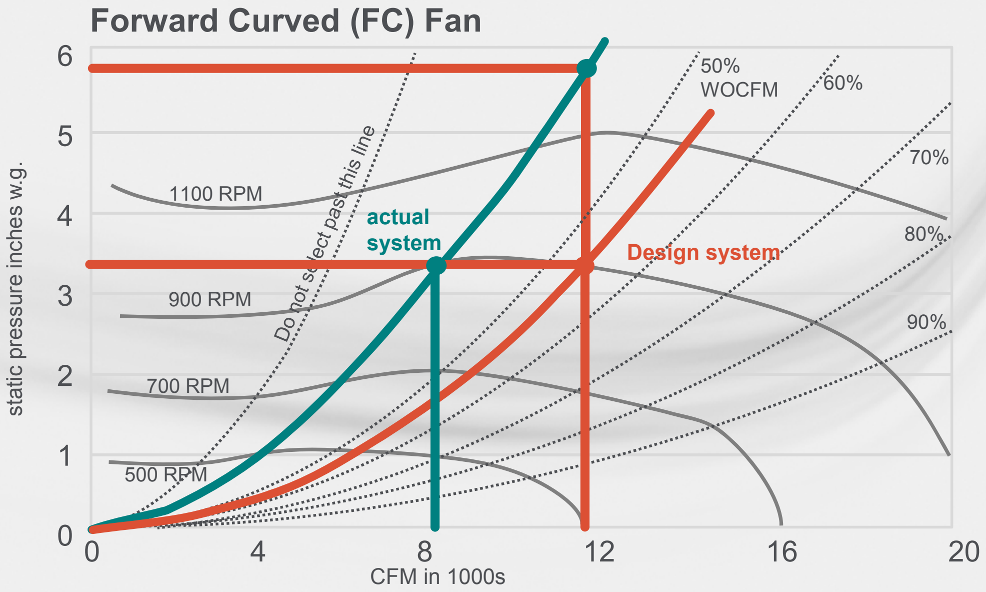
Fan Performance Test

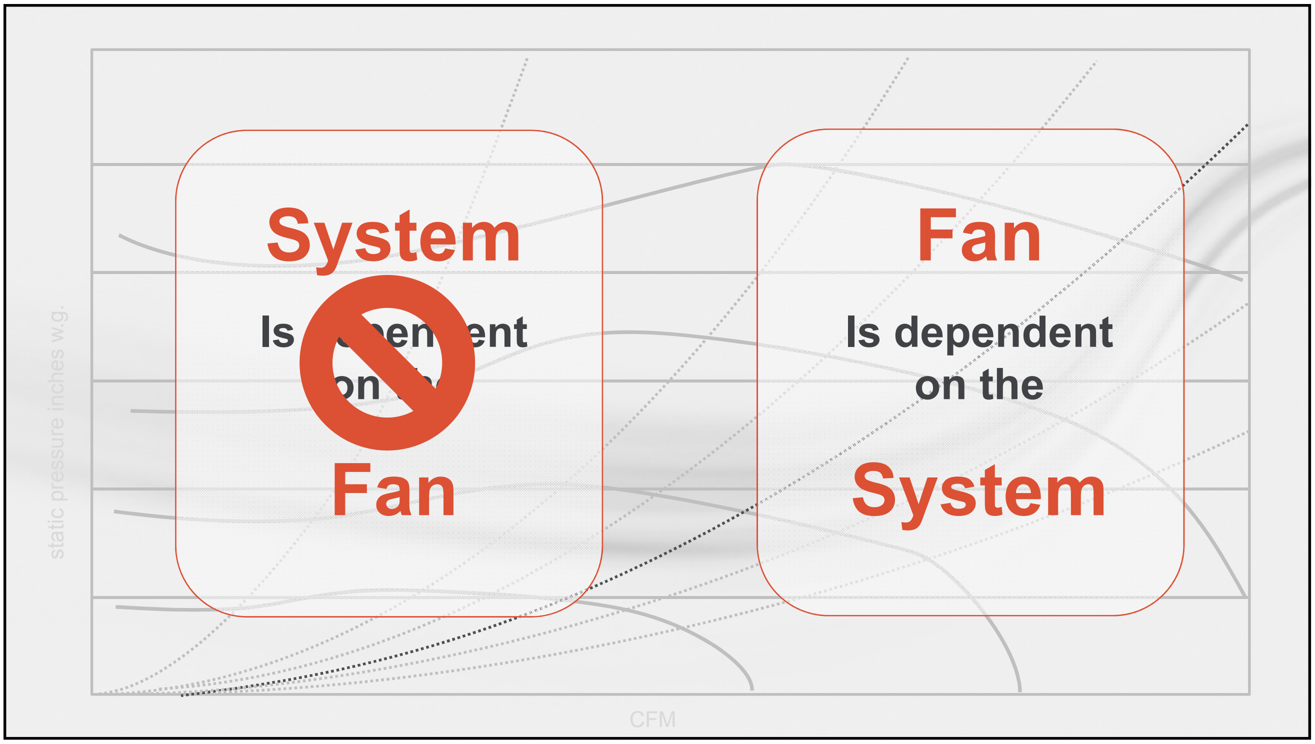


Pressure = f(Airflow)



Two Different Systems





System

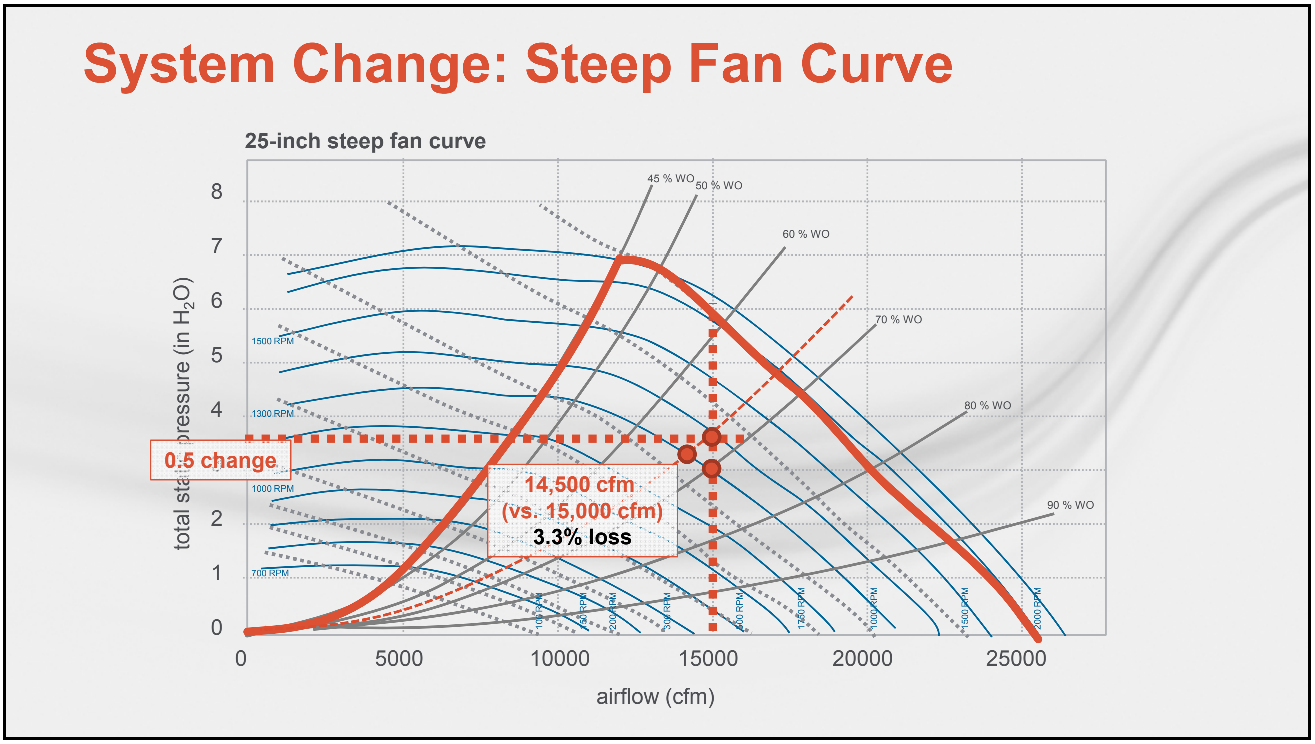
Is dependent on the

Fan

Fan

Is dependent on the

System



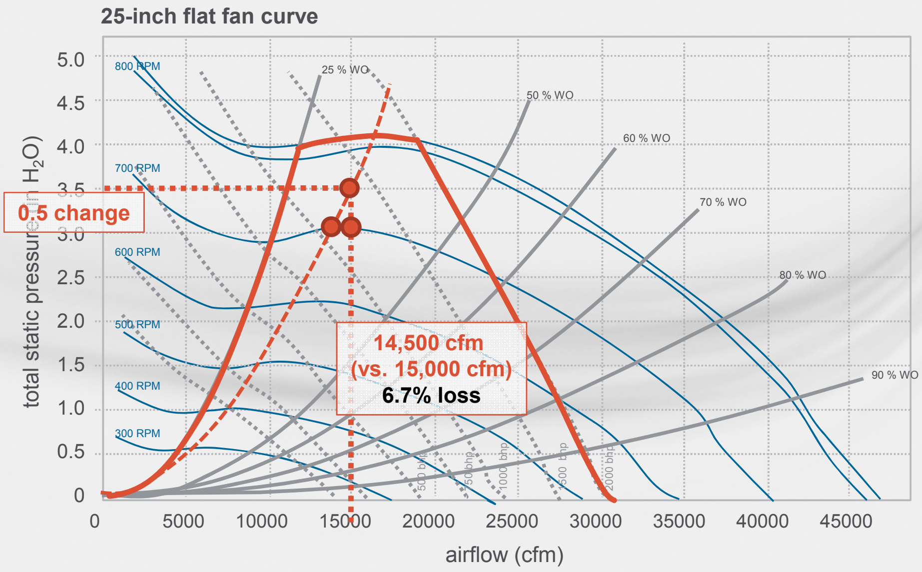
System Change: Steep Fan Curve

25-inch steep fan curve

0.5 change

14,500 cfm
(vs. 15,000 cfm)
3.3% loss

System Change: Flat Fan Curve



25-inch flat fan curve

0.5 change

14,500 cfm
(vs. 15,000 cfm)
6.7% loss

Myth Number 6

Small changes in p[illegible] can have a huge impact on airflow for flat fan curves and may cause a fan system to surge.

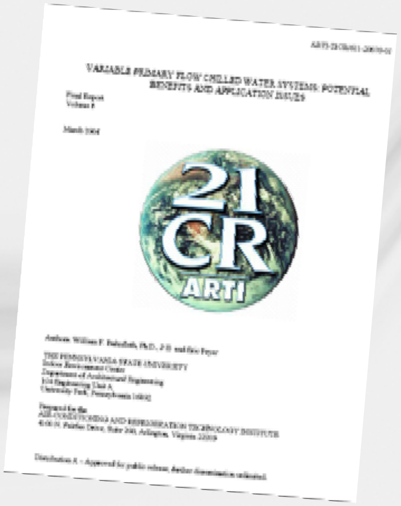
BUSTED

Myth Number 7

A chilled water system needs to be variable primary flow to be efficient.

Variable Primary Flow (VPF) Savings

- First cost: 4-8%
- Annual energy: 3-8%
- Life-cycle cost: 3-5%



VARIABLE PRIMARY FLOW CHILLED WATER SYSTEMS: POTENTIAL BENEFITS AND APPLICATION ISSUES

Compared VPF and Primary-Secondary

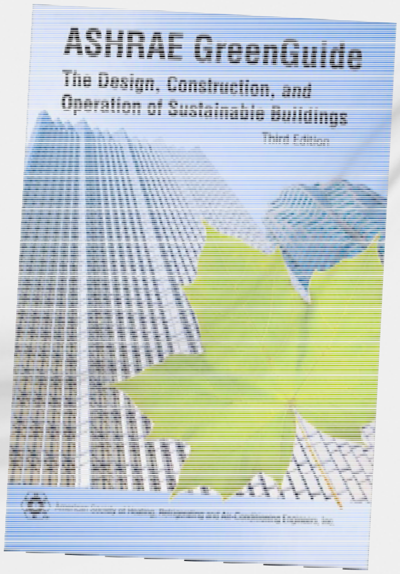
Low Pump Power

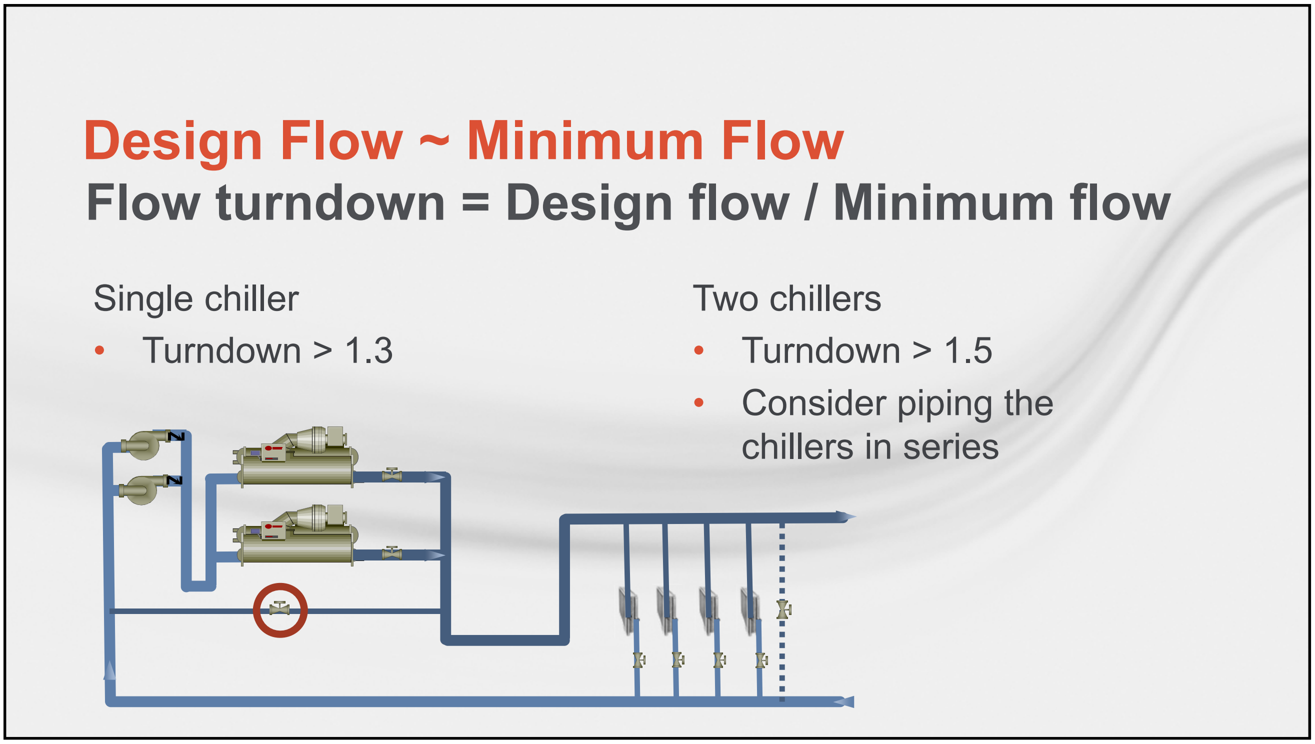
- Install pump VSD
- Use VSD to set design flow rate
- Open balancing valve
- Employ chilled water reset

If system is constant flow – reduce design flow rate further

Design Flow ~ Minimum Flow

- Chilled water ∆Ts
  - ASHRAE GreenGuide (16-18°F ∆T)
  - 90.1-2016 Section 6.5.4.7
    - Coil shall be selected to *"…provide 15°F or higher temperature difference between leaving and entering temperatures."*
- Chillers with limited evaporator choices





Design Flow ~ Minimum Flow

Flow turndown = Design flow / Minimum flow

Single chiller

- Turndown > 1.3

Two chillers

- Turndown > 1.5
- Consider piping the chillers in series

Conversion from Primary-Secondary

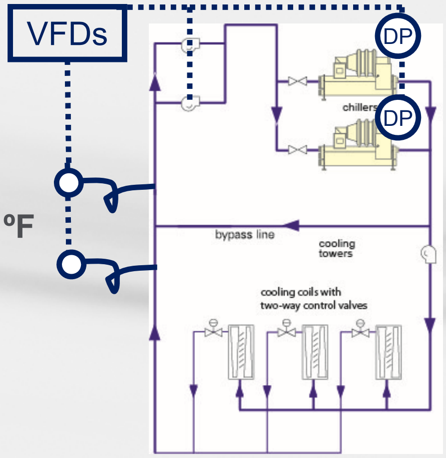
Convert to VPF

- If entire plant is being re-piped anyway
- Additional capacity is needed
- Install chiller where primary pumps used to be

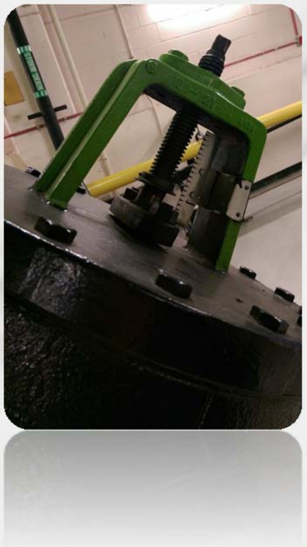
Change to Variable-Primary/Variable-Secondary

- Cooling capacity is adequate
- Piping changes are minimal

Convert to Variable Primary/Variable Secondary



ΔT = 1 or 2 ºF



Plant Operator

- Doesn't understand the plant
- Wants to manually control

Myth Number 7

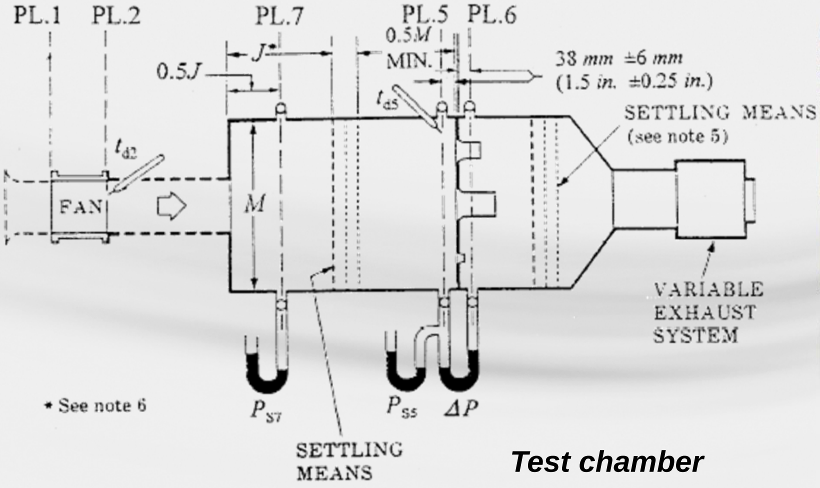
A chilled water system needs to be variable primary flow to be efficient.

Myth Number 8

Fans often don’t deliver the airflow they are supposed to—and it’s the fan’s fault.

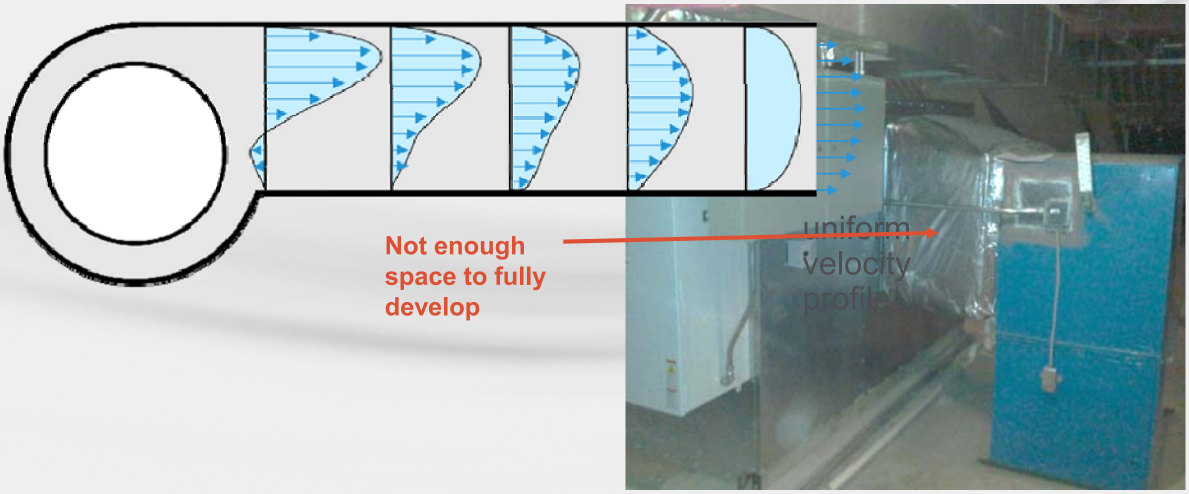
ANSI/ASHRAE 51 (AMCA 210-07)

Figure 12: “Outlet Chamber Setup-Multiple Nozzles in Chamber”



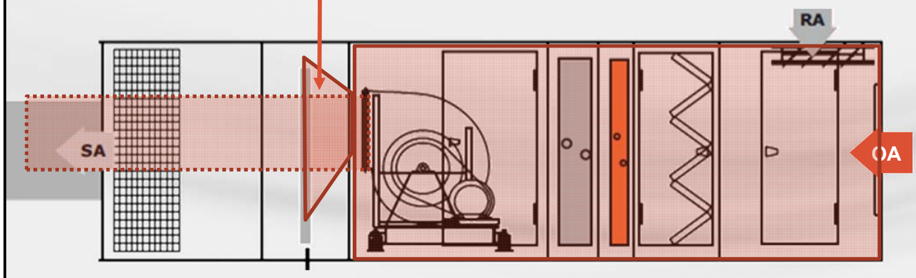
Test chamber

System Effect: Developing a Uniform Velocity Profile



Not enough space to fully develop

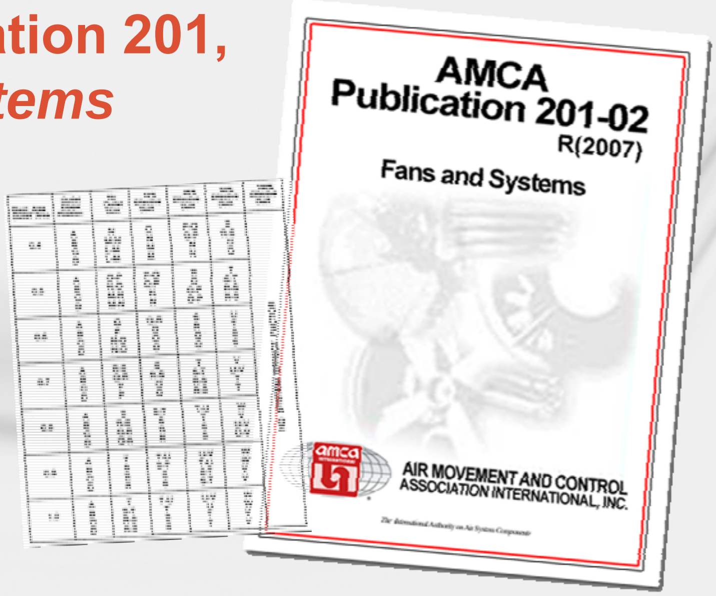
Free and Abrupt Discharge



"Hidden" loss as
high as 1.0
inches w.g.

AMCA Publication 201,
Fans and Systems

Prediction of
common System
Effect Factors



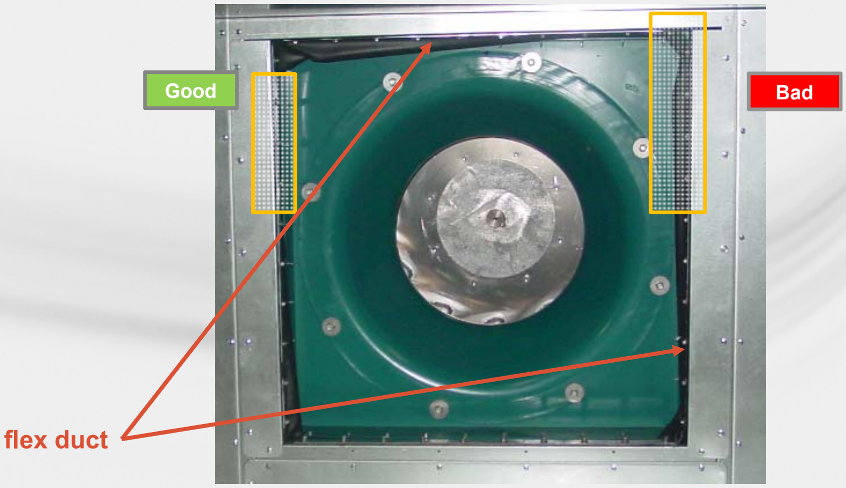
AMCA
Publication 201-02
R(2007)
Fans and Systems

AIR MOVEMENT AND CONTROL
ASSOCIATION INTERNATIONAL, INC.

Common System Effects

- Open discharge, elbow, branch, turning vanes, or damper located too close to the fan outlet
- Elbow, turning vanes, air straightener, or other obstruction located too close to the fan inlet
- Pre-swirling the air prior to it entering the fan wheel
- Use of an inlet plenum or cabinet

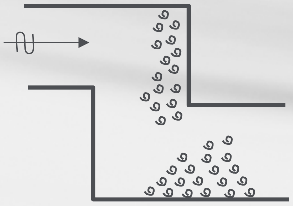
Flex Duct Problems



Low Airflow Troubleshooting

Common problems:

- Unexpectedly high system pressures
- Leaks
- Fan installed or running backwards

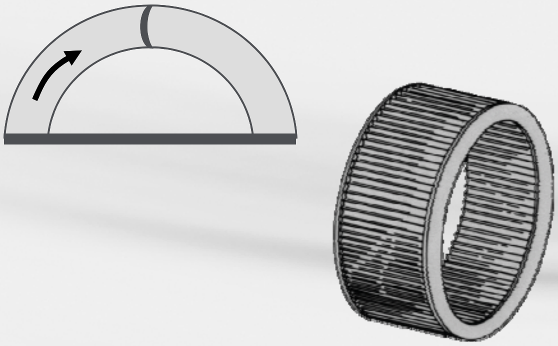


Close-coupled fittings

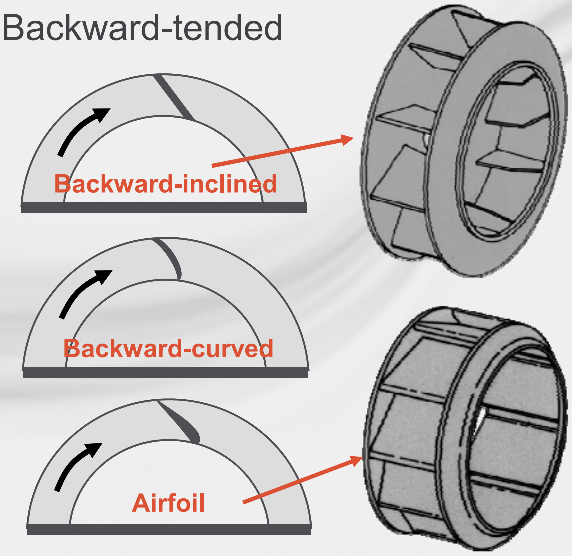
From A Practical Guide to Noise and Vibration Control for HVAC Systems (M. Schaffer, 1991)

Fan Rotation

Forward-curved



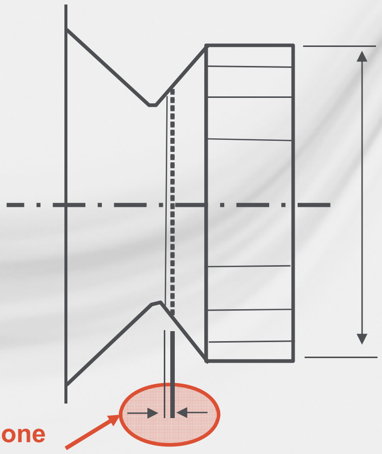
Backward-tended



Low Airflow Troubleshooting

Over-amping problems:

- Bad component (motor or bearings)
- Installation:
  - Wheel-cone overlap
  - Belt tension, belt/shaft alignment

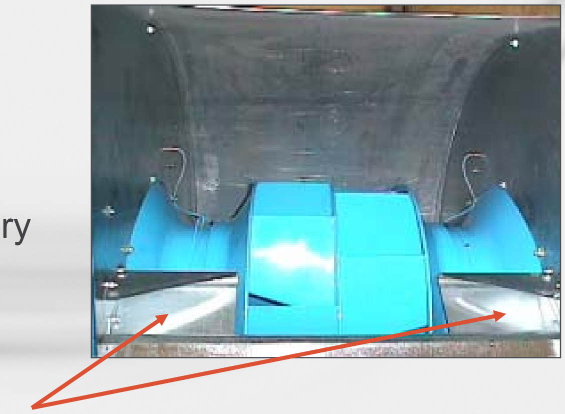


Wheel-cone overlap

Low Airflow Troubleshooting

Uncommon problems:

- Wrong fan installed
- Cutoff issues (housed fans only)
- Software/catalog error
- Quantum mechanics & string theory



Fan cutoff (split in this case)

Field Measurements

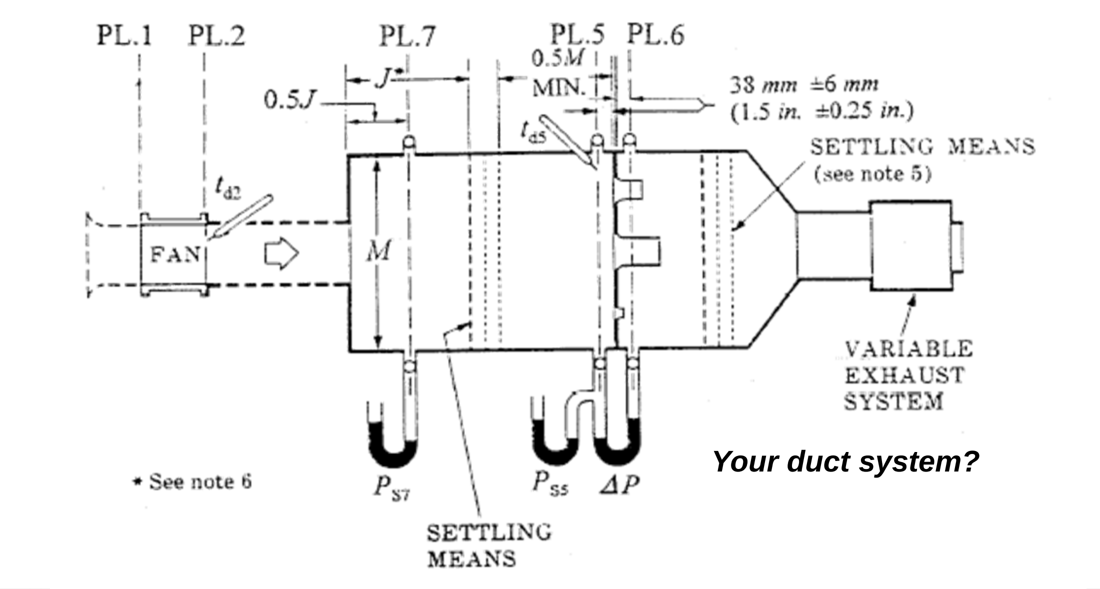
Evaluating the right parameters:

- Airflow
- Pressure
- Speed
- Power

Things to watch out for:

- VFD settings
- Damper position (systems with a return or exhaust fan)
- Parameter measurement error

Parameter Measurement Error



Your duct system?

Parameter Measurement Error

AMCA 203 “Field Performance Measurement of Fan Systems”

http://www.amca.org/

AMCA Fan Application Manual

Publication 201 “Fans and Systems”

- Lists possible causes for low flow, including:
    - Improper inlet duct design
    - Improper outlet duct design
    - Improper fan installation
    - Unexpected system resistance characteristics
    - Improper allowance for fan system effect
    - Dirty filters, ducts, coils
    - “Performance” determined using uncertain field measurement techniques
- Includes much help for system effect corrections

AMCA Fan Application Manual

Publication 202 “Troubleshooting”

- Lists possible causes for low airflow, including:
    - Improper fan installation or assembly
    - Damage in handling or transit
    - System design error
    - Deterioration of system
    - Faulty controls
    - Poor fan selection
- Includes detailed troubleshooting checklists

Myth Number 8

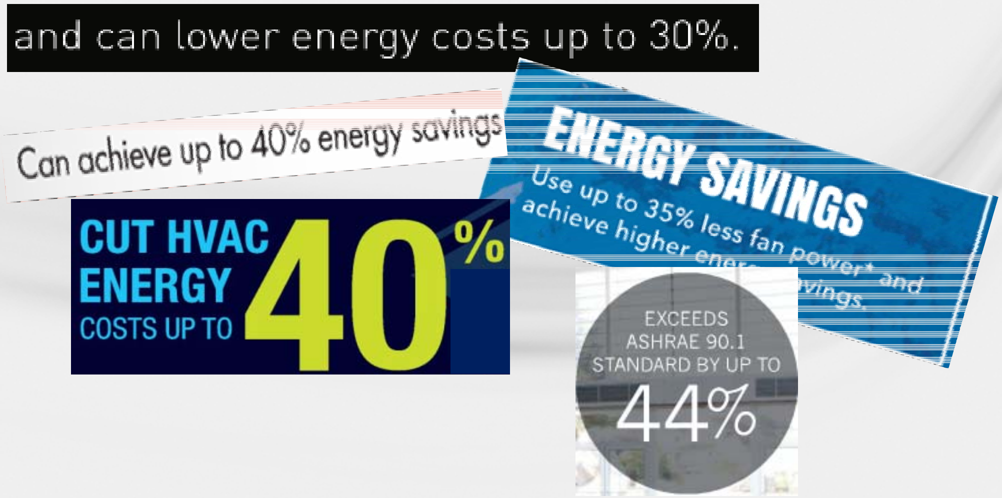
DEPENDS

Fans oft[illegible] airflow they ar[illegible]—and it’s the fan’s fault.

Myth Number 9

You can save (20, 30, 40, 50 80) percent….just by doing this…

Savings Claims – 40 is the new 30!



and can lower energy costs up to 30%.

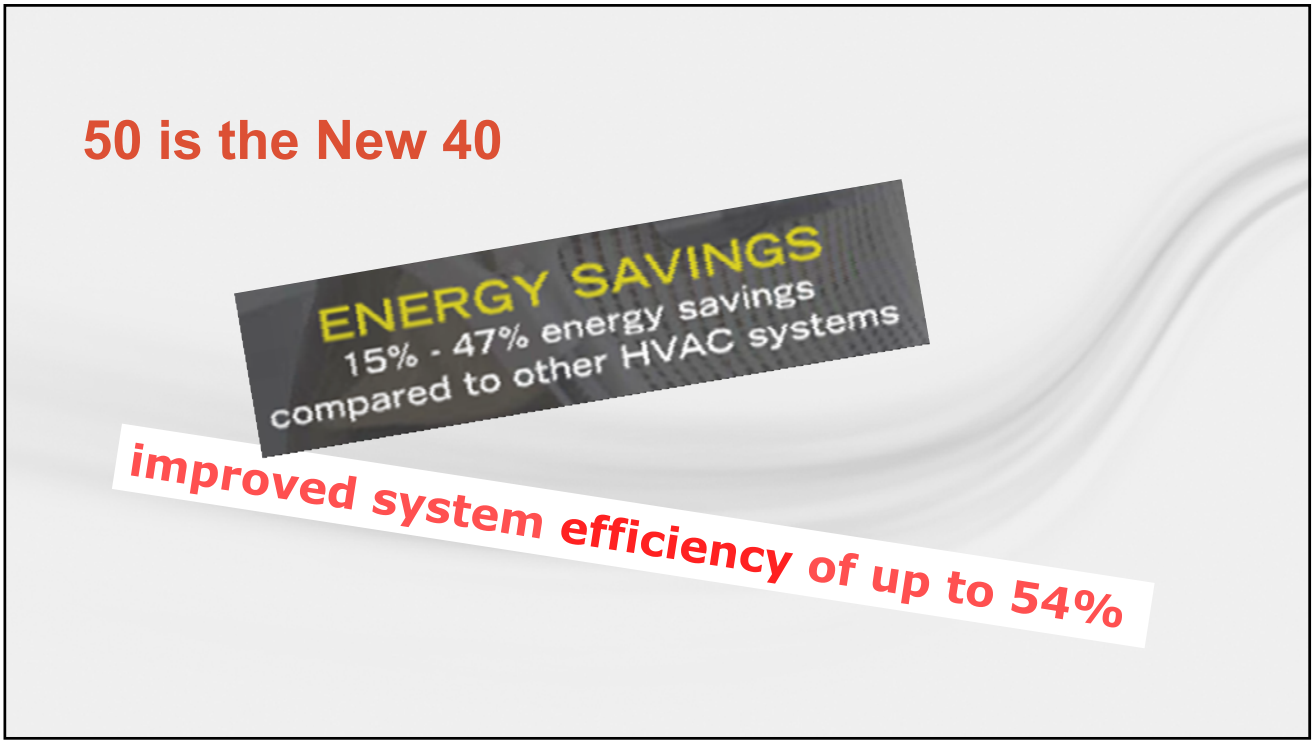
Can achieve up to 40% energy savings

CUT HVAC ENERGY COSTS UP TO 40%

ENERGY SAVINGS

Use up to 35% less fan power* and achieve higher ener… …vings.

EXCEEDS ASHRAE 90.1 STANDARD BY UP TO 44%

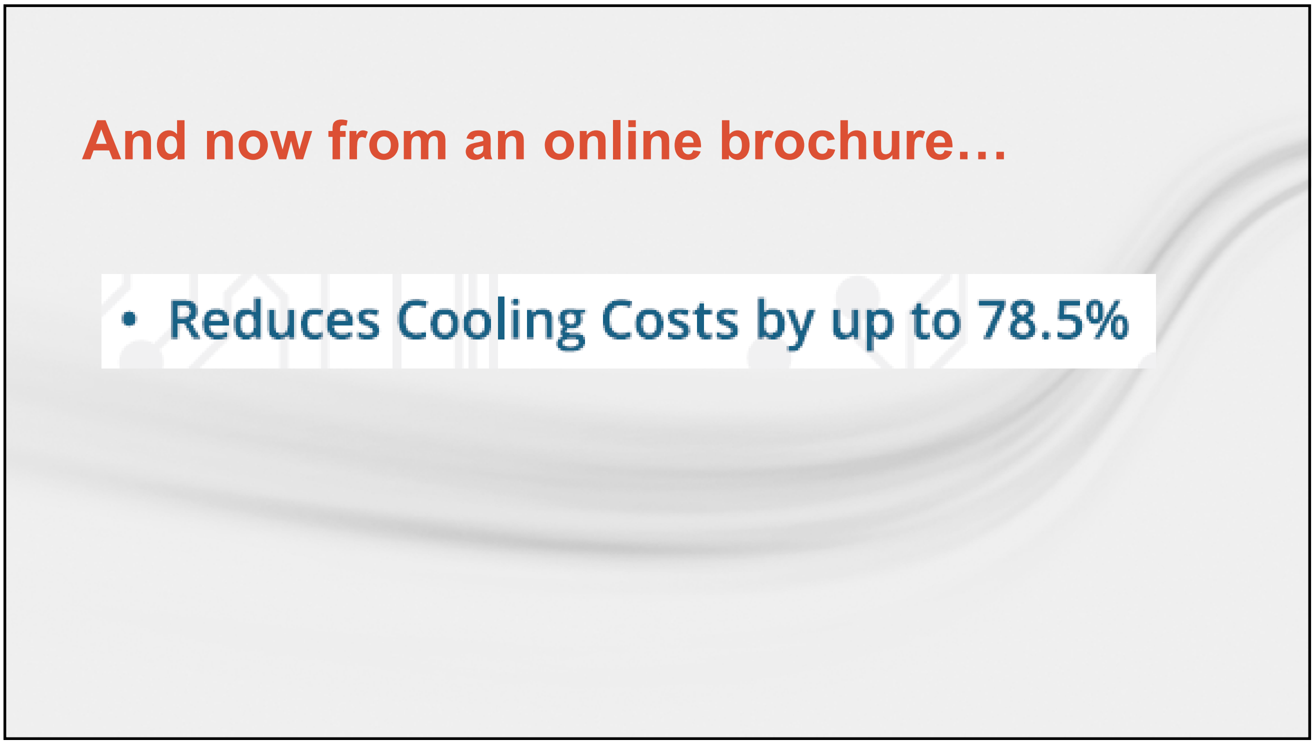


50 is the New 40

ENERGY SAVINGS

15% - 47% energy savings compared to other HVAC systems

improved system efficiency of up to 54%



And now from an online brochure…

- Reduces Cooling Costs by up to 78.5%

Percent Savings: Questions to Ask

- Compared to what? What is the baseline?
- What else changed? (particularly for retrofits).

Percent Savings: Questions to Ask

- Compared to what? What is the baseline?
- What else changed? (particularly for retrofits).
- Is the comparison valid for
  - Your building?
  - Your application and load profile?
  - Your climate?

78.5 Percent Claim

- Indirect evaporative cooling
- Compared to compressor cooling
- Dry climate
- Water is available

Does the solution meet my customers needs?

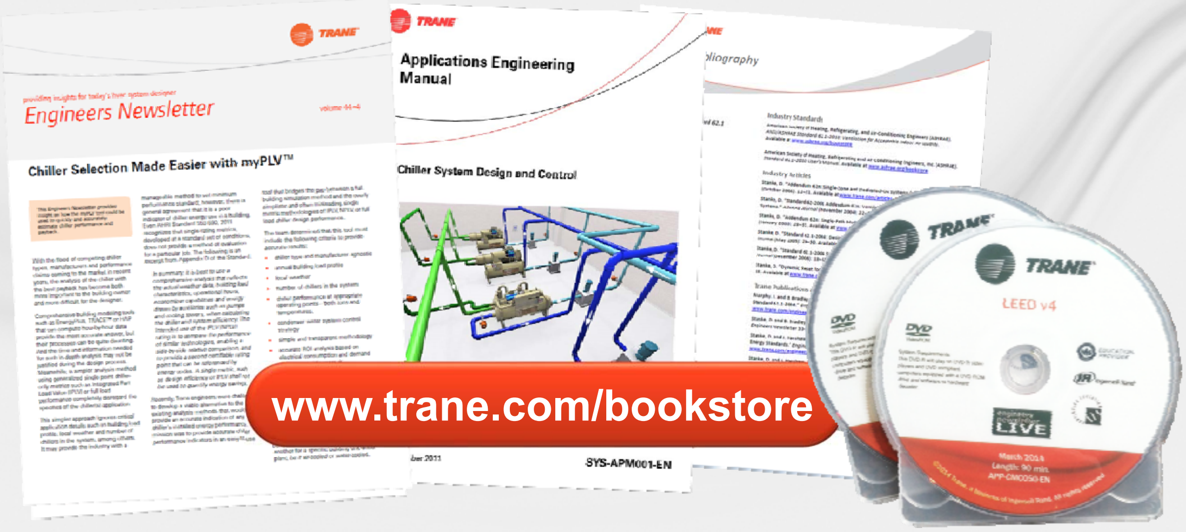
Myth Number 9

You can s
percent….
this…

- Make sure the baseline and comparison are valid for the specific project
- Perform an analysis on energy savings, energy cost savings, and ROI
- Help the client determine if the solution is in both their short-term and long-term interests

www.trane.com/bookstore

Where to Learn More



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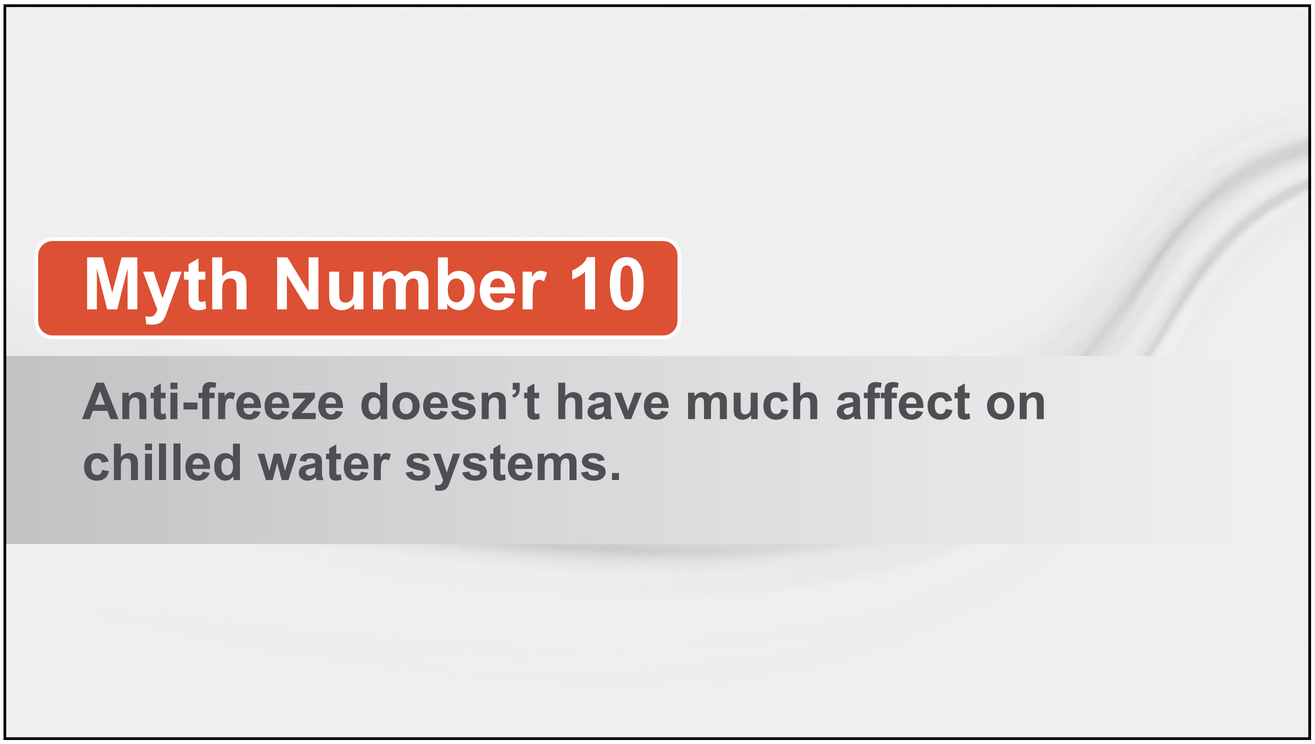
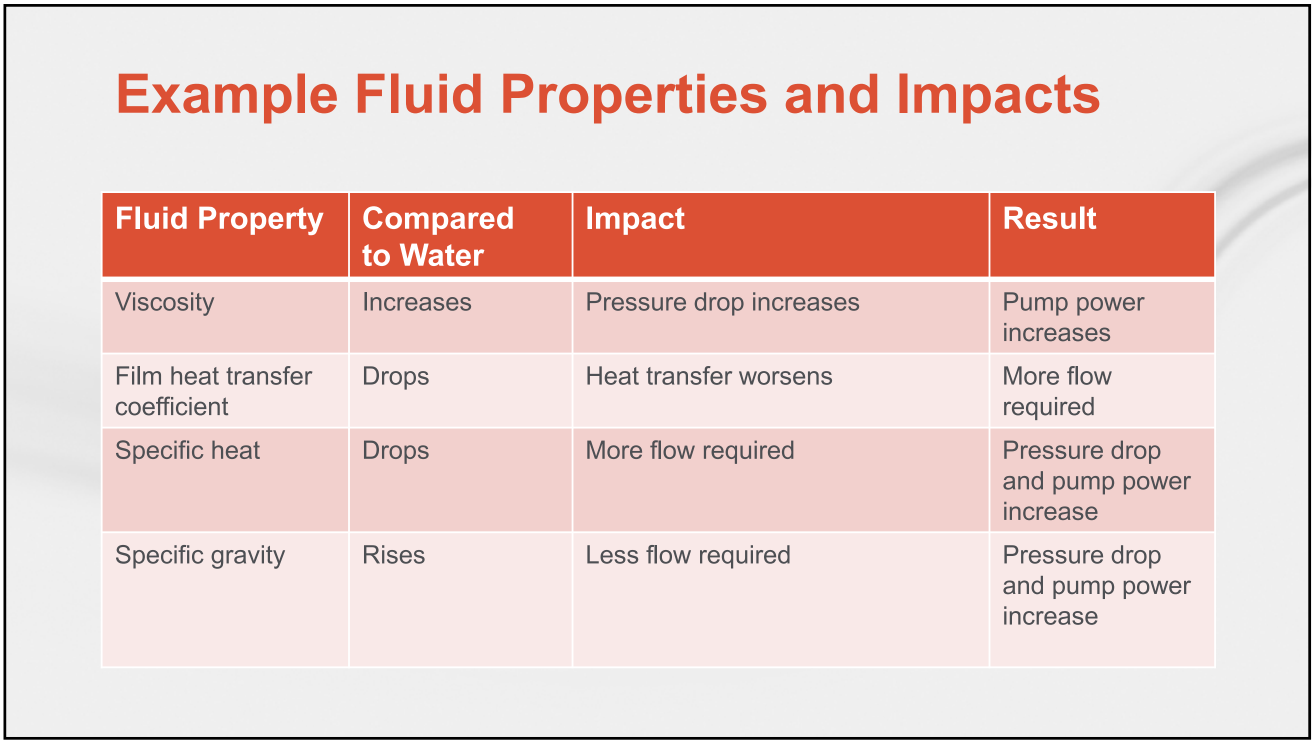


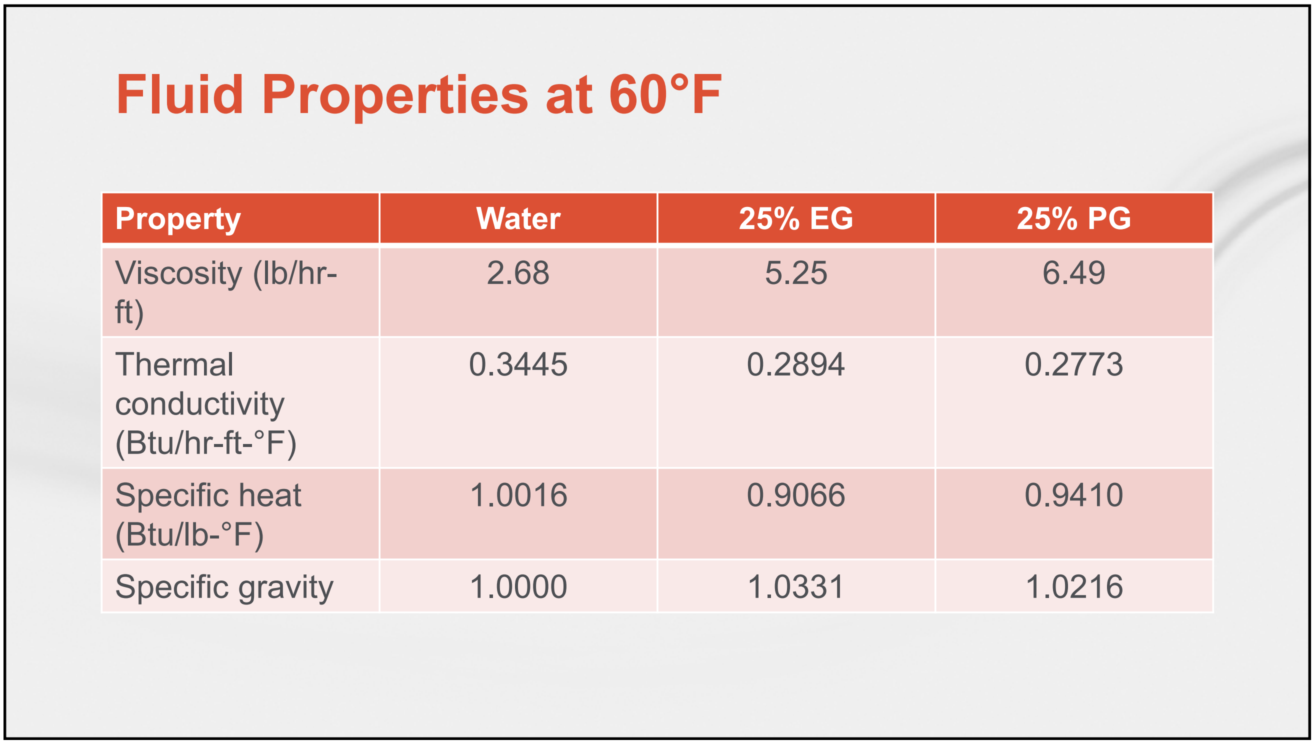
Myth Number 10

Anti-freeze doesn’t have much affect on chilled water systems.



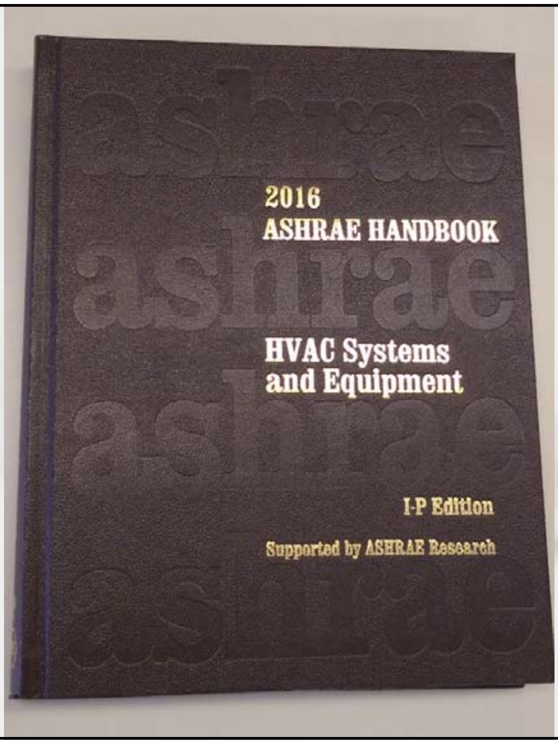
Example Fluid Properties and Impacts

Fluid Property	Compared to Water	Impact	Result
Viscosity	Increases	Pressure drop increases	Pump power increases
Film heat transfer coefficient	Drops	Heat transfer worsens	More flow required
Specific heat	Drops	More flow required	Pressure drop and pump power increase
Specific gravity	Rises	Less flow required	Pressure drop and pump power increase



Fluid Properties at 60°F

Property	Water	25% EG	25% PG
Viscosity (lb/hr-ft)	2.68	5.25	6.49
Thermal conductivity (Btu/hr-ft-°F)	0.3445	0.2894	0.2773
Specific heat (Btu/lb-°F)	1.0016	0.9066	0.9410
Specific gravity	1.0000	1.0331	1.0216



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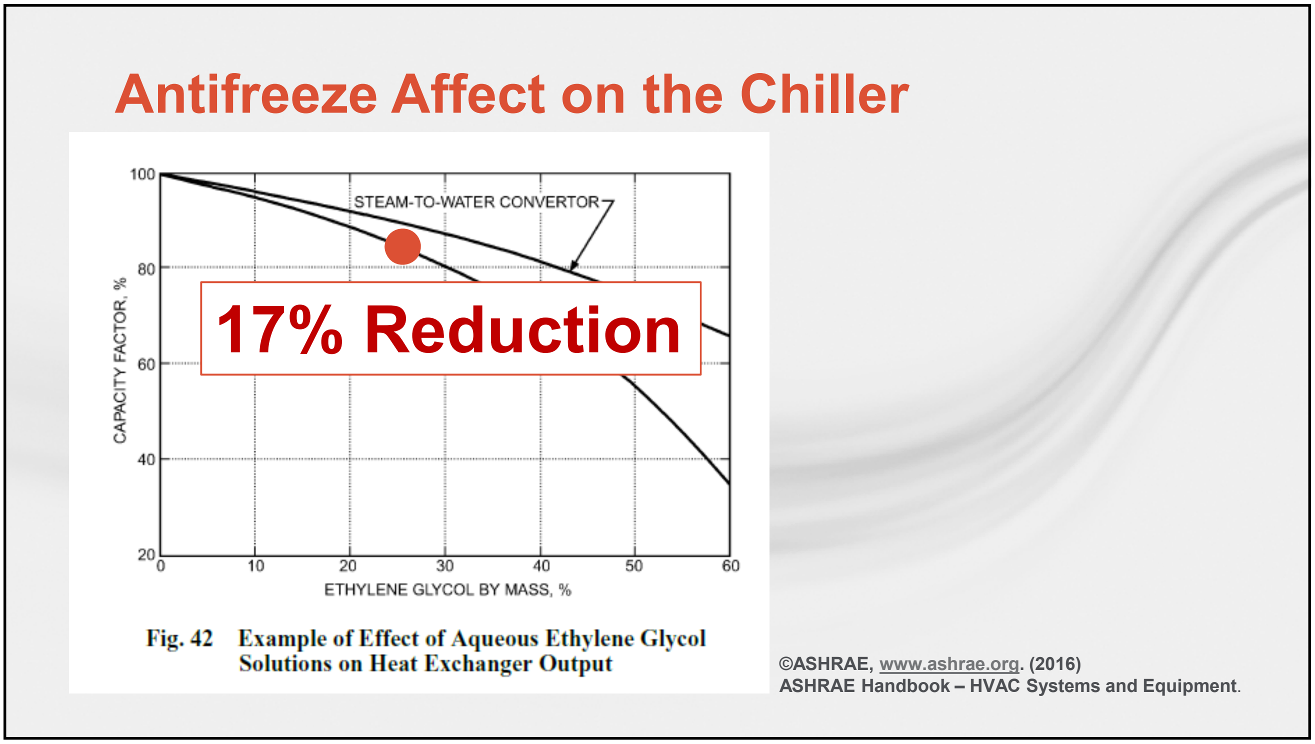
ASHRAE Handbook – HVAC Systems and Equipment.

Antifreeze Affect on the Coil

- "…heat transfer capability can change by 40% or more when antifreeze solutions are used…
- …consult the manufacturer's rating data…in glycol systems."



Antifreeze Affect on the Chiller

17% Reduction

Fig. 42 **Example of Effect of Aqueous Ethylene Glycol Solutions on Heat Exchanger Output**

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ASHRAE Handbook – HVAC Systems and Equipment.

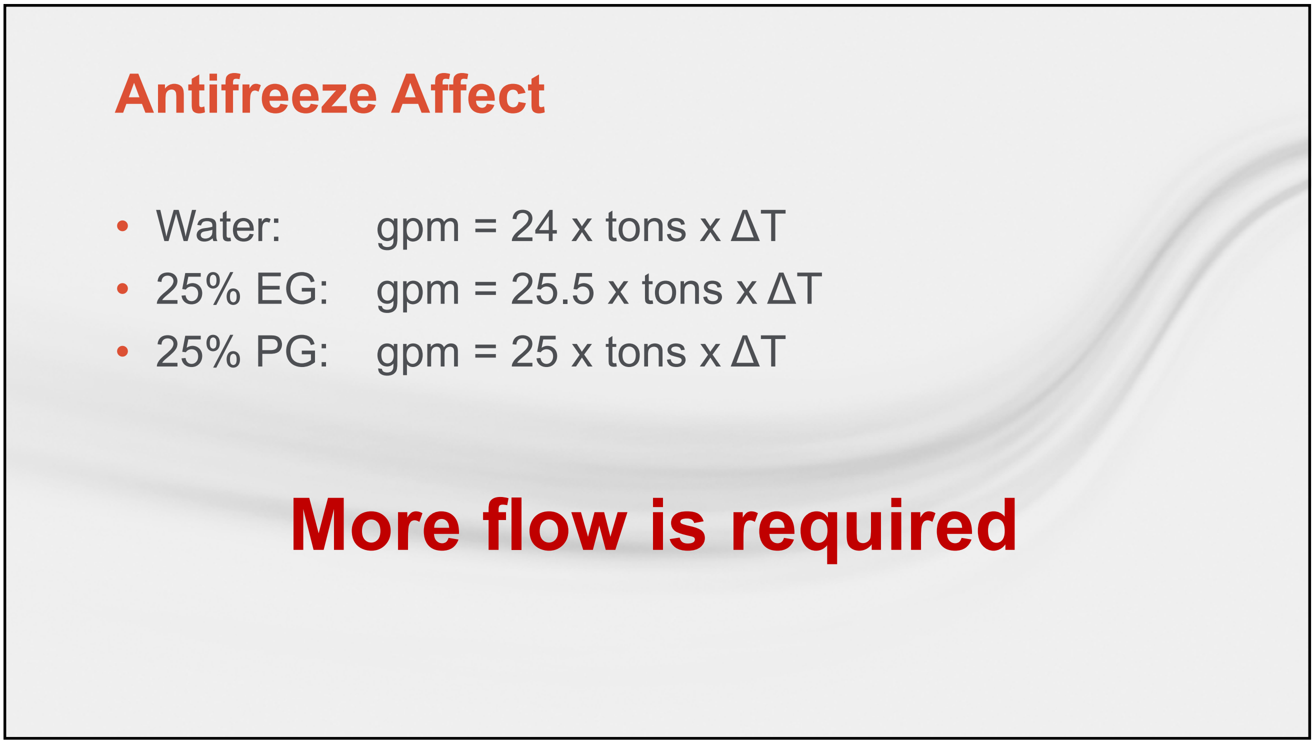
Required Flow

$$\text{Tons}=\frac{500\ \times \text{gpm}\ \times\ \Delta T\ \times \text{cp}\ \times \left(\frac{\rho}{\rho_w}\right)}{12{,}000}$$

$$\text{Tons(water)}=\frac{\text{gpm}\times\Delta T}{24}$$

$$\text{Tons(25\% EG)}=\frac{\text{gpm}\times\Delta T}{25.5}$$

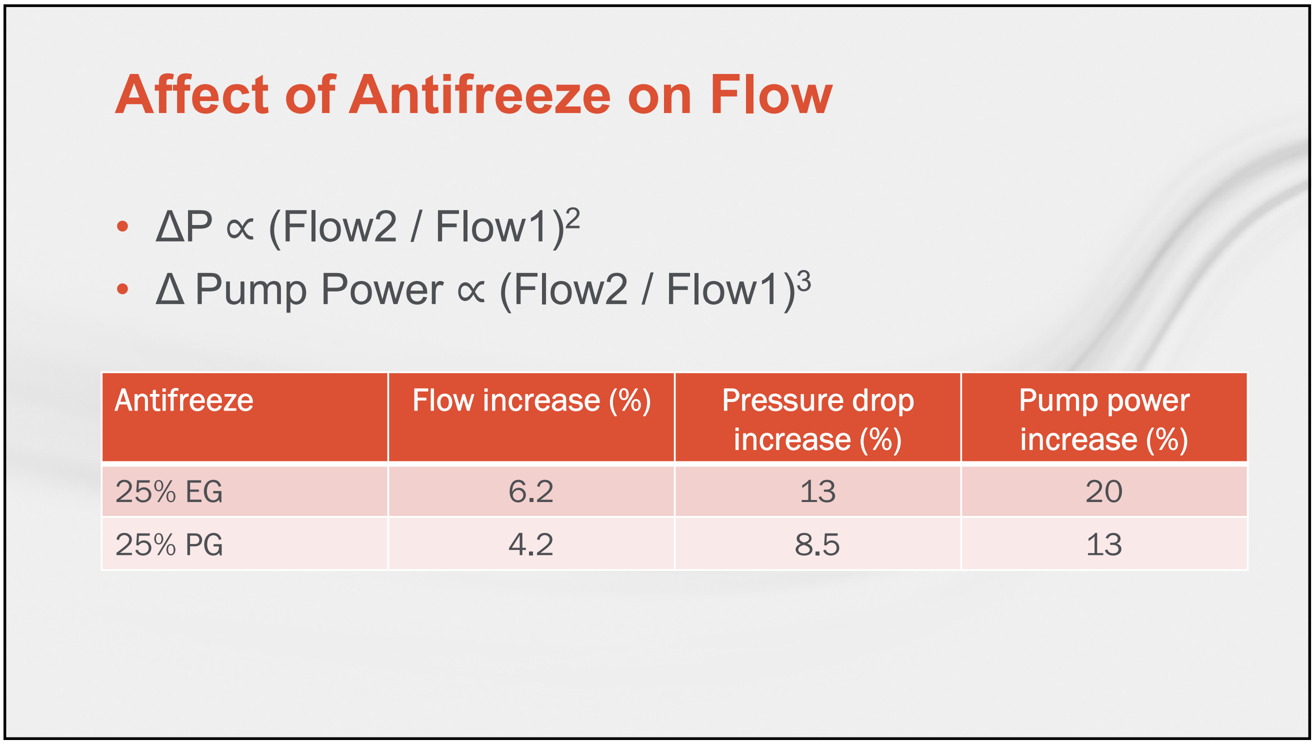
$$\text{Tons(25\% PG)}=\frac{\text{gpm}\times\Delta T}{25}$$



Antifreeze Affect

- Water: gpm = 24 x tons x ΔT
- 25% EG: gpm = 25.5 x tons x ΔT
- 25% PG: gpm = 25 x tons x ΔT

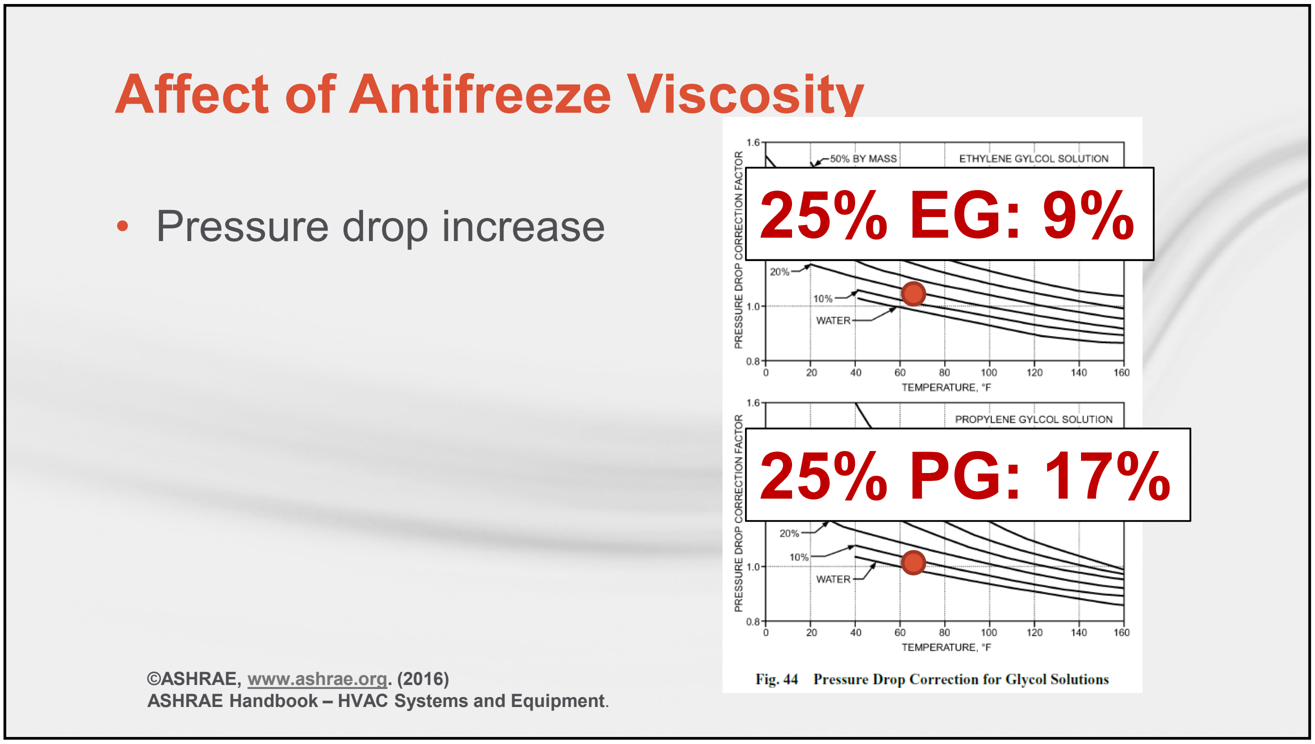
More flow is required



Affect of Antifreeze on Flow

- $\Delta P \propto (\text{Flow2} / \text{Flow1})^2$
- Δ Pump Power $\propto (\text{Flow2} / \text{Flow1})^3$

Antifreeze	Flow increase (%)	Pressure drop increase (%)	Pump power increase (%)
25% EG	6.2	13	20
25% PG	4.2	8.5	13



Affect of Antifreeze Viscosity

- Pressure drop increase

25% EG: 9%

25% PG: 17%

Fig. 44 Pressure Drop Correction for Glycol Solutions

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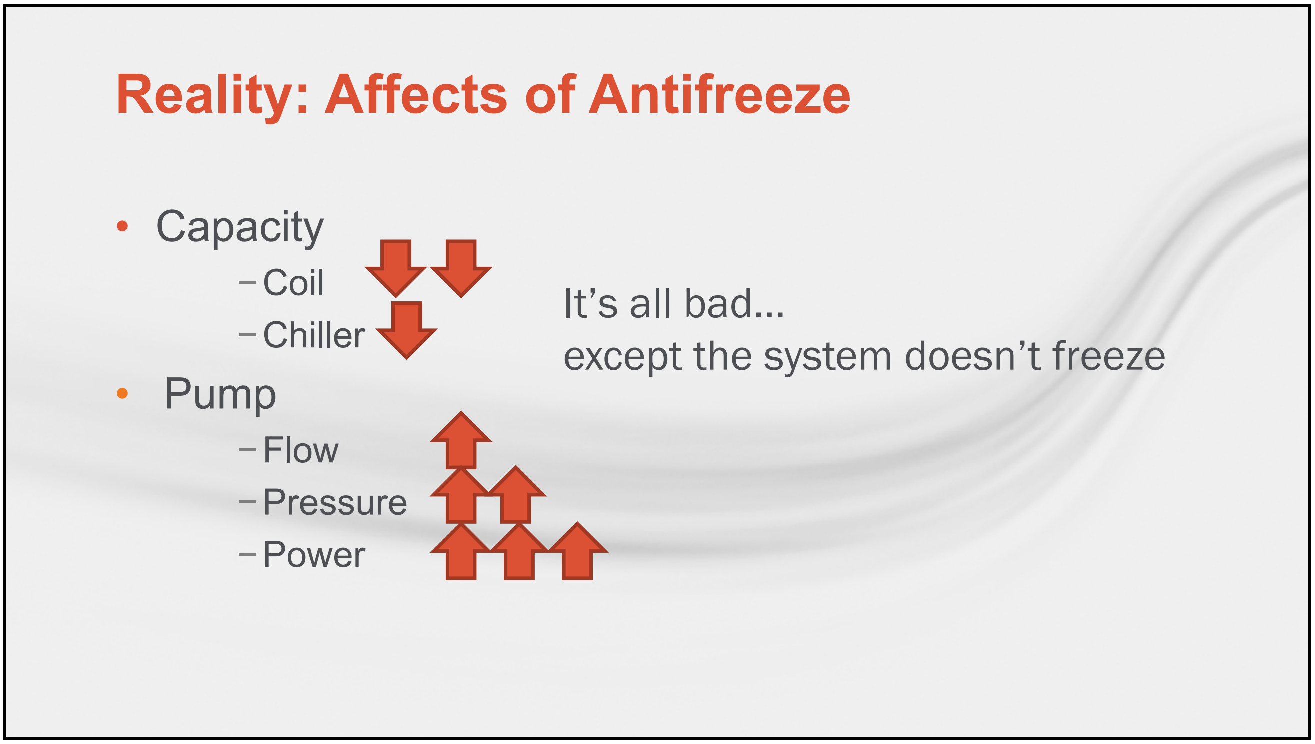

Reality: Affects of Antifreeze

- Capacity
  - Coil ⬇⬇
  - Chiller ⬇
- Pump
  - Flow ⬆
  - Pressure ⬆⬆
  - Power ⬆⬆⬆

It's all bad…
except the system doesn't freeze

Guidance

> *"…use the smallest possible concentration to produce the desired antifreeze properties."*

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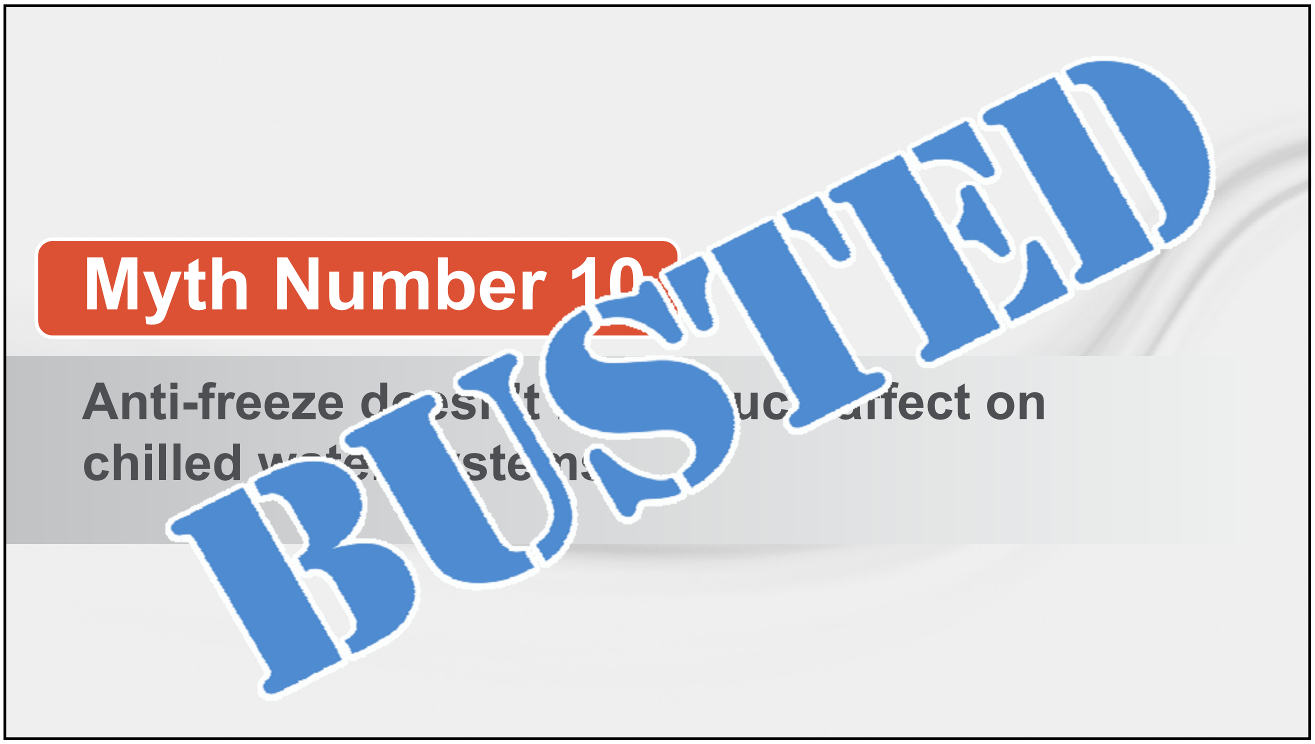
Freeze and Burst Protection

- Burst protection
  - Keep pipes from bursting
  - Crystal formation is ok
  - Use when equipment is not going to run in winter
- Freeze protection
  - Solution must remain 100% liquid
  - Necessary when equipment operates in freezing conditions

Freeze and Burst Protection

	Ethylene Glycol %		Propylene Glycol %	
Temperature (°F)	Freeze	Burst	Freeze	Burst
20	16.8	11.5	18	12
10	26.2	17.8	29	20
0	34.6	23.1	36	24
-10	40.9	27.3	42	28
-20	46.1	31.4	46	30
-30	50.3	31.4	50	33
-40	54.5	31.4	54	35



Myth Number 10

Anti-freeze doesn't have much affect on chilled water systems

Myth Number 11

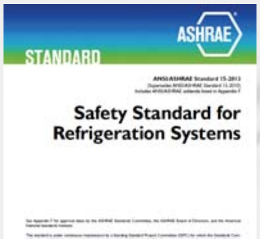
If refrigerant volume is too high for an occupied space to satisfy ASHRAE Standard 15 requirements, putting a refrigerant monitor in that occupied space meets the Standard 15 requirements.

NSPE Code of Ethics for Engineers

“Engineers, in the fulfillment of their professional duties, shall:

- *Hold paramount the safety, health, and welfare of the public…”*

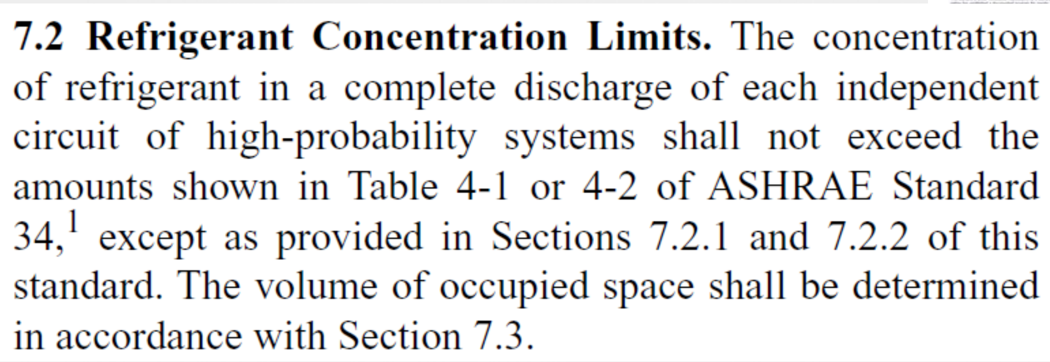
ASHRE Standard 15 - RCL


STANDARD

ASHRAE

ANSI/ASHRAE Standard 15-2013

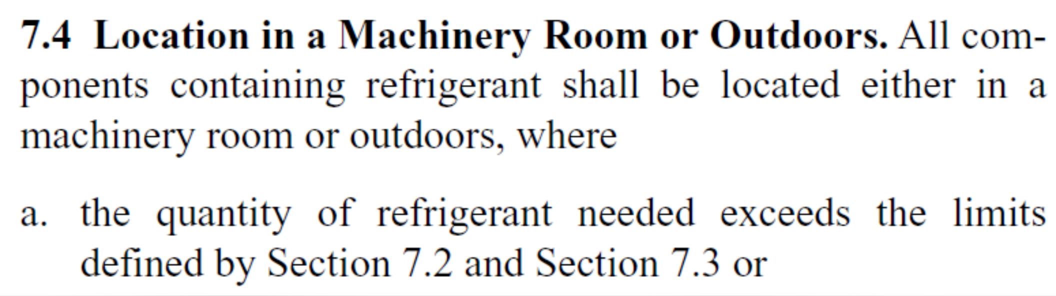
Safety Standard for
Refrigeration Systems


7.2 Refrigerant Concentration Limits. The concentration of refrigerant in a complete discharge of each independent circuit of high-probability systems shall not exceed the amounts shown in Table 4-1 or 4-2 of ASHRAE Standard 34,[1] except as provided in Sections 7.2.1 and 7.2.2 of this standard. The volume of occupied space shall be determined in accordance with Section 7.3.

Occupied Space Definition

“occupied space: that portion of the premises accessible to or occupied by people, excluding the machinery rooms.”

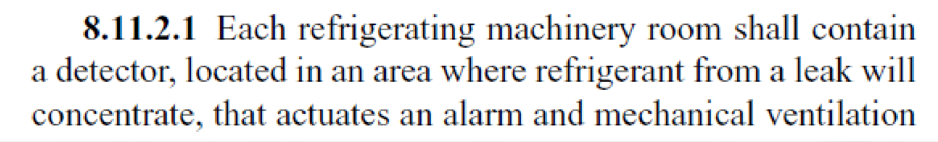
ASHRE Standard 15 – Machinery Room



7.4 Location in a Machinery Room or Outdoors. All components containing refrigerant shall be located either in a machinery room or outdoors, where

a. the quantity of refrigerant needed exceeds the limits defined by Section 7.2 and Section 7.3 or

ASHRAE Standard 15 – Leak Detection



8.11.2.1 Each refrigerating machinery room shall contain a detector, located in an area where refrigerant from a leak will concentrate, that actuates an alarm and mechanical ventilation

Occupied Space Definition

"occupied space: that portion of the premises accessible to or occupied by people,

***excluding the machinery rooms*.**"

Myth Number 11

If refrigerant volume is too much for the occupied space to satisfy ASHRAE Standard 15 requirements, putting a refrigerant monitor in the occupied space meets the Standard 15 requirements.

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May 2017

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HVAC Myths and Realities

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APP-CMC062-EN QUIZ

1. Which of the following are true?
    a. ASHRAE Standards 34 and 15 are sold together
    b. Standard 34 includes acceptable refrigerant concentration limits
    c. Standard 15 includes requirements for safe use of refrigerants
    d. All of the above

2. Which of the following are true about energy saving claims?
    a. All manufacturers stretch the truth
    b. Engineers should perform due diligence to determine in which applications and climates the savings are valid
    c. Since they are printed, they can be shared with the rest of the project team without further study
    d. Caveat emptor (let the buyer beware)

3. Which of the following will NOT cause coil low chilled water Delta T in a variable flow system
    a. Dirty filters in a constant volume air system.
    b. Lowering the leaving air setpoint in a VAV system 5°F below design.
    c. Colder than design temperature chilled water supplied to a coil.
    d. AHUs with 3-way control valves on the some coils.
    e. Unstable valve control.

4. Chillers with little flow turndown have no impact on system pumping energy.
    a. True
    b. False

5. Which systems types allow a dynamic flow device to most closely follow the affinity laws? (centrifugal: fan, pump or chiller)
    a. A system with a control valve for flow modulation.
    b. An open or closed system with only frictional losses.
    c. A system with its lift dependent on outside wetbulb temperature.
    d. A system with a fixed control setpoint (temperature or pressure).
    e. None of the above.
    f.

6. If the sensible load in the space is reduced, the relative humidity of the space will be _______ if the discharge air temperature isn't changed.
    a. higher
    b. lower
    c. remain the same

7. Oversizing a single-zone VAV system will result in improved dehumidification performance.
    a. True
    b. False

8. When selecting a fan, it is good practice to choose one where the operating point will fall to the right of peak pressure. Selecting a fan in this manner is important to: (choose all that apply)
    a. Avoid large fluctuations in airflow as the pressure changes
    b. Maximize the efficiency
    c. Avoid fan instability

9. Suppose an air-handling unit has a housed return fan. Which two values are commonly needed to calculate the total static pressure rise of the fan section?
    a. Fan section pressure plus adjustment
    b. Fan section pressure
    c. Downstream section pressure
    d. Downstream section pressure plus adjustment