HVAC Myths and Realities

Presenters: Systems and Applications Engineers Lee Cline, Dustin Meredith and Mick Schwedler with Jeanne Harshaw (host)
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
**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.
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www.USGBC.org

www.RCEP.net
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 = \( \frac{\Delta T \times \text{GPM}}{24} \)

Solving for gpm…

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

Pumping power…

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

Coil Delta T = 17°F
System Delta T = 8.5°F

CHWR = [(42°F x 50) + (59°F x 50)] / 100 = 50.5°F
**Reason 2: Supply Air Setpoint Depression**
overdriving coil capacity

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

- \(55^\circ\text{LAT} = 16^\circ\text{DT}\)
- \(52^\circ\text{LAT} = 11^\circ\text{DT}\)
- \(50^\circ\text{LAT} = 8.5^\circ\text{DT}\)

**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*

- \(42^\circ\text{CHWS} = 16^\circ\text{DT}\)
- \(47^\circ\text{CHWS} = 7.5^\circ\text{DT}\)
- \(50^\circ\text{CHWS} = 5^\circ\text{DT}\)
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

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?

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 1

Low Delta T is unavoidable
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)
Improving Dehumidification

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

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)

50°F DB
97% RH
52 gr/lb

50°F DB
97% RH
52 gr/lb

80°F DB
50% RH
77 gr/lb (60°F DP)

(10,000 cfm)

75°F DB
67% RH
87 gr/lb (64°F DP)

50°F DB
97% RH
52 gr/lb

8-12 rph

MA'

CA

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)

OB

humidity ratio, grains/lb of dry air

60
70
80
90
100
110

dry-bulb temperature, °F

20
30
40
50
60
70
80
90
100
110

humidity ratio, grains/lb of dry air

60
70
80
90
100
110

dry-bulb temperature, °F

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Myth Number 2

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

Myth Number 3

ASHRAE Standard 15 has to be updated before new refrigerants can be used.
Refrigerant safety groups from ANSI/ASHRAE Standard 34-2013

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 3

ASHRAE Standards have to be updated before new refrigerants can be used.
Myth Number 4

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

Classroom Example

basic CV system

<table>
<thead>
<tr>
<th>outdoor condition</th>
<th>96°F DB, 76°F WB</th>
</tr>
</thead>
<tbody>
<tr>
<td>sensible load</td>
<td>29,750 Btu/h</td>
</tr>
<tr>
<td>latent load</td>
<td>5,250 Btu/h</td>
</tr>
<tr>
<td>space SHR</td>
<td>0.85</td>
</tr>
<tr>
<td>supply airflow</td>
<td>1,500 cfm</td>
</tr>
<tr>
<td>outdoor airflow</td>
<td>450 cfm</td>
</tr>
<tr>
<td>space temp</td>
<td>74°F</td>
</tr>
<tr>
<td>supply air temp</td>
<td>55.7°F</td>
</tr>
</tbody>
</table>

Peak DB

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

Jacksonville, Florida
Example: K-12 Classroom

<table>
<thead>
<tr>
<th>CV full load</th>
<th>CV part load</th>
</tr>
</thead>
<tbody>
<tr>
<td>OA 96°F DBT</td>
<td>84°F DBT</td>
</tr>
<tr>
<td>68°F DPT</td>
<td>76°F DPT</td>
</tr>
<tr>
<td>(450 cfm)</td>
<td>(450 cfm)</td>
</tr>
</tbody>
</table>

| 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)   |

Jacksonville, FL
### Example: K-12 Classroom

<table>
<thead>
<tr>
<th>Constant-speed Fan</th>
<th>Peak DPT Day</th>
<th>Mild/rainy Day</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zone humidity, %RH</td>
<td>67%</td>
<td>73%</td>
</tr>
<tr>
<td>Cooling load, tons</td>
<td>3.7</td>
<td>1.6</td>
</tr>
<tr>
<td>Fan airflow, cfm</td>
<td>1500</td>
<td>1500</td>
</tr>
</tbody>
</table>

**Hot Gas Reheat**

- Packaged DX units

**Diagram:**
- Evaporator
- Condenser
- Reheat coil
- Reheat valve
- Condenser
- Evaporator
Example: K-12 Classroom

<table>
<thead>
<tr>
<th></th>
<th>constant-speed fan</th>
<th>constant-speed fan with hot gas reheat</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>peak DPT day</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>zone humidity, %RH</td>
<td>67%</td>
<td></td>
</tr>
<tr>
<td>cooling load, tons</td>
<td>3.7</td>
<td></td>
</tr>
<tr>
<td>fan airflow, cfm</td>
<td>1500</td>
<td></td>
</tr>
<tr>
<td><strong>mild/rainy day</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>zone humidity, %RH</td>
<td>73%</td>
<td>60%</td>
</tr>
<tr>
<td>cooling load, tons</td>
<td>1.6</td>
<td>2.4</td>
</tr>
<tr>
<td>fan airflow, cfm</td>
<td>1500</td>
<td>1500</td>
</tr>
<tr>
<td>compressor energy</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Space humidity is maintained

Improved Part-Load Dehumidification

- **OA** 450 cfm
- **EA** 1050 cfm
- **RA** 1500 cfm
- **SA** 96°F DBT 68°F DPT
- Zone 74°F
Example: K-12 Classroom

<table>
<thead>
<tr>
<th></th>
<th>peak DPT day</th>
<th>mild/rainy day</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>constant-speed fan</td>
<td>constant-speed fan with hot gas reheat</td>
</tr>
<tr>
<td>zone humidity, %RH</td>
<td>67%</td>
<td>73%</td>
</tr>
<tr>
<td>cooling load, tons</td>
<td>3.7</td>
<td>1.6</td>
</tr>
<tr>
<td>fan airflow, cfm</td>
<td>1500</td>
<td>1500</td>
</tr>
</tbody>
</table>

Example: K-12 Classroom

<table>
<thead>
<tr>
<th></th>
<th>constant-speed fan</th>
<th>constant-speed fan with hot gas reheat</th>
<th>variable-speed fan</th>
</tr>
</thead>
<tbody>
<tr>
<td>OA</td>
<td>96°F DBT 84°F DBT</td>
<td>96°F DBT 84°F DBT</td>
<td>96°F DBT 84°F DBT</td>
</tr>
<tr>
<td>full load</td>
<td>(450 cfm)</td>
<td>(450 cfm)</td>
<td>(450 cfm)</td>
</tr>
<tr>
<td>part load</td>
<td>68°F DPT 76°F DPT</td>
<td>68°F DPT 76°F DPT</td>
<td>68°F DPT 76°F DPT</td>
</tr>
<tr>
<td>(1500 cfm)</td>
<td>(450 cfm)</td>
<td>(450 cfm)</td>
<td>(450 cfm)</td>
</tr>
<tr>
<td>RA</td>
<td>74°F DBT 74°F DBT</td>
<td>74°F DBT 74°F DBT</td>
<td>74°F DBT 74°F DBT</td>
</tr>
<tr>
<td>52% RH</td>
<td>67% RH</td>
<td>57% RH</td>
<td>67% RH</td>
</tr>
<tr>
<td>MA</td>
<td>81°F DBT 77°F DBT</td>
<td>81°F DBT 77°F DBT</td>
<td>81°F DBT 77°F DBT</td>
</tr>
<tr>
<td>SA</td>
<td>55°F DBT 63°F DBT</td>
<td>55°F DBT 63°F DBT</td>
<td>55°F DBT 63°F DBT</td>
</tr>
<tr>
<td>(1500 cfm)</td>
<td>(1500 cfm)</td>
<td>(1500 cfm)</td>
<td>(1500 cfm)</td>
</tr>
<tr>
<td>(4.8 tons)</td>
<td>(3.7 tons)</td>
<td>(4.0 tons)</td>
<td>(3.7 tons)</td>
</tr>
</tbody>
</table>
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

<table>
<thead>
<tr>
<th></th>
<th>SZVAV full load</th>
<th>SZVAV Oversized</th>
<th>SZVAV part load</th>
</tr>
</thead>
<tbody>
<tr>
<td>OA</td>
<td>96°F DBT 68°F DPT (450 cfm)</td>
<td>96°F DBT 68°F DPT (450 cfm)</td>
<td>84°F DBT 76°F DPT (450 cfm)</td>
</tr>
<tr>
<td>RA</td>
<td>74°F DBT 52% RH (450 cfm)</td>
<td>74°F DBT 56% RH (450 cfm)</td>
<td>74°F DBT 59% RH (450 cfm)</td>
</tr>
<tr>
<td>MA</td>
<td>81°F DBT 74°F DBT (450 cfm)</td>
<td>80°F DBT 74°F DBT (450 cfm)</td>
<td>78°F DBT 58°F DBT (450 cfm)</td>
</tr>
<tr>
<td>SA</td>
<td>55°F DBT 55°F DBT (1500 cfm)</td>
<td>58°F DBT 58°F DBT (1750 cfm)</td>
<td>52°F DBT 52°F DBT (1050 cfm)</td>
</tr>
</tbody>
</table>
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
Single-zone VAV units do not need hot gas reheat

Myth Number 5
Slap on a VFD and you are entitled to get full advantage of the affinity laws.

= Speed³ 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

![Diagram showing the relationship between speed, pressure, flow, and power](image)
Systems and the Affinity Laws

compliant systems

Systems that comply
- Cooling towers
- Single zone VAV air systems.

Cooling Tower Fans

affinity laws

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Trane Engineers Newsletter LIVE: HVAC Myths and Realities

APP-CMC062-EN
Cooling Tower Fans

affinity laws

For “Free Discharge” Fans

\[ W_2 = W_1 \times \left( \frac{S_2}{S_1} \right)^3 \]

\[ W_2 = 15 \text{ kW} \times \left( \frac{30}{60} \right)^3 \]

\[ W_2 = 1.88 \text{ kW} \]

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.**

<table>
<thead>
<tr>
<th>Systems that comply</th>
<th>Systems that don’t comply</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Cooling towers</td>
<td>- Chilled water</td>
</tr>
<tr>
<td>- Single zone VAV HVAC systems</td>
<td>- Condenser water</td>
</tr>
<tr>
<td></td>
<td>- Multi-zone VAV</td>
</tr>
<tr>
<td></td>
<td>- HVAC cooling units</td>
</tr>
<tr>
<td></td>
<td>- HVAC heating units (HP)</td>
</tr>
</tbody>
</table>

### 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

Fan Performance Test
Pressure = f(Airflow)

Forward Curved (FC) Fan

Two Different Systems

Forward Curved (FC) Fan
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

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.

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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%

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 ΔTs
  - ASHRAE GreenGuide (16-18°F ΔT)
  - 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

\[ \Delta T = 1 \text{ or } 2 \, ^\circ\text{F} \]

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.

BUSTED

Myth Number 8

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


Fan Rotation

Forward-curved

Backward-tended

Backward-inclined

Backward-curved

Airfoil
Low Airflow Troubleshooting

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

Uncommon problems:
• Wrong fan installed
• Cutoff issues (housed fans only)
• Software/catalog error
• Quantum mechanics & string theory
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

Fans often don’t deliver the airflow they are supposed to — 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%.
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).

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 save (20, 30, 40, 50, 80) percent….just by doing 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
LEED Continuing Education Courses

on-demand, no charge, up to 1.5 CE credits

• Variable-Speed Compressors on Chillers
• Coil Selection and Optimization
• Specifying Quality Sound
• ASHRAE Standard 62.1-2010
• ASHRAE Standard 90.1-2010
• High-Performance VAV Systems
• Single-Zone VAV Systems
• Ice Storage Design and Control
• All Variable-Speed Chiller Plant Operation
plan to attend...
2017 Engineers Newsletter Live!

Energy-Saving Strategies for Small Rooftop Systems

HVAC Myths and Realities

High-Performance Air Systems

Demand Response in Commercial Buildings

Please contact your local Trane office for event details and registration or visit trane.com/ENL.

If you need a lot of HVAC insight but have limited time...

RSVP Today!
Myth Number 10

Anti-freeze doesn’t have much affect on chilled water systems.
Example Fluid Properties and Impacts

<table>
<thead>
<tr>
<th>Fluid Property</th>
<th>Compared to Water</th>
<th>Impact</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity</td>
<td>Increases</td>
<td>Pressure drop increases</td>
<td>Pump power increases</td>
</tr>
<tr>
<td>Film heat transfer coefficient</td>
<td>Drops</td>
<td>Heat transfer worsens</td>
<td>More flow required</td>
</tr>
<tr>
<td>Specific heat</td>
<td>Drops</td>
<td>More flow required</td>
<td>Pressure drop and pump power increase</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>Rises</td>
<td>Less flow required</td>
<td>Pressure drop and pump power increase</td>
</tr>
</tbody>
</table>

Fluid Properties at 60°F

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

Required Flow

\[
\text{Tons} = \frac{500 \times \text{gpm} \times \Delta T \times \text{cp} \times (\frac{\rho}{12,000})}{24}
\]

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

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

\[
\text{Tons(25\% PG)} = \frac{\text{gpm} \times \Delta T}{25}
\]
Antifreeze Affect

- Water: \( \text{gpm} = 24 \times \text{tons} \times \Delta T \)
- 25% EG: \( \text{gpm} = 25.5 \times \text{tons} \times \Delta T \)
- 25% PG: \( \text{gpm} = 25 \times \text{tons} \times \Delta T \)

More flow is required

Affect of Antifreeze on Flow

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

<table>
<thead>
<tr>
<th>Antifreeze</th>
<th>Flow increase (%)</th>
<th>Pressure drop increase (%)</th>
<th>Pump power increase (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25% EG</td>
<td>6.2</td>
<td>13</td>
<td>20</td>
</tr>
<tr>
<td>25% PG</td>
<td>4.2</td>
<td>8.5</td>
<td>13</td>
</tr>
</tbody>
</table>
Affect of Antifreeze Viscosity

- Pressure drop increase

25% EG: 9%
25% PG: 17%

Reality: Affects of Antifreeze

- Capacity
  - Coil
  - Chiller
  - 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.”

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

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Ethylene Glycol %</th>
<th>Propylene Glycol %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Freeze</td>
<td>Burst</td>
</tr>
<tr>
<td>20</td>
<td>16.8</td>
<td>11.5</td>
</tr>
<tr>
<td>10</td>
<td>26.2</td>
<td>17.8</td>
</tr>
<tr>
<td>0</td>
<td>34.6</td>
<td>23.1</td>
</tr>
<tr>
<td>-10</td>
<td>40.9</td>
<td>27.3</td>
</tr>
<tr>
<td>-20</td>
<td>46.1</td>
<td>31.4</td>
</tr>
<tr>
<td>-30</td>
<td>50.3</td>
<td>31.4</td>
</tr>
<tr>
<td>-40</td>
<td>54.5</td>
<td>31.4</td>
</tr>
</tbody>
</table>

### Myth Number 10

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

**BUSTED**
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

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 high for an occupied space to satisfy ASHRAE Standard 15 requirements, putting a refrigerant monitor in the occupied space meets the Standard’s requirements.

BUSTED
Industry Resources


Articles


Engineers Newsletter Live - Audience Evaluation

HVAC Myths and Realities

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Rate the pace of the program.  Appropriate ☐ Too fast ☐ Too slow

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Are there any other events/topics you would like Trane to offer to provide additional knowledge of their products and services?

Additional questions or comments:
Trane Engineers Newsletter LIVE: HVAC Myths and Realities

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. None

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