Agenda and Objectives

Trane Engineers Newsletter Live Series

Fans in Air-Handling Systems

Fans used in air-handling systems often have a significant impact on energy use and acoustics. How much of an impact depends on how a fan is selected, installed, and operated. Trane applications engineers discuss fan performance curves and fan laws, different fan types (fan blade shape, housed vs. plenum fans, direct-drive plenum fans, fan arrays), how a fan interacts with various types of systems, considerations when selecting a fan (efficiency, acoustics, footprint), and the ASHRAE Standard 90.1 fan power limitations.

By attending this event you will be able to:
1. Summarize how to use fan performance curves
2. How to select, install, and operate fans to avoid problems related to inadequate airflow or excessive noise
3. When to select the different fan types (housed vs. plenum, belt-drive vs. direct-drive, single fan vs. a fan array)
4. How to design fan systems to meet the prescriptive requirements of ASHRAE Standard 90.1-2007

Agenda
1) Fan performance curves
    a) How developed (lab setup, difference with AHU vs. RTU)
    b) What they are for (selection) and not for (predicting field performance)
    c) Fan laws
    d) Interaction of fans in a system (system curve)
2) Fan/unit selection considerations
    a) Types of fans (energy – bhp or motor input kW, acoustics, footprint, maintenance, redundancy)
    b) Impact of system configuration on fan selection (examples)
    c) System effect (example using AMCA guide)
    d) Acoustics topics
3) Common problems
    a) Fan is not delivering enough airflow
    b) Fan is making too much noise
4) Meeting ASHRAE 90.1 requirements
    a) Option 1 vs. Option 2 (fan power limitation)
    b) Lowering bhp/cfm
5) Summary

Presenters

Trane Engineers Newsletter Live Series

Fans in Air-Handling Systems

(2010)

Dave Guckelberger | application engineer | Trane

Dave has a wide range of product and system responsibilities as a Trane applications engineer. His expertise includes acoustic analysis and modeling of HVAC systems, electrical distribution system design, and the equipment-room design requirements established by ASHRAE Standard 15. He also provides research and interpretation on how building, mechanical, and fire codes impact HVAC equipment and systems.

In addition to traditional applications engineering support, Dave has authored a variety of technical articles on subjects ranging from acoustics to ECM motors to codes. After graduating from Michigan Tech with a BSME in thermo-fluids, he joined Trane as a development engineer in 1982 and moved into his current position in Applications Engineering in 1987.

Dustin Meredith | application engineer | Trane

Dustin is an application engineer with expertise in sound predictions, fan selection, and vibration analysis. He also leads development and implementationprojects for new and upcoming air-handling options. Dustin has authored various technical engineering bulletins and applications engineering manuals.

Dustin is a corresponding member on ASHRAE TC 2.6 – Sound & Vibration Control – and ASHRAE TC5.1 – Fans. After graduating from the University of Kentucky with BSME, BSCS and MBA degrees, hejoined Trane as a marketing engineer in 2000 and moved into his current position in Application Engineering in 2005. Dustin is a member of ASHRAE and is the primary Trane contact for AMCA.

John Murphy | application 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, air-to-air energy recovery, psychrometry, ventilation, and ASHRAE Standards 15, 62.1, and 90.1.

John is the author of numerous Trane application manuals, Engineers Newsletters and ASHRAE Journal articles. He is a member of ASHRAE, and has served on ASHRAE's "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, and technical reviewer for The ASHRAE Guide for Buildings in Hot and Humid Climates.

Dennis Stanke | staff application engineer | Trane

With a BSME from the University of Wisconsin, Dennis joined Trane in 1973, as a controls development engineer. He is now a Staff Applications Engineer specializing in airside systems including controls, ventilation, indoor air quality, and dehumidification. He has written numerous applications manuals, newsletters, and technical articles and columns.

An ASHRAE Fellow, he recently served as Chairman for SSPC62.1, the ASHRAE committee responsible for Standard 62.1, "Ventilation for Acceptable Indoor Air Quality," and he serves on the USGBC LEED Technical Advisory Group for Indoor Environmental Quality (the LEED EQ TAG).

Fans in Air-Handling Systems

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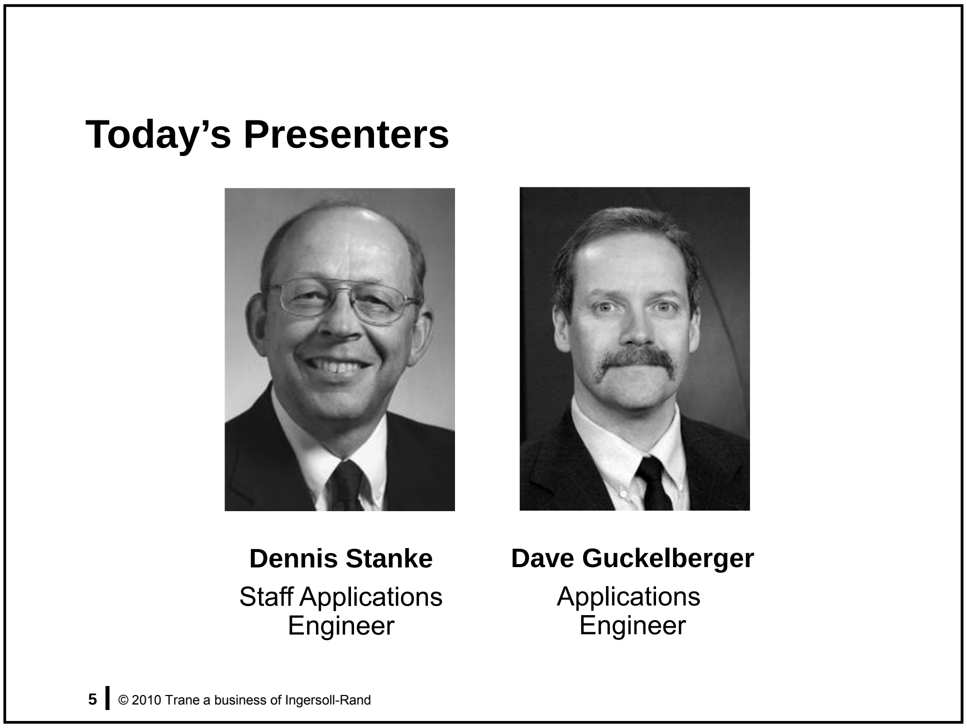
Fans in Air-Handling Systems

Today’s Topics

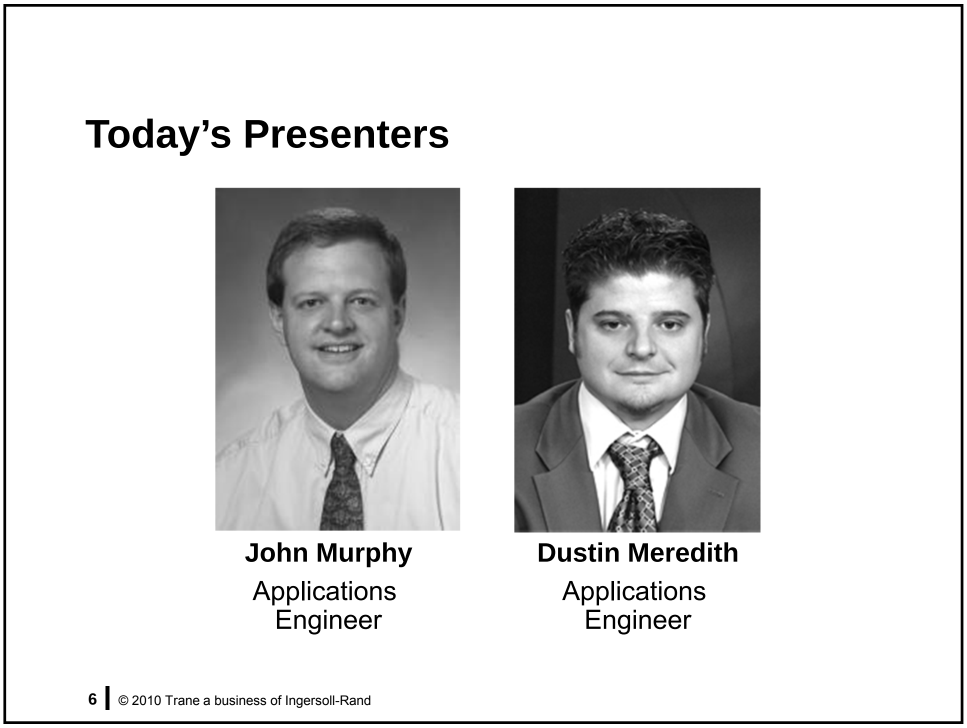
- Fan fundamentals
  - Performance curves
- Fan/unit selection considerations
  - Fan types
  - Impact of system configuration
  - System effect
  - Acoustics
- Common problems
- ASHRAE 90.1 requirements



Today's Presenters

Dennis Stanke
Staff Applications Engineer

Dave Guckelberger
Applications Engineer



Today's Presenters

John Murphy
Applications Engineer

Dustin Meredith
Applications Engineer

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Fundamentals

Fan Performance Curves

AMCA 210/ASHRAE 51

“Laboratory Methods of Testing Fans for Aerodynamic Performance Rating”

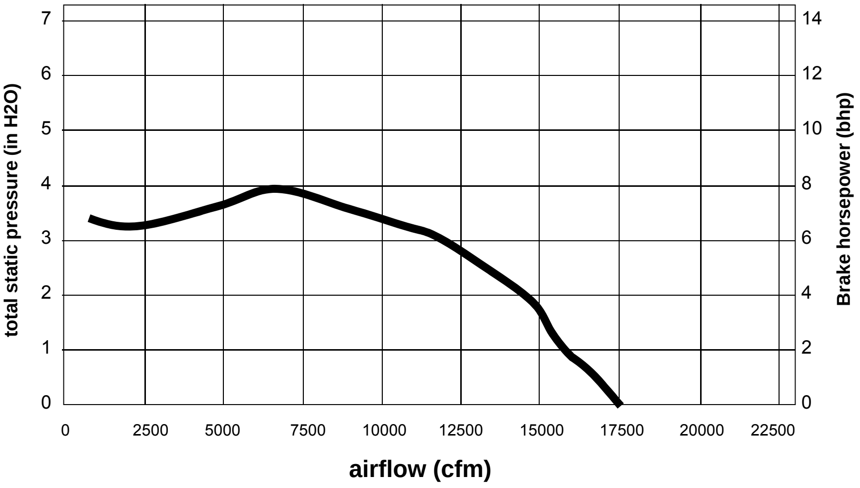
Static Pressure: … that portion of the air pressure which exists by virtue of the degree of compression only.

Velocity Pressure: … that portion of the air pressure which exists by virtue of the rate of motion only.

Total Pressure: … the algebraic sum of the velocity pressure and the static pressure at a point.

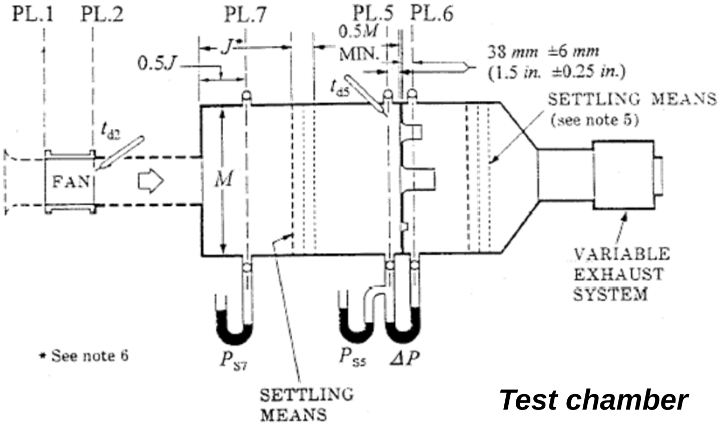
$$P_t = P_v + P_s$$

Fan Curves

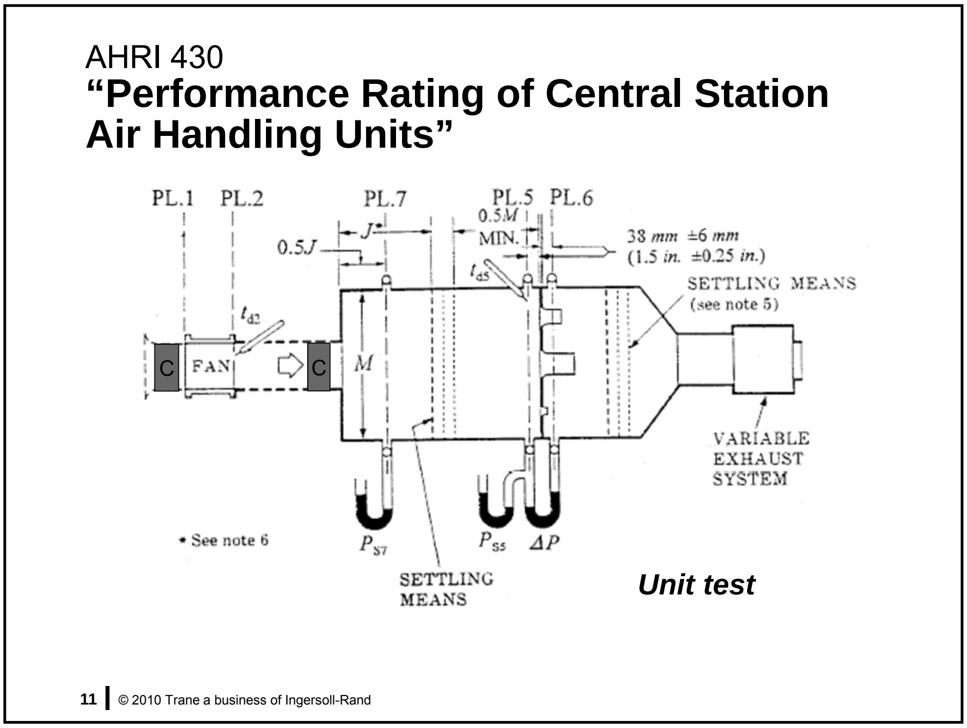


ANSI/ASHRAE 51(AMCA 210-07), Figure 12

“Outlet Chamber Setup-Multiple Nozzles in Chamber”



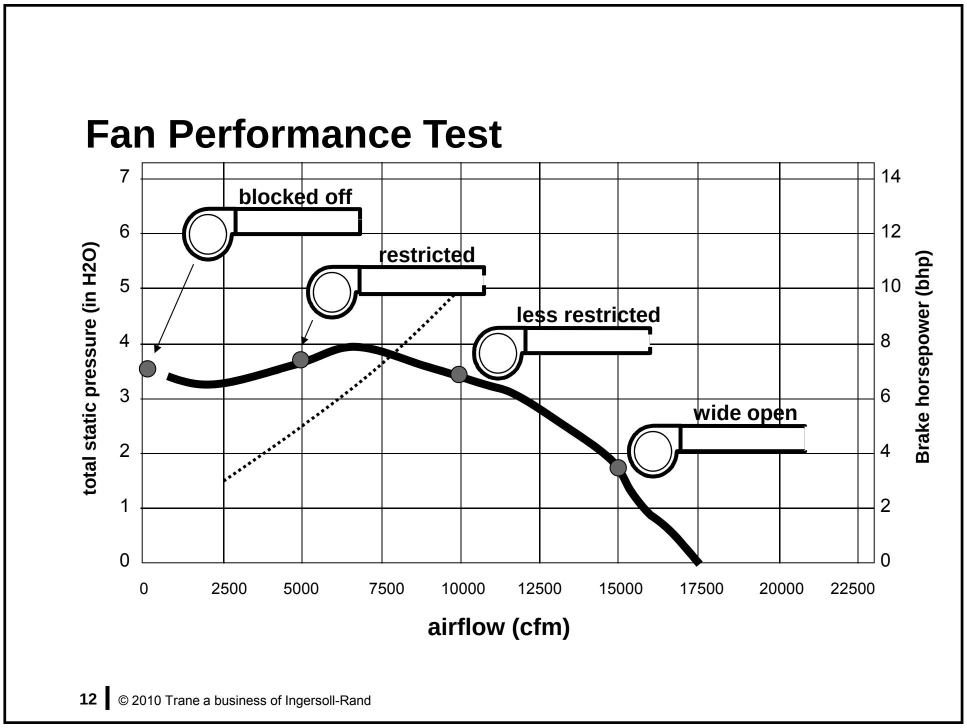
Test chamber



AHRI 430

“Performance Rating of Central Station Air Handling Units”

Unit test

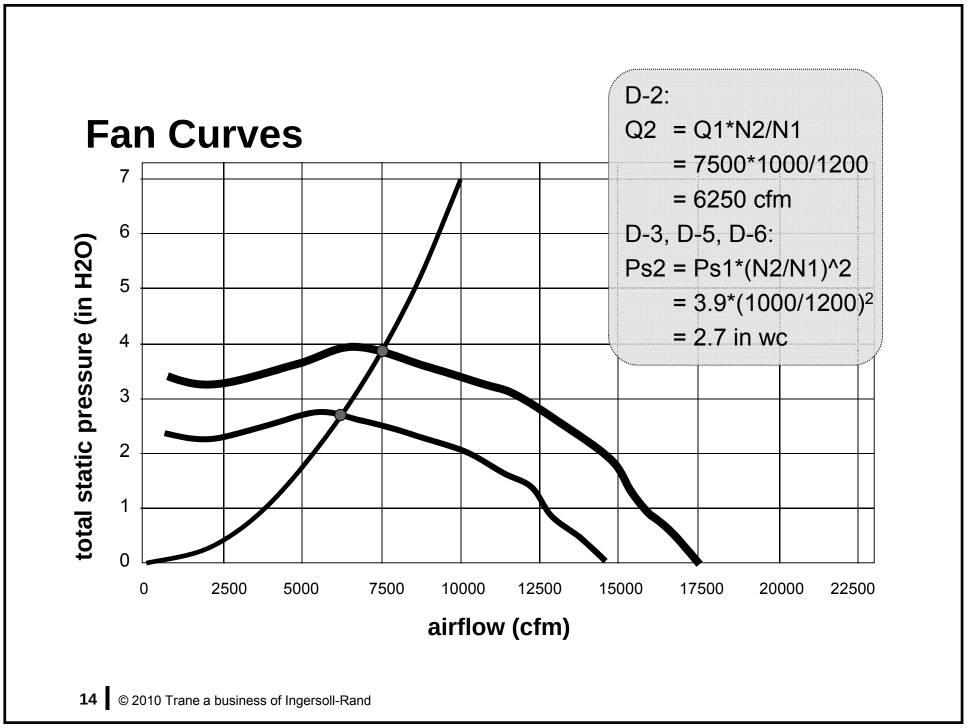


Fan Performance Test

Fan Laws for Incompressible Flow

- Fan total efficiency (D-1) $\eta_{t2} = \eta_{t1}$
- Airflow rate (D-2) $Q_2 = Q_1 \cdot \left(\frac{D_2}{D_1}\right)^3 \cdot \left(\frac{N_2}{N_1}\right)$
- Fan total pressure (D-3) $P_{t2} = P_{t1} \cdot \left(\frac{D_2}{D_1}\right)^2 \cdot \left(\frac{N_2}{N_1}\right)^2 \cdot \left(\frac{\rho_2}{\rho_1}\right)$
- Fan power input (D-4) $H_2 = H_1 \cdot \left(\frac{D_2}{D_1}\right)^5 \cdot \left(\frac{N_2}{N_1}\right)^3 \cdot \left(\frac{\rho_2}{\rho_1}\right)$
- Fan velocity pressure (D-5) $P_{v2} = P_{v1} \cdot \left(\frac{D_2}{D_1}\right)^2 \cdot \left(\frac{N_2}{N_1}\right)^2 \cdot \left(\frac{\rho_2}{\rho_1}\right)$
- Fan static pressure (D-6) $P_{s2} = P_{t2} - P_{v2}$
- Fan static efficiency (D-7) $\eta_{s2} = \eta_{t1} \cdot \left(\frac{P_{s2}}{P_{t2}}\right)$

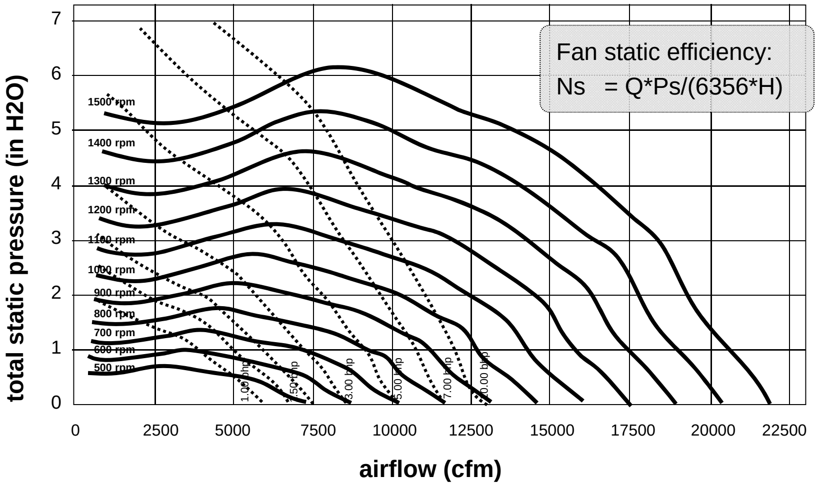
Where
η_t = total efficiency
η_s = static efficiency
ρ = density
D = diameter
H = horsepower
N = speed
P_s = static press
P_t = total pressure
P_v = velocity press
Q = airflow



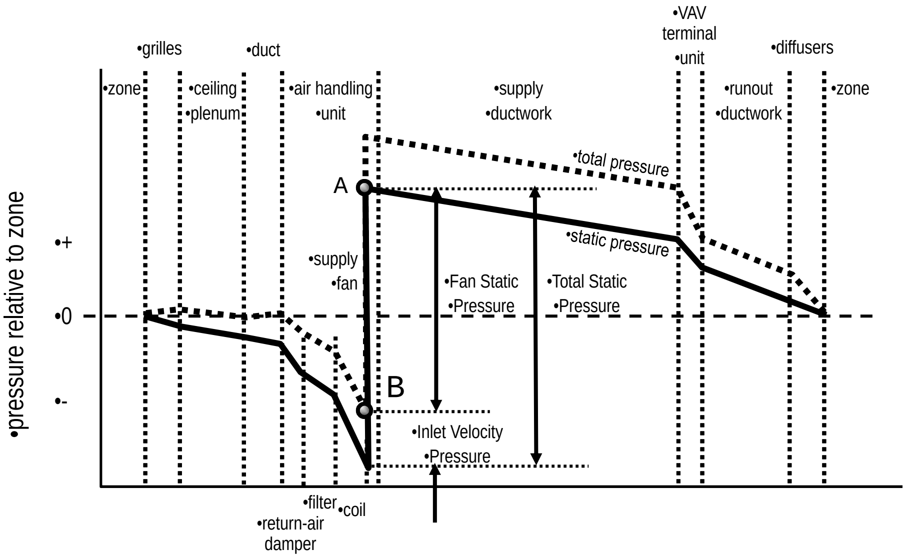
Fan Curves

D-2:
Q2 = Q1*N2/N1
= 7500*1000/1200
= 6250 cfm
D-3, D-5, D-6:
Ps2 = Ps1*(N2/N1)^2
= 3.9*(1000/1200)^2
= 2.7 in wc

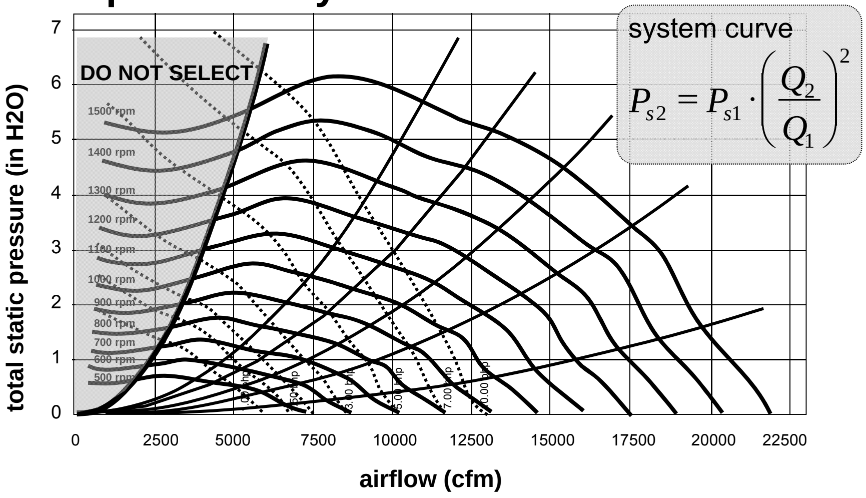
Complete Family of Fan Curves



Fan static efficiency:
Ns = Q*Ps/(6356*H)



Complete Family of Fan Curves



system curve

$$P_{s2} = P_{s1} \cdot \left(\frac{Q_2}{Q_1} \right)^2$$

Summary of Fan Basics

- Accurate fan performance curves are generated in the lab according to industry standards
  - AMCA 210 (ASHRAE 51)
  - AHRI 430
- Use fan laws to predict fan parameters
- System resistance curves characterize air systems in terms of static pressure and airflow
- “Do Not Select” or “Surge” line limits the range of fan operation at low flow conditions

Fans in Air-Handling Units

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Fan/Unit Considerations

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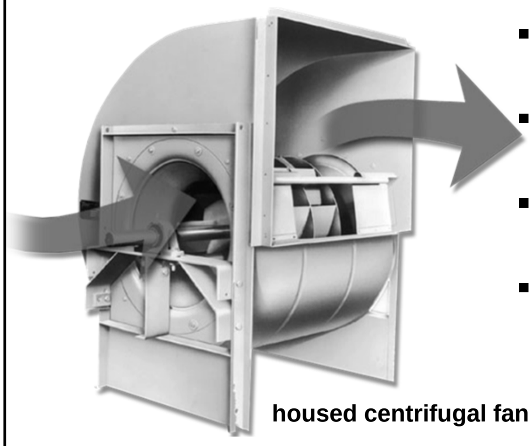
types of fans

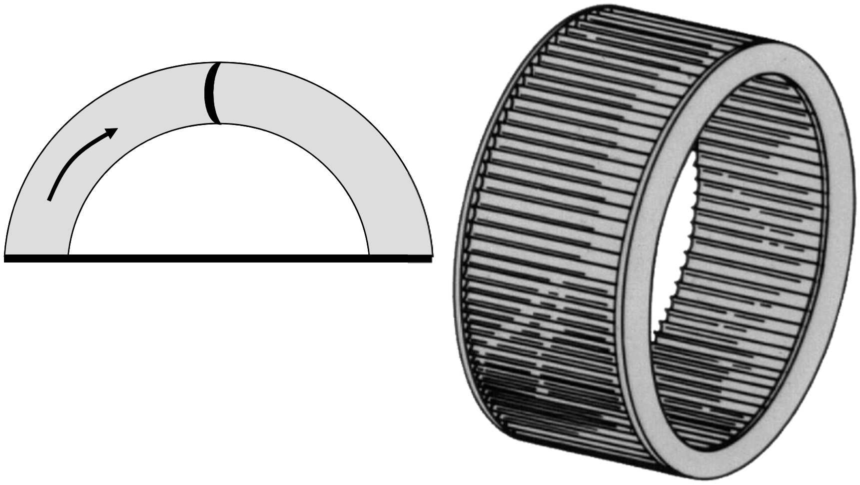
Characteristics of Centrifugal Fans



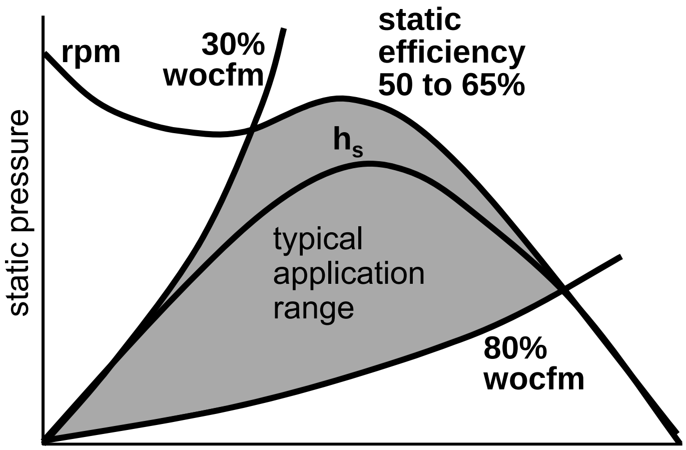
housed centrifugal fan

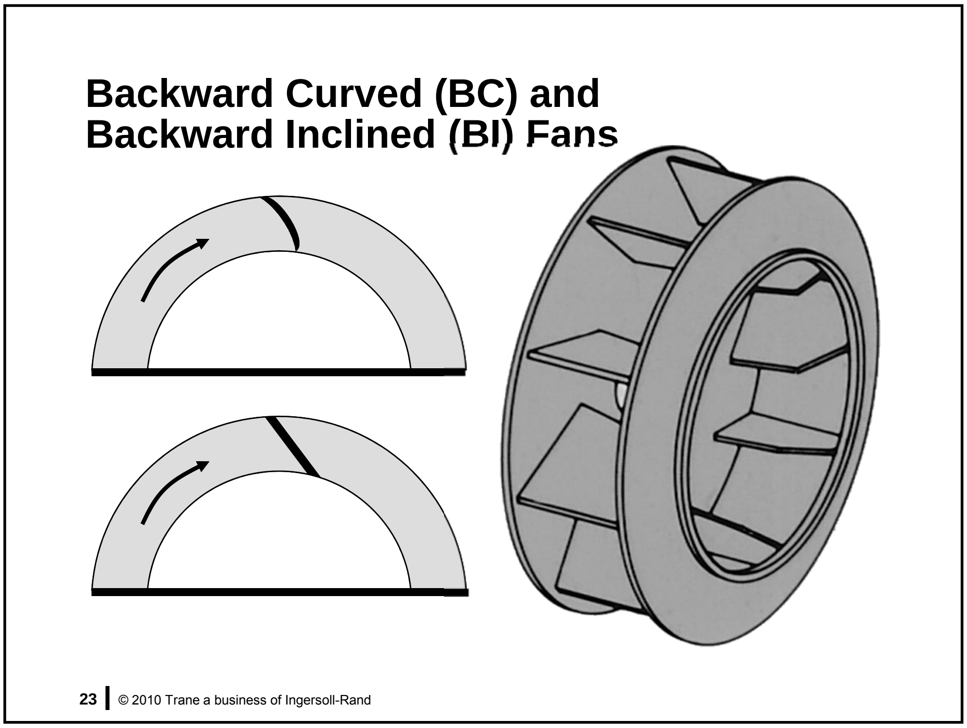
- Shape of fan blades (FC, BC, BI, AF)
- Housed versus unhoused (plenum)
- Belt-driven versus direct-driven
- Single fan versus a multiple-fan array

Forward Curved (FC) Fan

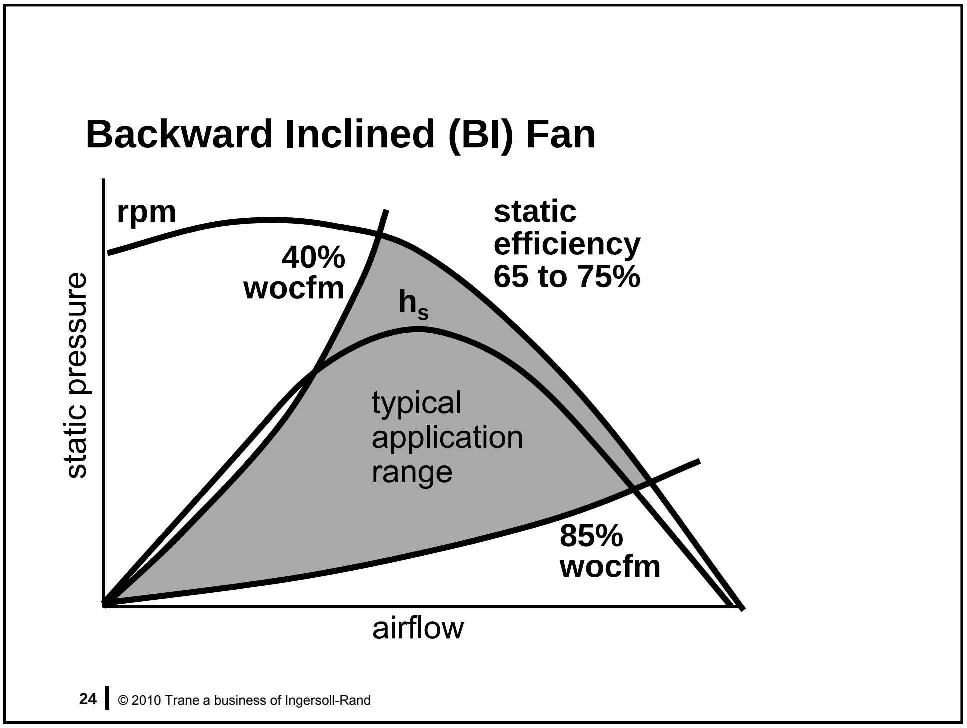


Forward Curved (FC) Fan

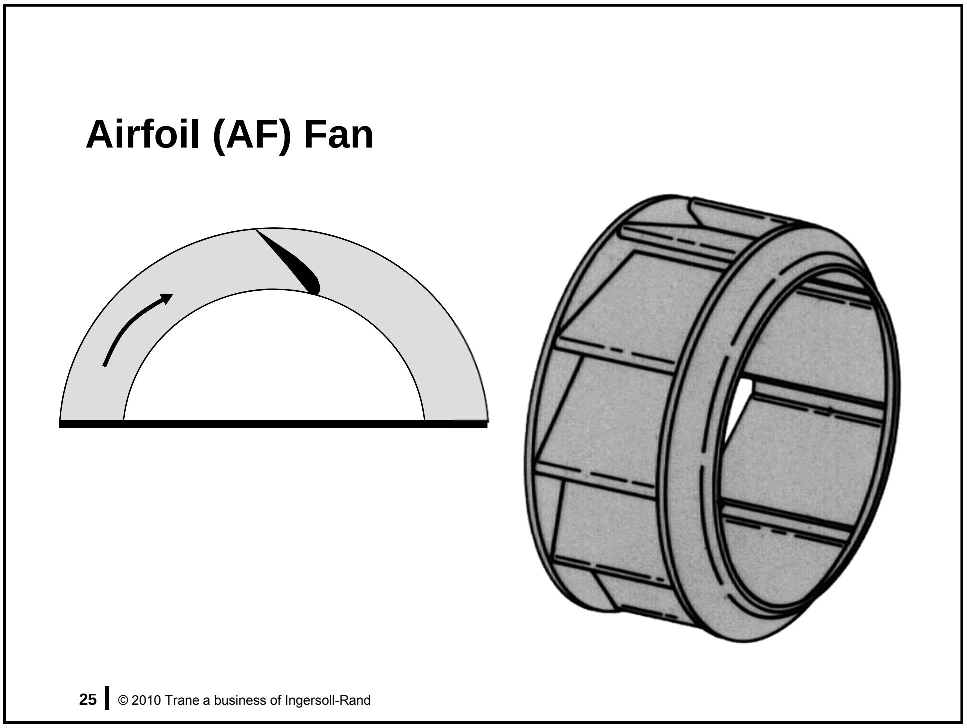




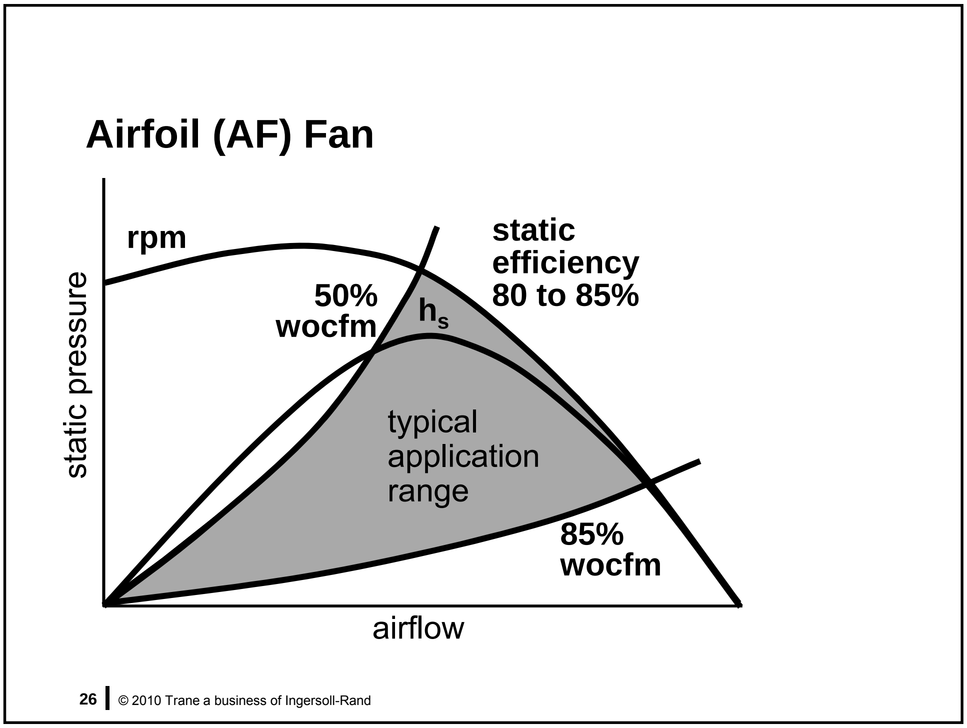
Backward Curved (BC) and Backward Inclined (BI) Fans



Backward Inclined (BI) Fan



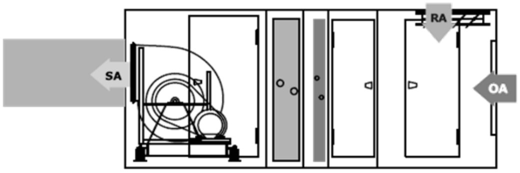
Airfoil (AF) Fan



Airfoil (AF) Fan

Impact of Blade Shape on Fan Input Power

Fan type and wheel diameter	Input power, bhp	Rotational speed, rpm
Housed FC, 25 in.	13.0	775
Housed AF, 25 in.	11.8	1320



Based on a typical VAV air-handling unit configuration (OA/RA mixing box, high-efficiency filter, hot-water heating coil, chilled-water cooling coil, and draw-thru supply fan with a single discharge opening off the fan section) operating at 13,000 cfm and a 3.8 in. H_2O total static pressure drop.

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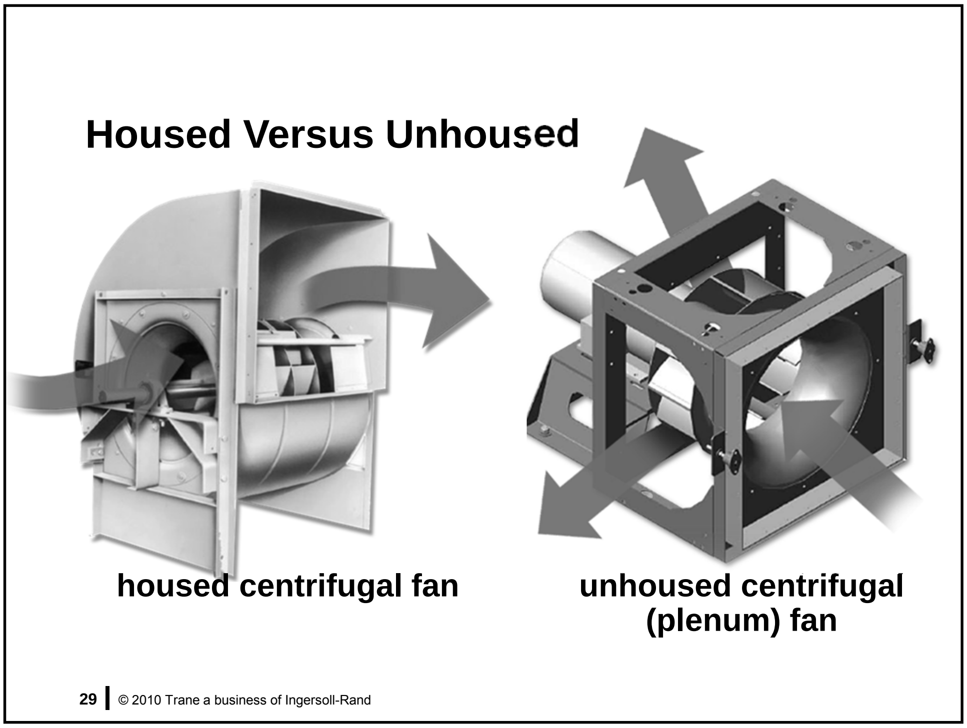
summary

Shape of Fan Blades

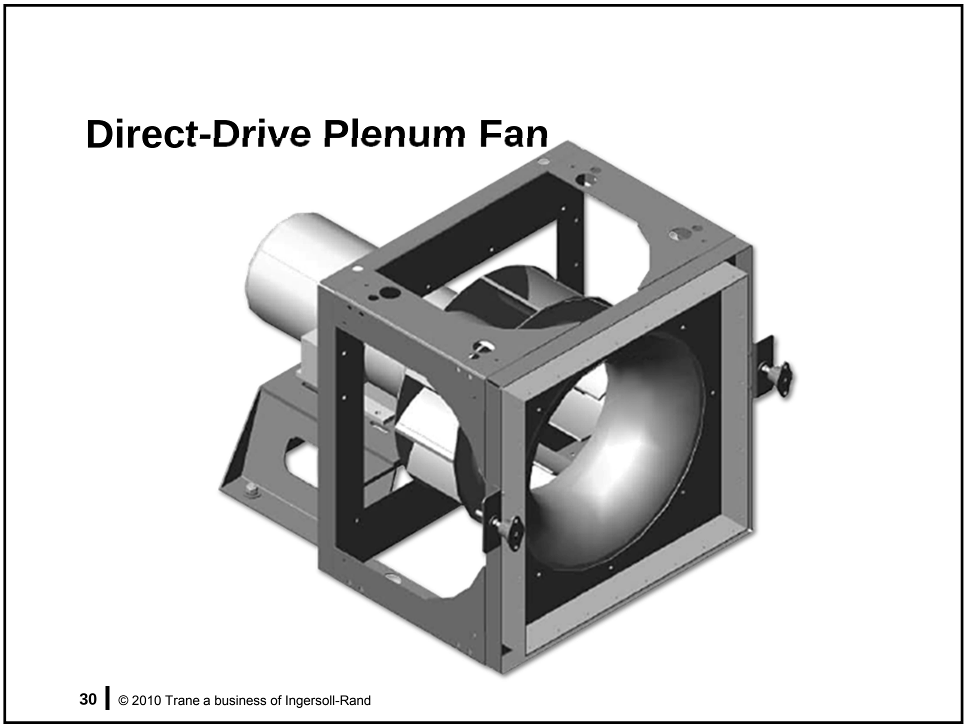
- FC fans are typically the lowest cost and are often the most forgiving (wide application range, less severe surge characteristics)
  - Very popular in packaged units and light commercial equipment, where less attention is given to duct connections and layout
- AF fans are typically the most efficient, but require more attention to avoid surge
  - More common in larger packaged rooftops and air-handling units, where more attention is given to proper duct connections and layout



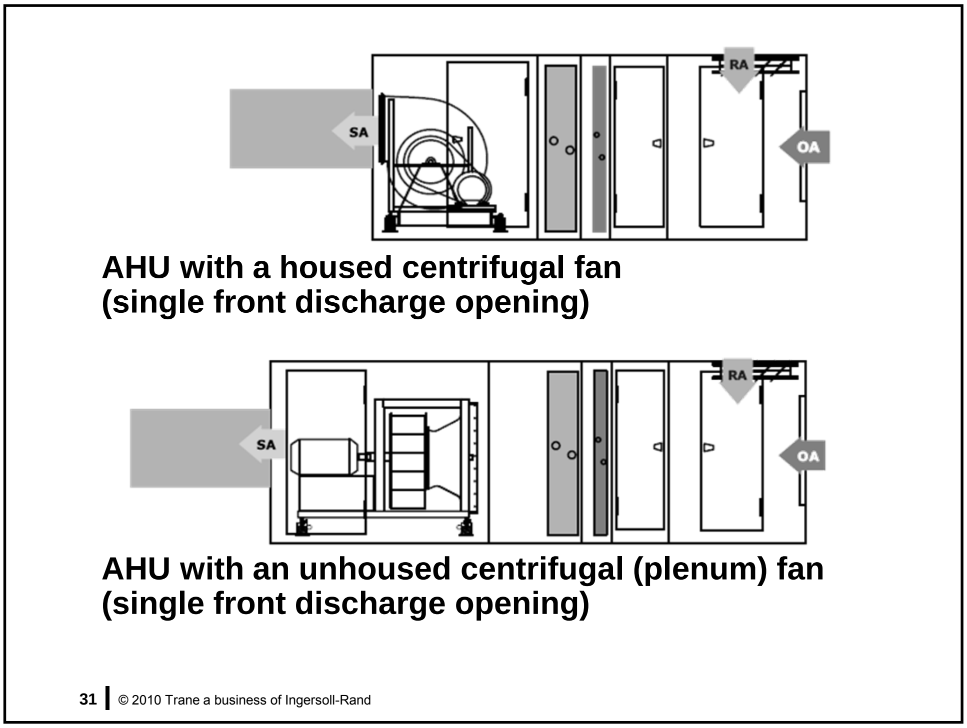
Housed Versus Unhoused

housed centrifugal fan

unhoused centrifugal (plenum) fan

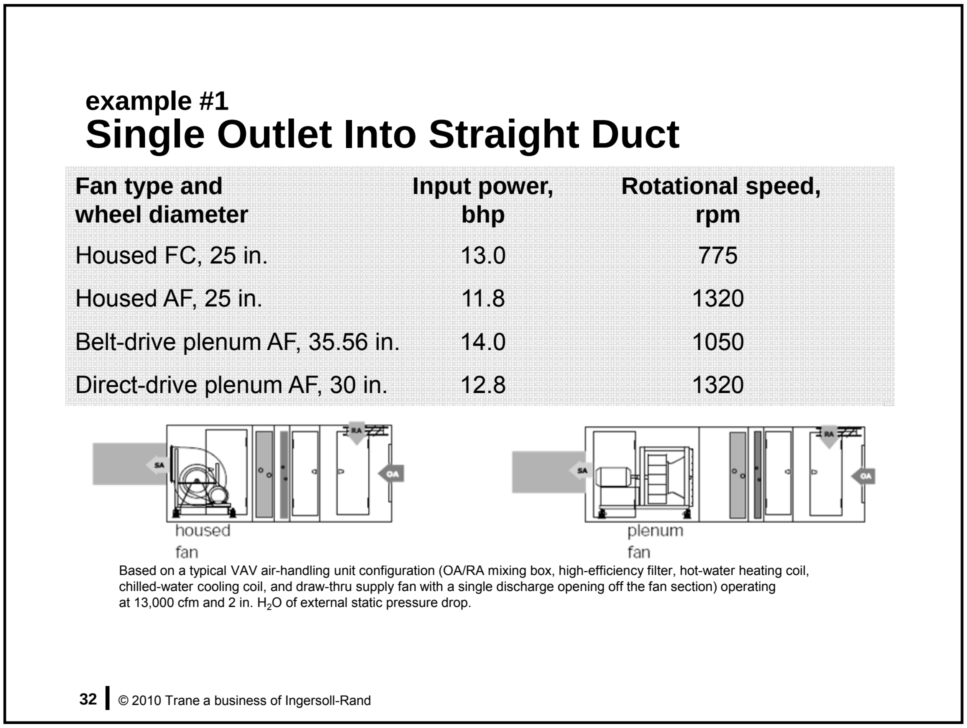


Direct-Drive Plenum Fan



**AHU with a housed centrifugal fan
(single front discharge opening)**

**AHU with an unhoused centrifugal (plenum) fan
(single front discharge opening)**



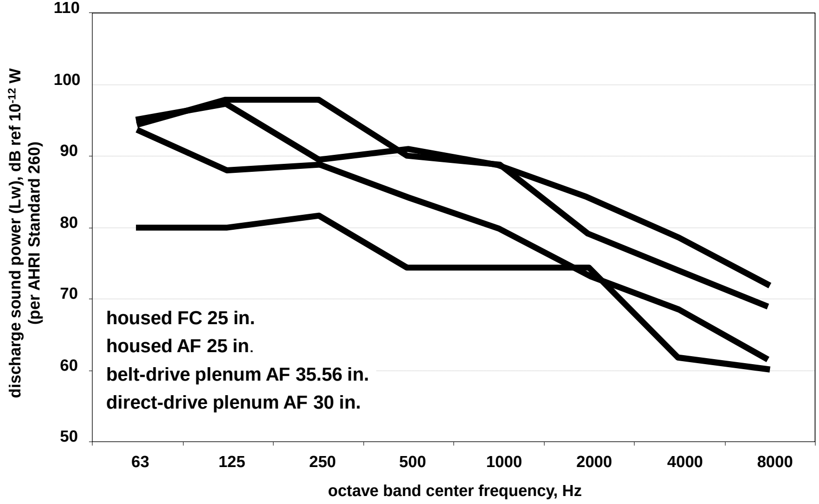
example #1

Single Outlet Into Straight Duct

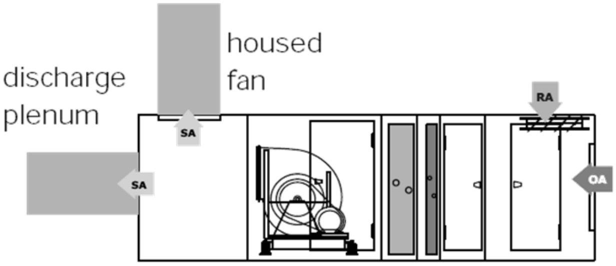
Fan type and wheel diameter	Input power, bhp	Rotational speed, rpm
Housed FC, 25 in.	13.0	775
Housed AF, 25 in.	11.8	1320
Belt-drive plenum AF, 35.56 in.	14.0	1050
Direct-drive plenum AF, 30 in.	12.8	1320

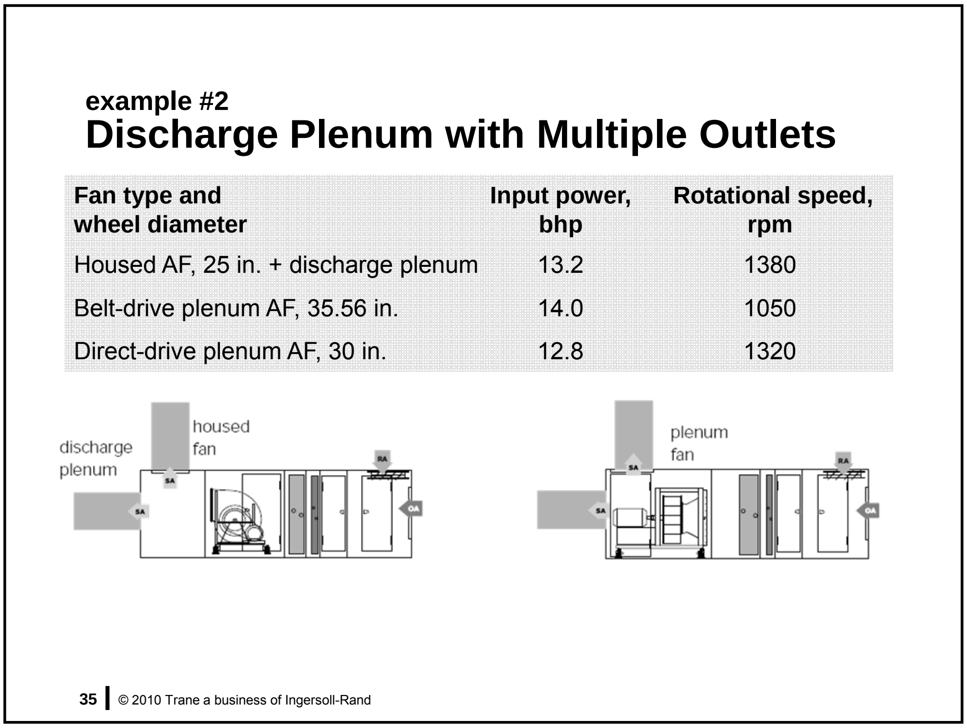
Based on a typical VAV air-handling unit configuration (OA/RA mixing box, high-efficiency filter, hot-water heating coil, chilled-water cooling coil, and draw-thru supply fan with a single discharge opening off the fan section) operating at 13,000 cfm and 2 in. H_2O of external static pressure drop.

example #1
Single Outlet Into Straight Duct



example #2
Discharge Plenum with Multiple Outlets

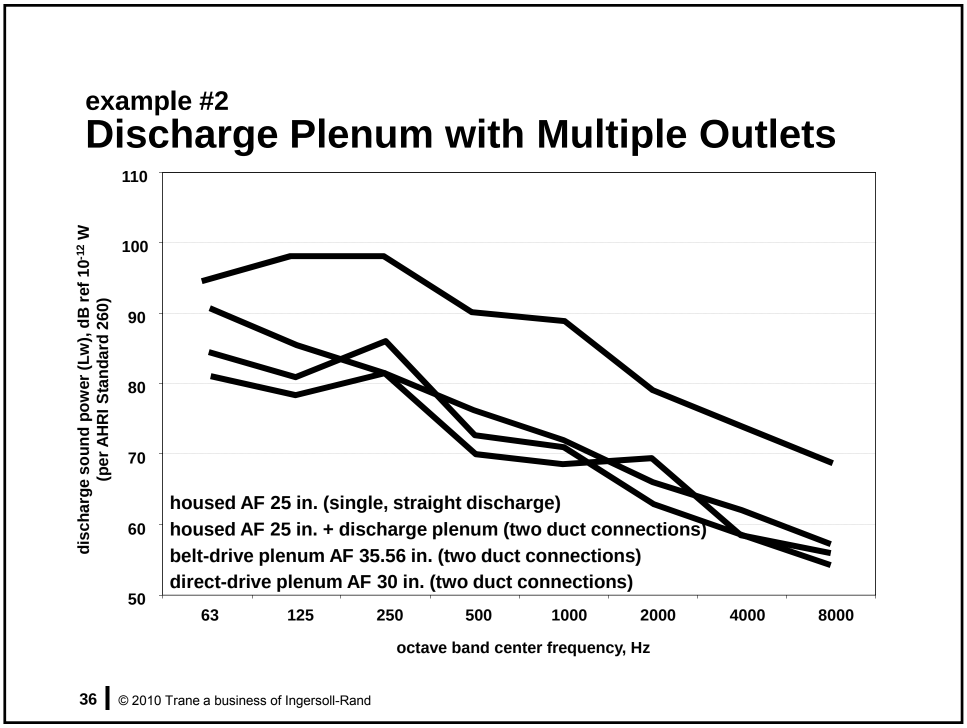




example #2

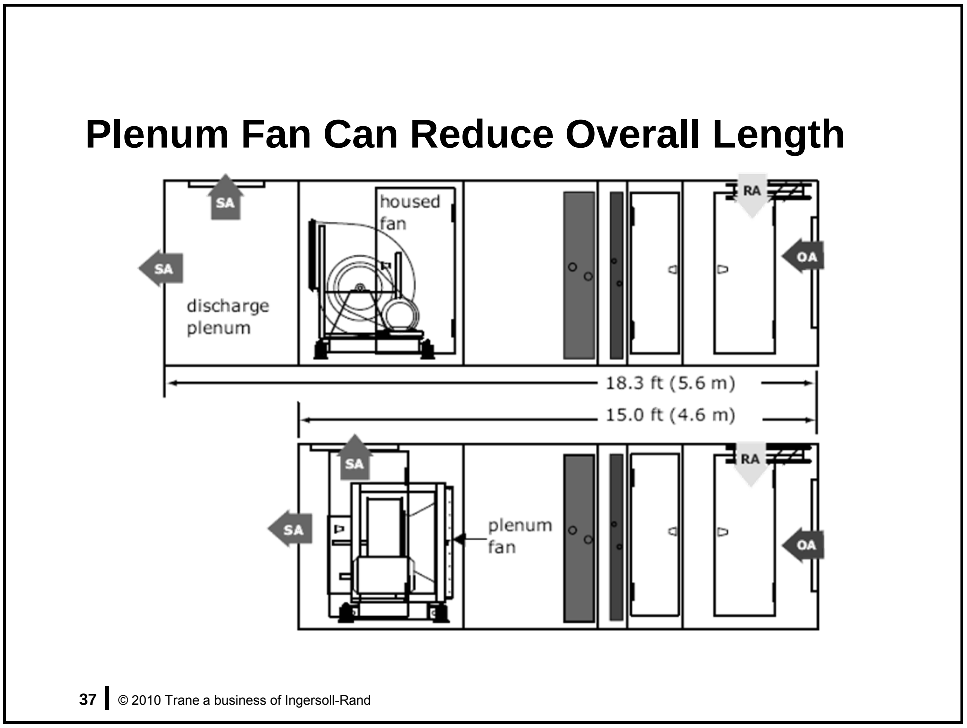
Discharge Plenum with Multiple Outlets

Fan type and wheel diameter	Input power, bhp	Rotational speed, rpm
Housed AF, 25 in. + discharge plenum	13.2	1380
Belt-drive plenum AF, 35.56 in.	14.0	1050
Direct-drive plenum AF, 30 in.	12.8	1320

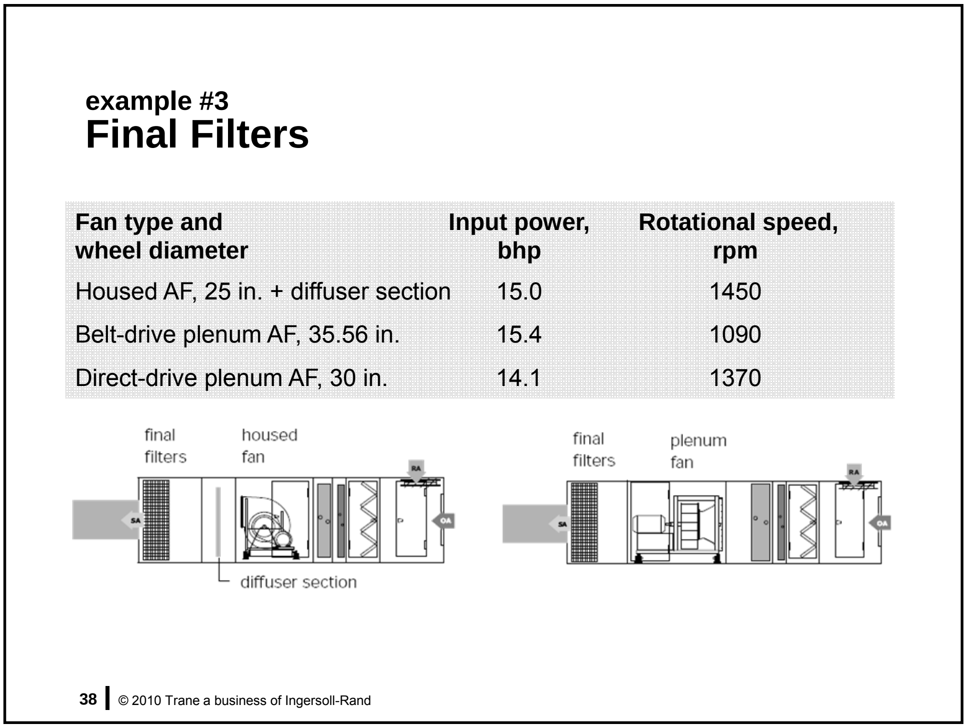


example #2

Discharge Plenum with Multiple Outlets



Plenum Fan Can Reduce Overall Length



example #3

Final Filters

Fan type and wheel diameter	Input power, bhp	Rotational speed, rpm
Housed AF, 25 in. + diffuser section	15.0	1450
Belt-drive plenum AF, 35.56 in.	15.4	1090
Direct-drive plenum AF, 30 in.	14.1	1370

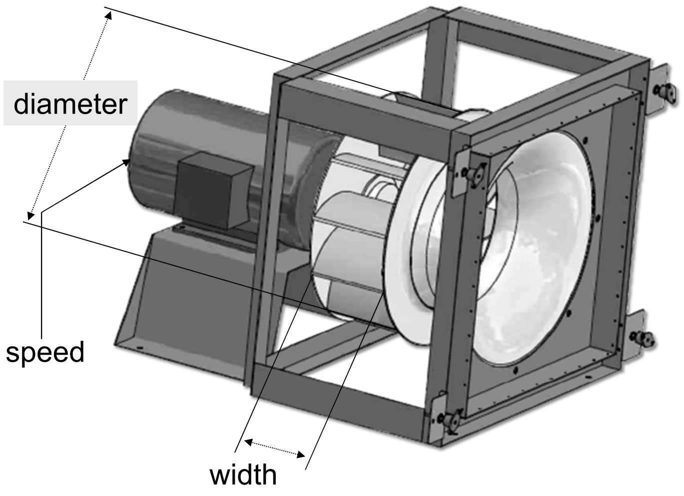
summary

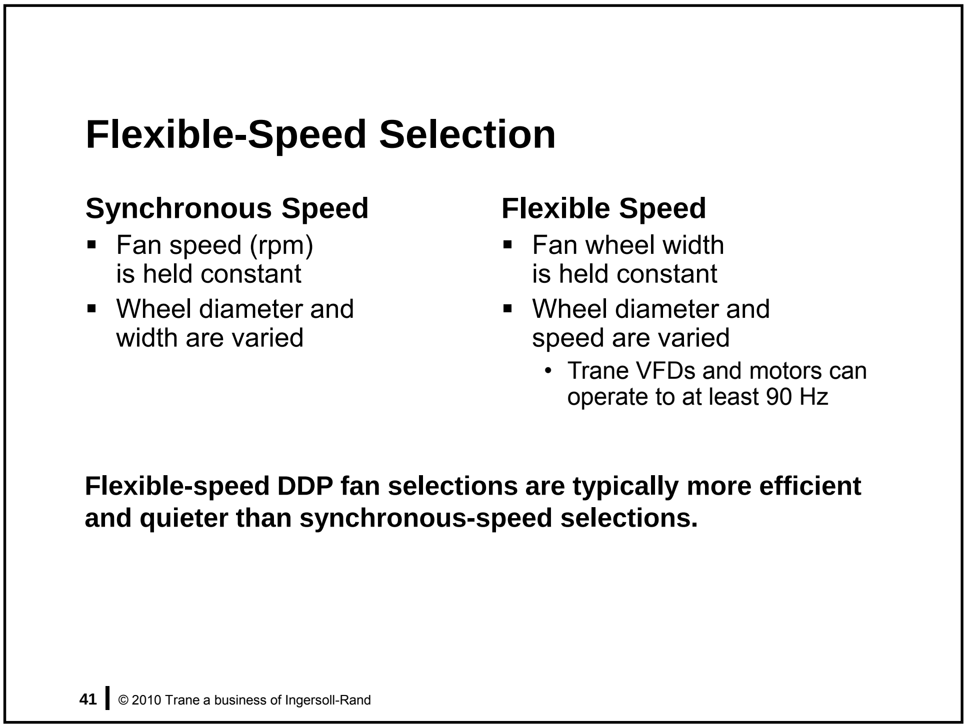
Housed vs. Plenum Fans

- When discharging into a single, sufficiently-long, straight section of duct that is about the same size as the fan outlet, a housed fan will likely require less power than a plenum fan, but a plenum fan will likely have lower discharge sound levels.
- If a discharge plenum is added downstream of a housed fan to reduce sound levels or to allow for discharge flexibility, a direct-drive plenum fan will likely require less power than a housed airfoil fan, with similar discharge sound levels. But the plenum fan will likely result in a shorter air-handling unit.
- With downstream sections (such as a discharge plenum, final filter, gas heater, or even a blow-thru cooling coil), a direct-drive plenum fan will likely require less power than either a housed or belt-driven plenum fan.

direct-drive plenum fan

Selection Parameters





Flexible-Speed Selection

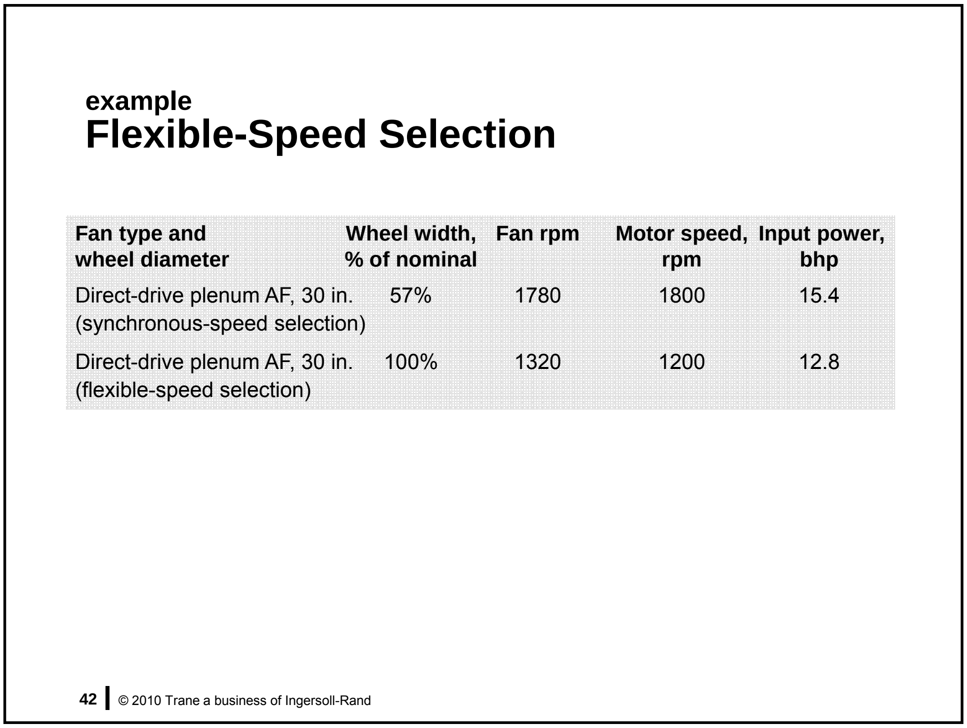
Synchronous Speed

- Fan speed (rpm) is held constant
- Wheel diameter and width are varied

Flexible Speed

- Fan wheel width is held constant
- Wheel diameter and speed are varied
  - Trane VFDs and motors can operate to at least 90 Hz

Flexible-speed DDP fan selections are typically more efficient and quieter than synchronous-speed selections.

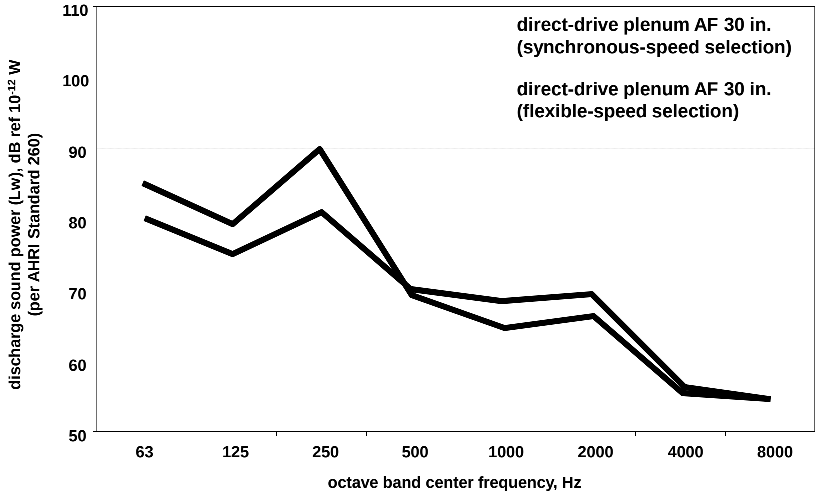


example
Flexible-Speed Selection

Fan type and wheel diameter	Wheel width, % of nominal	Fan rpm	Motor speed, rpm	Input power, bhp
Direct-drive plenum AF, 30 in. (synchronous-speed selection)	57%	1780	1800	15.4
Direct-drive plenum AF, 30 in. (flexible-speed selection)	100%	1320	1200	12.8

example

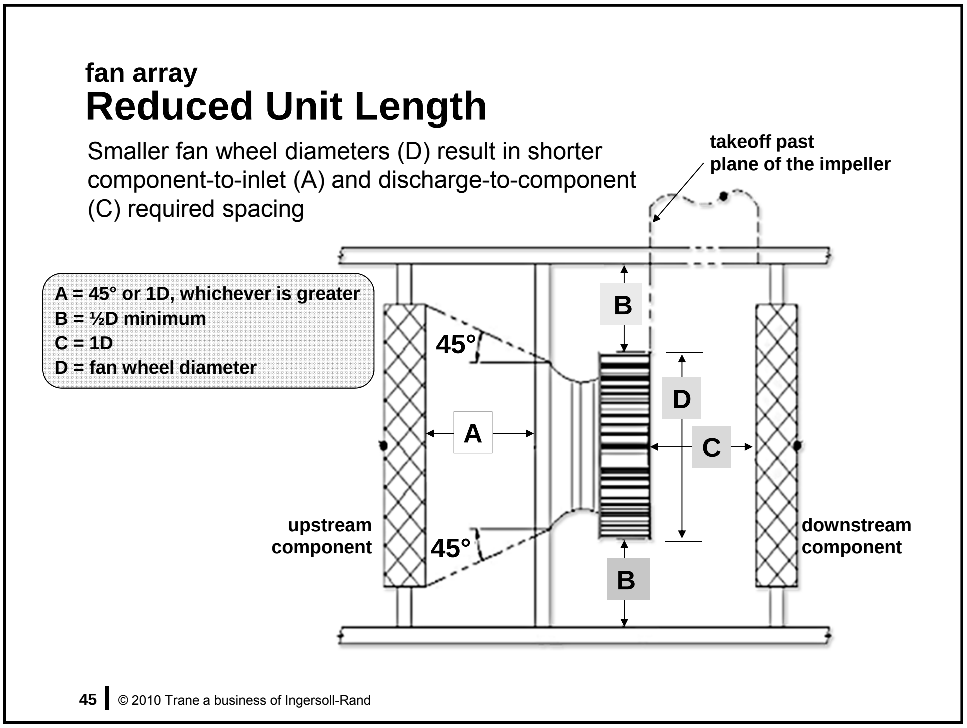
Flexible-Speed Selection



Multiple Fans (Fan Array)

upstream (inlet) side

downstream (outlet) side



fan array

Reduced Unit Length

Smaller fan wheel diameters (D) result in shorter component-to-inlet (A) and discharge-to-component (C) required spacing

A = 45° or 1D, whichever is greater
B = ½D minimum
C = 1D
D = fan wheel diameter

fan array

There is a Limit to the Length Reduction

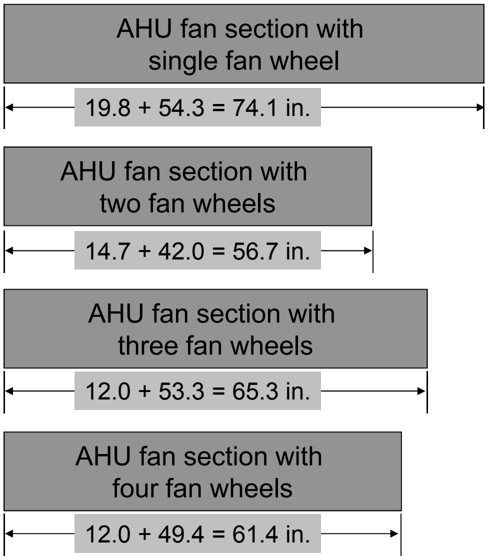
- Minimum service clearance for access doors, people, ladders, or a hoist
- For top, bottom, or side inlet or discharge connections, additional space may be needed for proper airflow distribution
- If backdraft or isolation dampers are provided, they typically add length to the fan section

example length reduction
Single Fan Versus Fan Array

Qty	Diameter, in.	Upstream spacing req'd, in.	Upstream service clear, in.	Upstream total, in.
1	33	19.8	12	19.8
2	24.5	14.7	12	14.7
3	20	12.0	12	12.0
4	18.75	11.0	12	12.0

example length reduction
Single Fan Versus Fan Array

Qty	Diameter, in.	Downstream spacing req'd, in.	Length of fan + motor, in.		Downstream service clear, in.	Downstream total, in.
1	33	50.5	54.3	+	0	54.3
2	24.5	38.8	42.0	+	0	42.0
3	20	33.1	35.3	+	18	53.3
4	18.75	29.9	31.4	+	18	49.4



AHU fan section with single fan wheel

19.8 + 54.3 = 74.1 in.

AHU fan section with two fan wheels

14.7 + 42.0 = 56.7 in.

AHU fan section with three fan wheels

12.0 + 53.3 = 65.3 in.

AHU fan section with four fan wheels

12.0 + 49.4 = 61.4 in.

example

Providing Redundancy with a Fan Array

Qty running	Diameter, in.	Level of redundancy	Airflow (each fan), cfm	Input power (each fan), bhp	Input power (total), bhp	Motor size (each fan), hp	
2	24.5	Design	7500	6.55	13.10	7.5	
1	24.5	100%	15000	16.13	16.13	20	(change from 7.5 to 20 hp motors)
1	24.5	70%	10500	7.13	7.13	7.5	(no change in motor sizes)

example

Providing Redundancy with a Fan Array

Qty running	Diameter, in.	Level of redundancy	Airflow (each fan), cfm	Input power (each fan), bhp	Input power (total), bhp	Motor size (each fan), hp	
2	24.5	Design	7500	6.55	13.10	7.5	
1	24.5	100%	15000	16.13	16.13	20	(change from 7.5 to 20 hp motors)
1	24.5	70%	10500	7.13	7.13	7.5	(no change in motor sizes)

Qty running	Diameter, in.	Level of redundancy	Airflow (each fan), cfm	Input power (each fan), bhp	Input power (total), bhp	Motor size (each fan), hp	
3	20	Design	5000	4.68	14.04	7.5	
2	20	100%	7500	7.43	14.86	7.5	(no change in motor sizes)

Qty running	Diameter, in.	Level of redundancy	Airflow (each fan), cfm	Input power (each fan), bhp	Input power (total), bhp	Motor size (each fan), hp	
4	18.25	Design	3750	3.53	14.12	5	
3	18.25	100%	5000	4.71	14.13	5	(no change in motor sizes)

Providing Redundancy with a Fan Array

- Two fans can often provide 100% redundancy and results in the lowest total power when all fans are operating, but may require larger fan motors to be provided.
    - If less than 100% is acceptable, two fans may not need to increase motor sizes.
- Three or four fans can typically provide 100% redundancy without significant changes in motor size.

for more information

Direct-Drive Plenum Fans and Fan Arrays

- “Direct-Drive Plenum Fans for Trane Climate Changer™ Air Handlers,” Trane engineering bulletin, CLCH-PRB021-EN

TRANE

Engineering Bulletin

Direct-Drive Plenum Fans

For Trane Climate Changer™ Air Handlers

June 2009

CLCH-PRB021-EN

summary

Single Fan Versus a Fan Array

	Single DDP Fan	Multiple DDP Fans (Fan Array)	
		Fewer Fans	More Fans
AHU footprint	✓	✓✓	✓✓✓
Redundancy	none	✓✓✓	✓✓✓
Serviceability	✓	✓✓	✓✓✓
AHU cost	✓✓✓	✓✓	✓
Efficiency	✓✓✓	✓✓	✓
AHU acoustics	✓✓✓	✓✓	✓
Fan reliability	✓✓✓	✓✓	✓

summary

Single Fan Versus a Fan Array

- Benefits of using a fan array
  - Reduction in overall length of air-handling unit
  - Redundancy
  - Easier to replace fans and motors
- Drawbacks of using a fan array
  - Increased air-handling unit cost
  - Higher input power
  - Higher sound levels
- When a fan array is desired, using fewer larger fans will typically be a better overall solution than using many smaller fans

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engineers newsletter

volume 39–1 • providing insights for today's hvac system designer

Direct-Drive Plenum Fans and Fan Arrays

Direct Drive Plenum Fans

Fans in Air-Handling Systems

engineers newsletter LIVE

Impact of System Configuration on Fan Selection

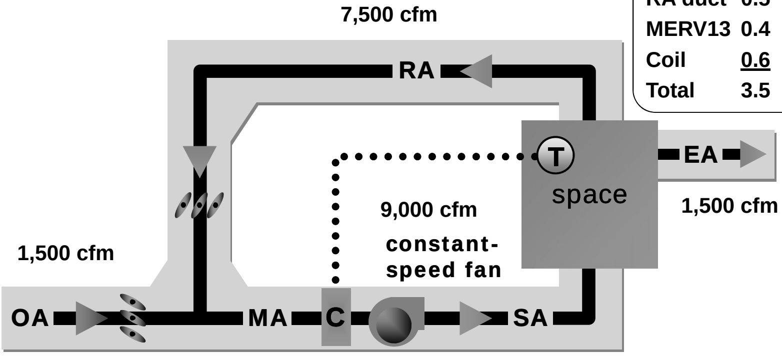
---

constant volume (CV)

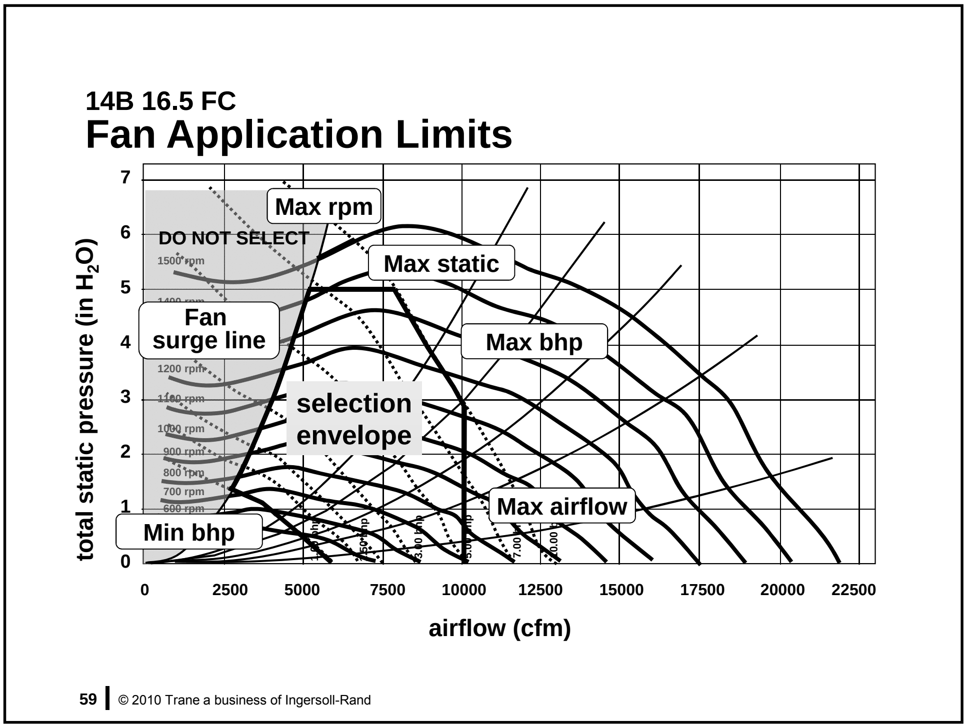
Basic System

Pressure drops
@ 9,000 cfm/7,500 cfm

Device	Low	High
SA duct	2.0	2.0
RA duct	0.5	0.5
MERV13	0.4	1.2
Coil	0.6	0.9
Total	3.5	4.6



a size 14 unit with a 16.5 FC fan might work



14B 16.5 FC
Fan Application Limits

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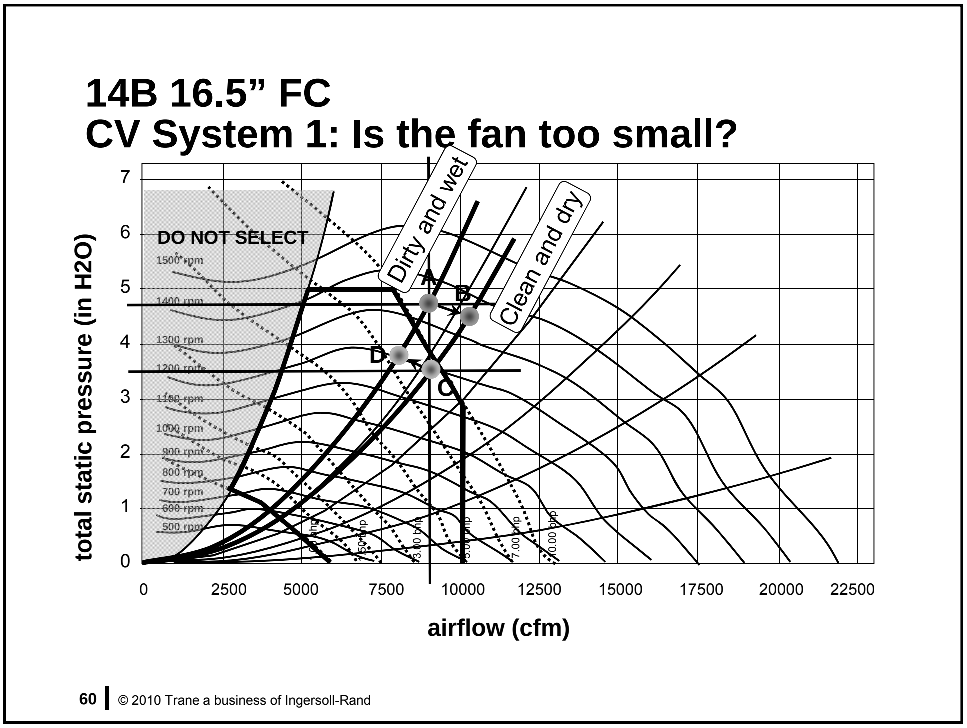

14B 16.5” FC
CV System 1: Is the fan too small?

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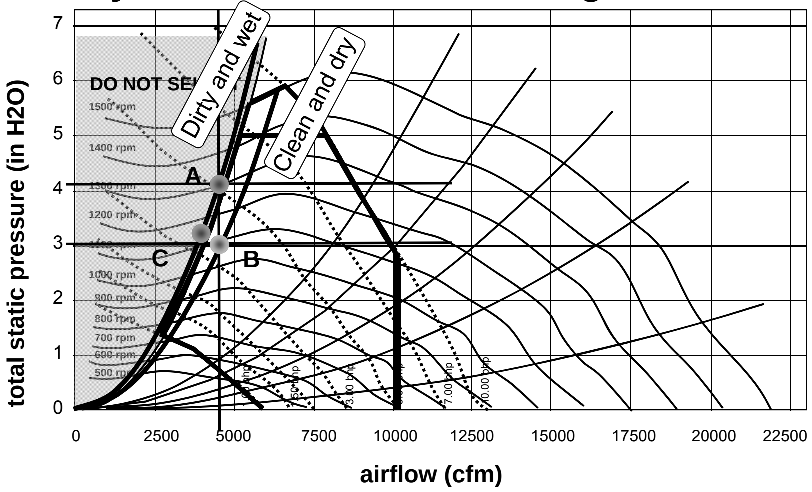
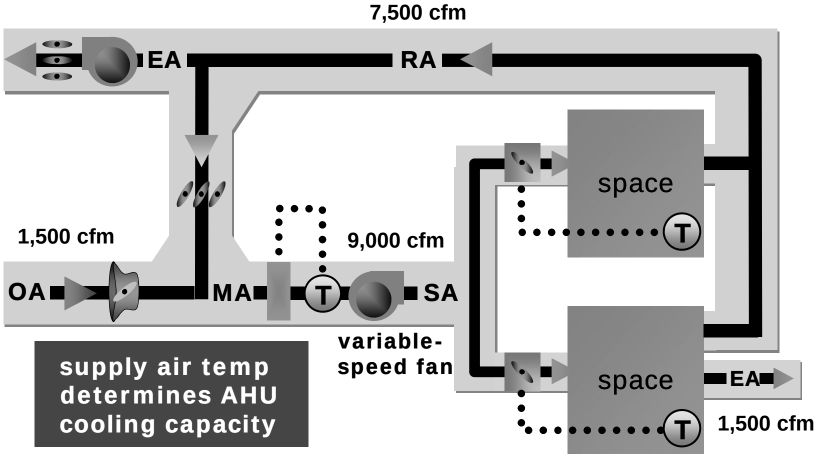
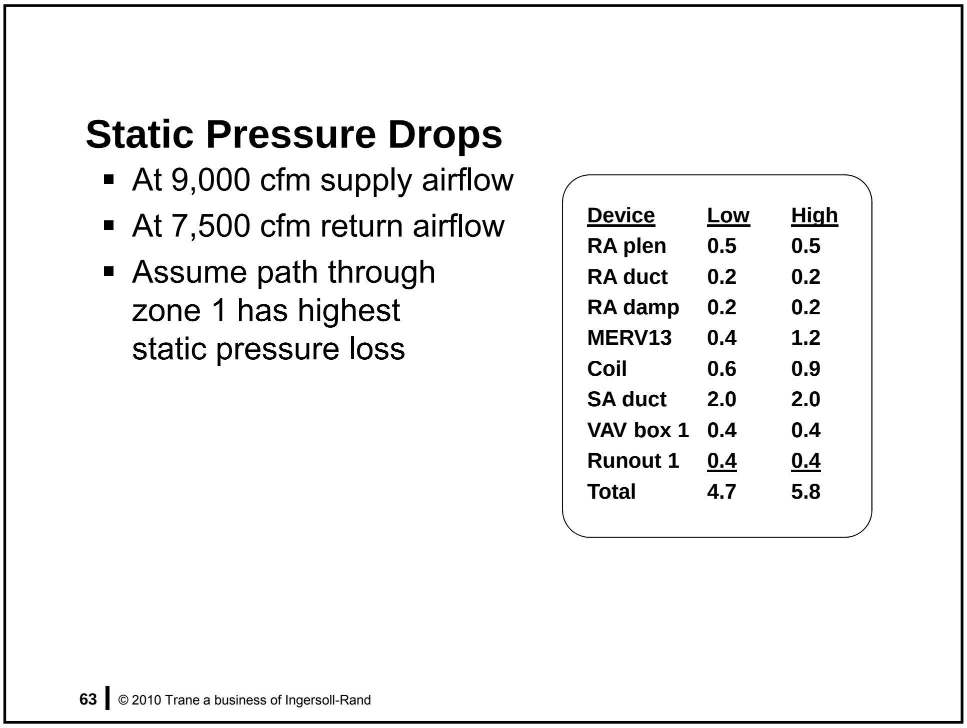
14B 16.5 FC
CV System 2: Is the fan too big?



Multiple-Zone VAV With Relief Fan



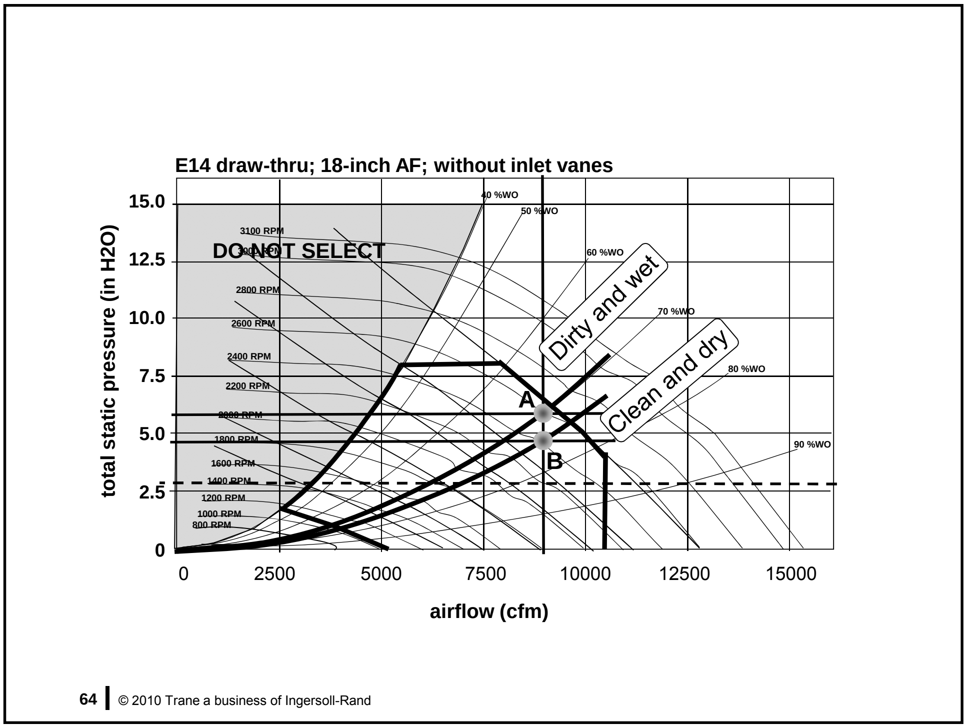
supply air temp determines AHU cooling capacity



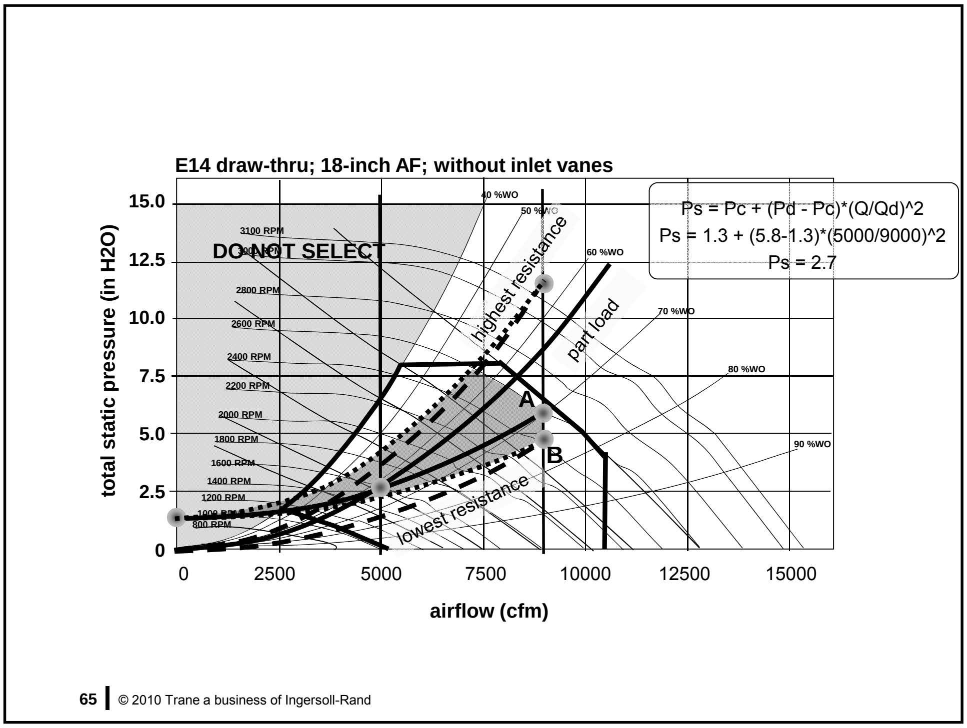
Static Pressure Drops

- At 9,000 cfm supply airflow
- At 7,500 cfm return airflow
- Assume path through zone 1 has highest static pressure loss

Device	Low	High
RA plen	0.5	0.5
RA duct	0.2	0.2
RA damp	0.2	0.2
MERV13	0.4	1.2
Coil	0.6	0.9
SA duct	2.0	2.0
VAV box 1	0.4	0.4
Runout 1	0.4	0.4
Total	4.7	5.8



E14 draw-thru; 18-inch AF; without inlet vanes



Ps = Pc + (Pd - Pc)*(Q/Qd)^2
Ps = 1.3 + (5.8-1.3)*(5000/9000)^2
Ps = 2.7

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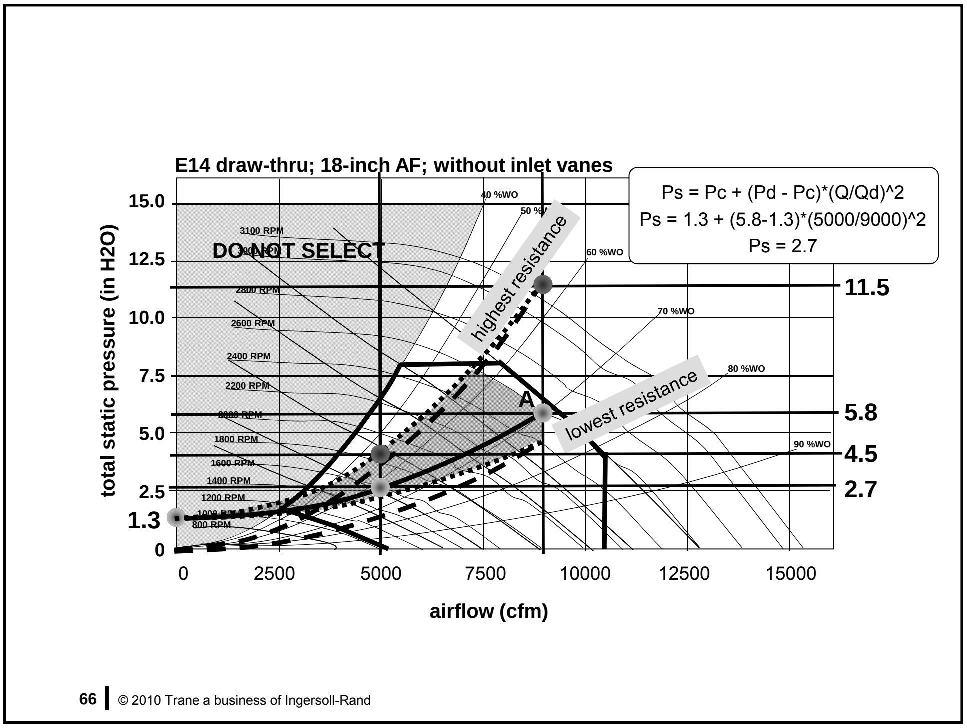

Ps = Pc + (Pd - Pc)*(Q/Qd)^2
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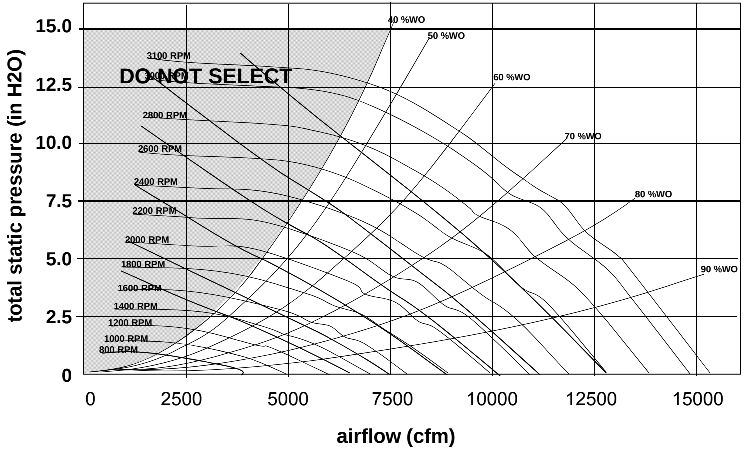
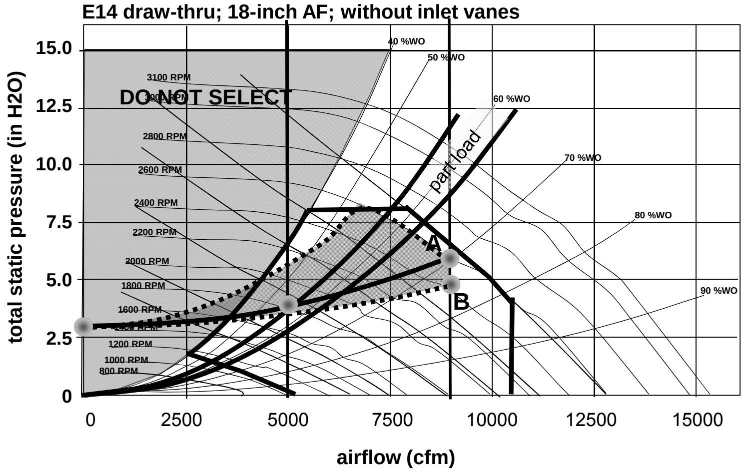
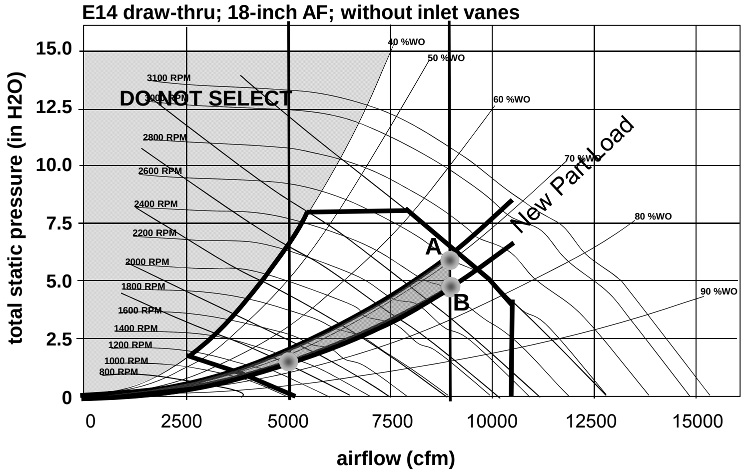
E14 draw-thru; 18-inch AF; without inlet vanes

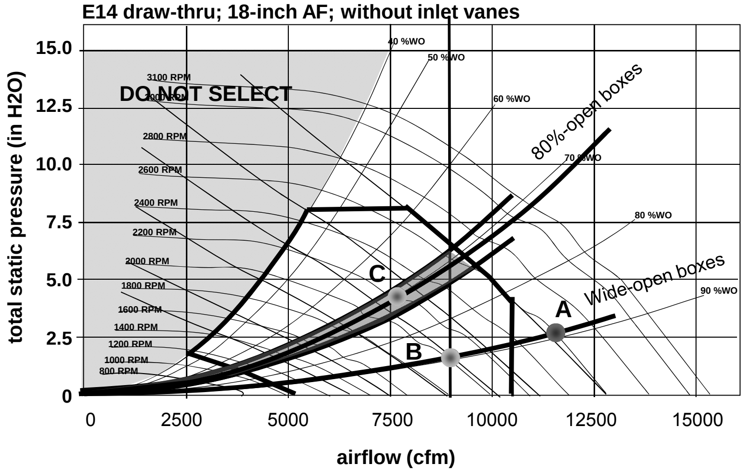


E14 draw-thru; 18-inch AF; without inlet vanes

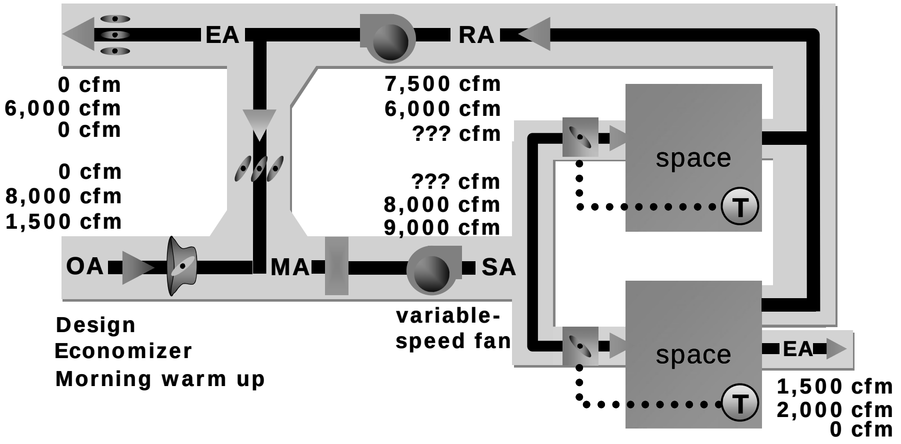




Morning warm-up operation



Multiple-Zone VAV with Return Fan



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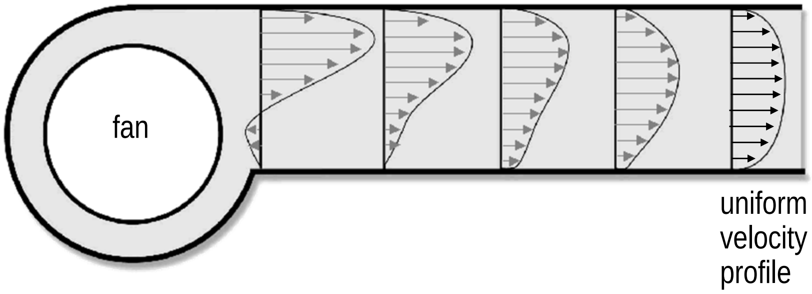
System Effect

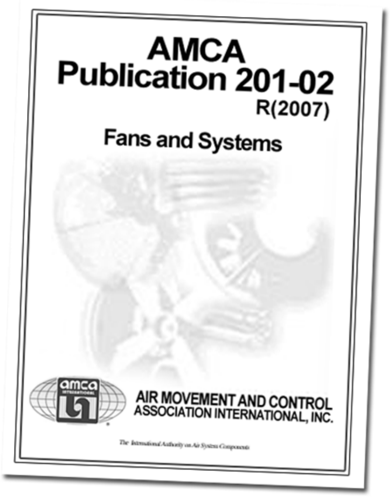
Developing a Uniform Velocity Profile



Common System Effects

- 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

AMCA Publication 201, *Fans and Systems*



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

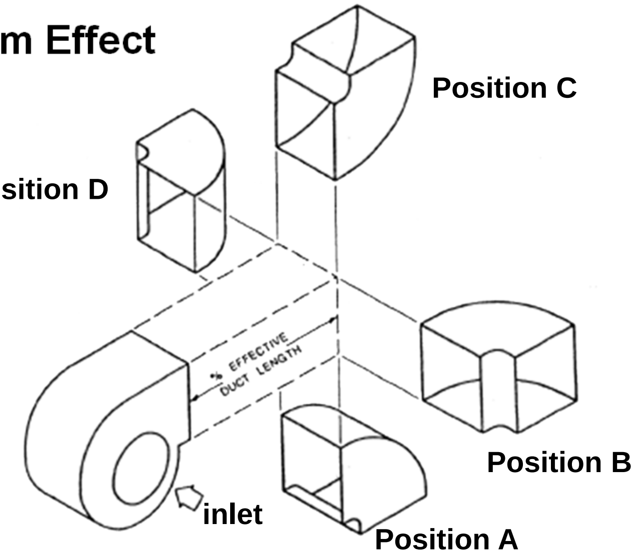
AIR MOVEMENT AND CONTROL
ASSOCIATION INTERNATIONAL, INC.

The International Authority on Air System Components

- Prediction of common System Effect Factors

example

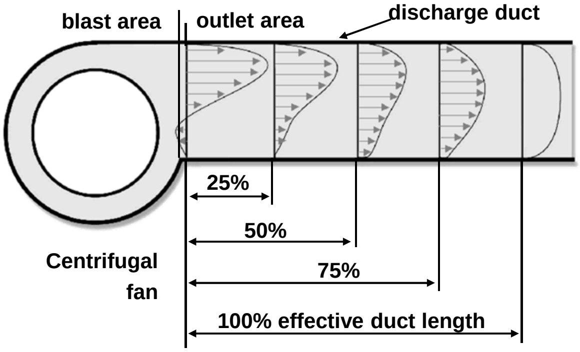
System Effect



Source: Air Movement and Control Association. 2002. *Fans and Systems*, Publication 201. Arlington Heights, IL: AMCA.

example

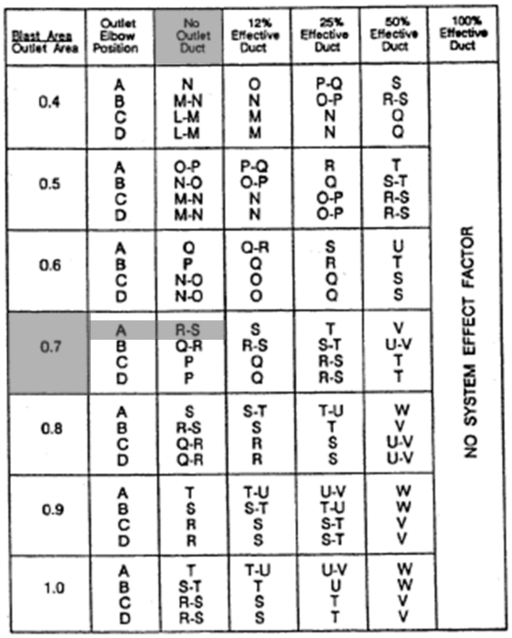
System Effect



100% Effective Duct Length

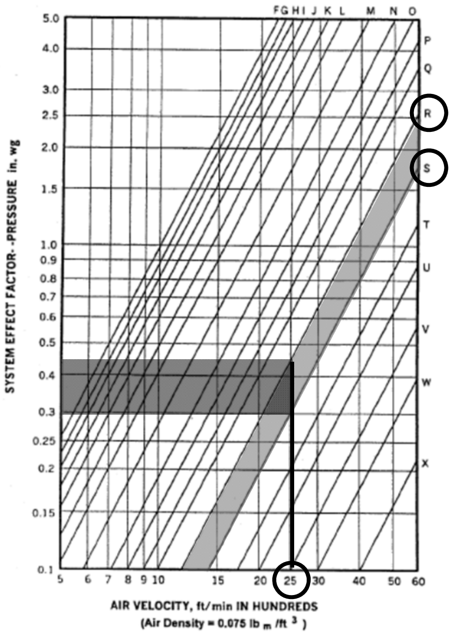
- 2.5 duct diameters for 2500 fpm (or less)
- Add 1 duct diameter for each additional 1000 fpm

Source: Air Movement and Control Association. 2002. *Fans and Systems*, Publication 201. Arlington Heights, IL: AMCA.



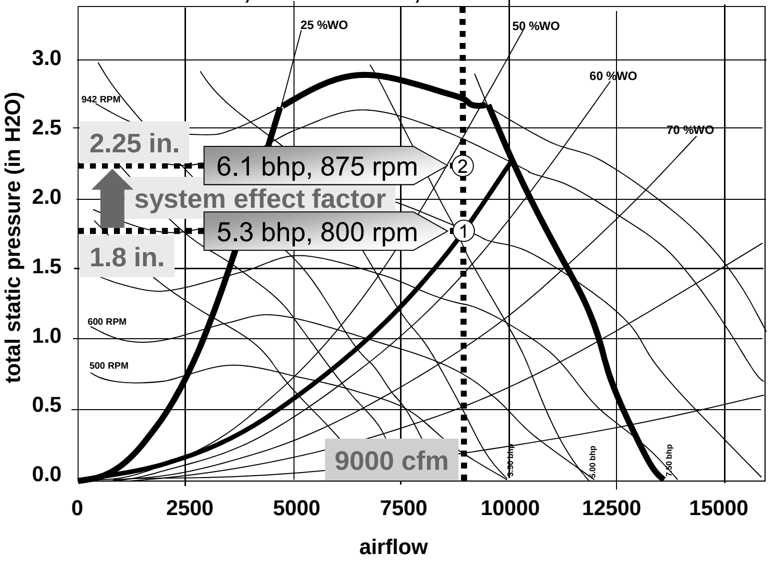
Blast Area / Outlet Area	Outlet Elbow Position	No Outlet Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
0.4	A	N	O	P-Q	S	NO SYSTEM EFFECT FACTOR
	B	M-N	N	O-P	R-S	
	C	L-M	M	N	Q	
	D	L-M	M	N	Q	
0.5	A	O-P	P-Q	R	T	
	B	N-O	O-P	Q	S-T	
	C	M-N	N	O-P	R-S	
	D	M-N	N	O-P	R-S	
0.6	A	Q	Q-R	S	U	
	B	P	Q	R	T	
	C	N-O	O	Q	S	
	D	N-O	O	Q	S	
0.7	A	R-S	S	T	V	
	B	Q-R	R-S	S-T	U-V	
	C	P	Q	R-S	T	
	D	P	Q	R-S	T	
0.8	A	S	S-T	T-U	W	
	B	R-S	S	T	V	
	C	Q-R	R	S	U-V	
	D	Q-R	R	S	U-V	
0.9	A	T	T-U	U-V	W	
	B	S	S-T	T-U	W	
	C	R	S	S-T	V	
	D	R	S	S-T	V	
1.0	A	T	T-U	U-V	W	
	B	S-T	T	U	W	
	C	R-S	S	T	V	
	D	R-S	S	T	V	

Source: Air Movement and Control Association. 2002. *Fans and Systems*, Publication 201. Arlington Heights, IL: AMCA.



Source: Air Movement and Control Association. 2002. *Fans and Systems*, Publication 201. Arlington Heights, IL: AMCA.

14A Draw-thru; 18.25-inch FC; without inlet vanes



2.25 in.

6.1 bhp, 875 rpm

system effect factor

5.3 bhp, 800 rpm

1.8 in.

9000 cfm

Fans in Air-Handling Systems

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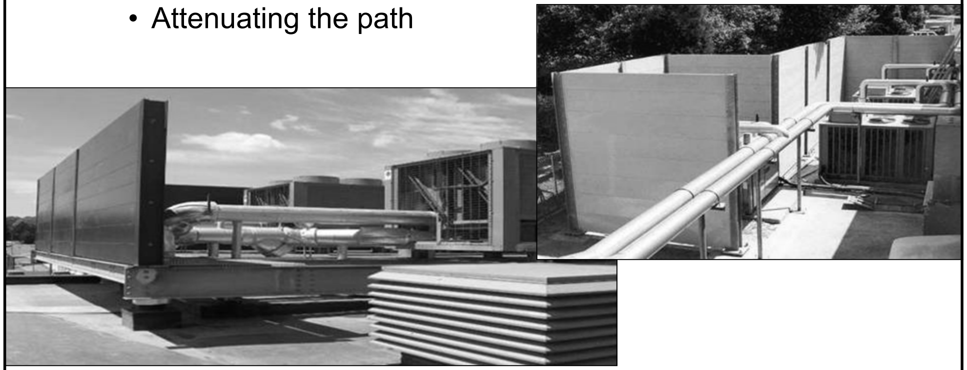
Fan Acoustics

Propeller Fans

- Reduce propeller fan sound by
  - Choosing the low noise fan option
  - Attenuating the path



Fan Sound

- Sound generation is influenced by
  - Fan type
  - Flow rate
  - Total pressure
  - Efficiency
  - Flow into and out of the fan

AHRI 260

- Includes unit impact on fan sound
  - Negative flow impacts
  - Benefits of plenums and lining
- Provides for “apples to apples” comparison

AHRI Standard 260
(formerly ARI Standard 260)

2001 Standard for

Sound Rating of Ducted Air Moving and Conditioning Equipment

AHRI Air-Conditioning, Heating, and Refrigeration Institute

2111 Wilson Boulevard, Suite 500
Arlington, VA 22201, USA
www.ahrinet.org

PH 703.524.8800
FX 703.562.1942

AHRI 260

See ***Sound Ratings and ARI Standard 260*** newsletter for additional information

engineers newsletter

providing insights for today's hvac system designer

getting to the source...

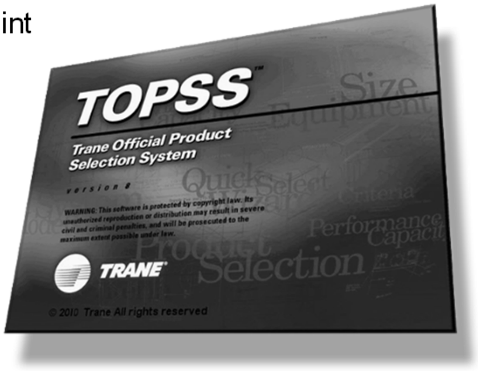
Sound Ratings and ARI Standard 260P

The Challenge of Providing Accurate Sound Ratings

"The objective of ARI 260P is to deliver application-specific sound data."

Selection Program

- Provides a convenient way to access sound data
- Shows acoustical impact of
  - Changing operating point
  - Changing fan type



TOPSS™

Trane Official Product Selection System

version 8

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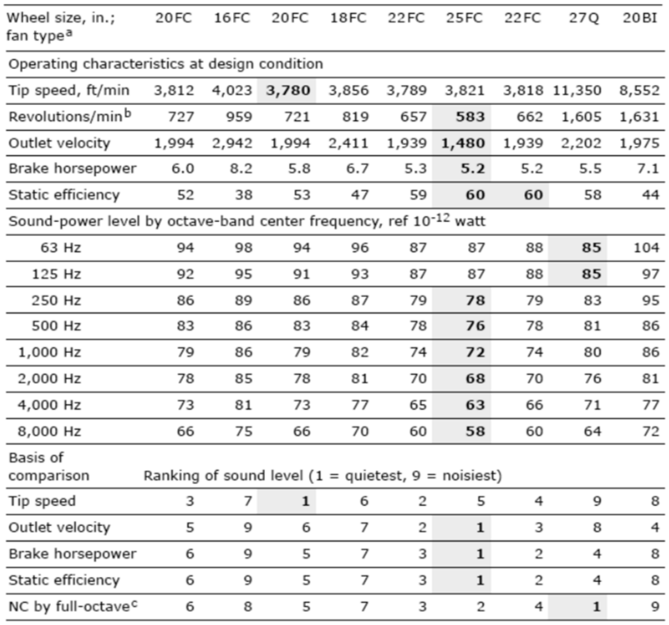
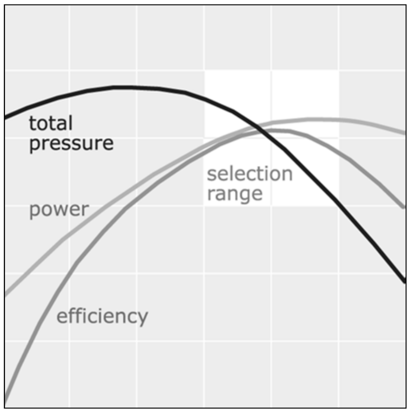
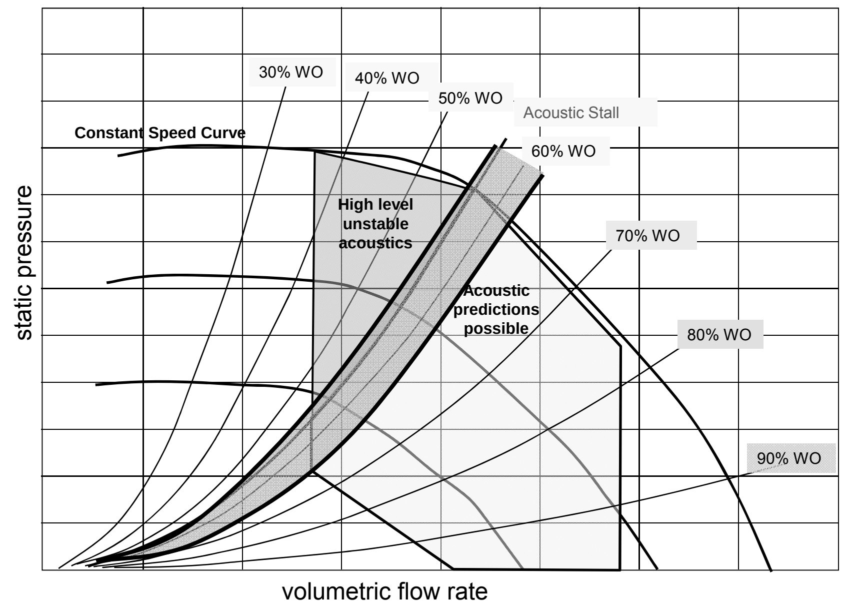


Wheel size, in.; fan type[a]	20FC	16FC	20FC	18FC	22FC	25FC	22FC	27Q	20BI
Operating characteristics at design condition									
Tip speed, ft/min	3,812	4,023	**3,780**	3,856	3,789	3,821	3,818	11,350	8,552
Revolutions/min[b]	727	959	721	819	657	**583**	662	1,605	1,631
Outlet velocity	1,994	2,942	1,994	2,411	1,939	**1,480**	1,939	2,202	1,975
Brake horsepower	6.0	8.2	5.8	6.7	5.3	**5.2**	5.2	5.5	7.1
Static efficiency	52	38	53	47	59	**60**	**60**	58	44
Sound-power level by octave-band center frequency, ref 10^{-12} watt									
63 Hz	94	98	94	96	87	87	88	**85**	104
125 Hz	92	95	91	93	87	87	88	**85**	97
250 Hz	86	89	86	87	79	**78**	79	83	95
500 Hz	83	86	83	84	78	**76**	78	81	86
1,000 Hz	79	86	79	82	74	**72**	74	80	86
2,000 Hz	78	85	78	81	70	**68**	70	76	81
4,000 Hz	73	81	73	77	65	**63**	66	71	77
8,000 Hz	66	75	66	70	60	**58**	60	64	72
Basis of comparison	Ranking of sound level (1 = quietest, 9 = noisiest)								
Tip speed	3	7	**1**	6	2	5	4	9	8
Outlet velocity	5	9	6	7	2	**1**	3	8	4
Brake horsepower	6	9	5	7	3	**1**	2	4	8
Static efficiency	6	9	5	7	3	**1**	2	4	8
NC by full-octave[c]	6	8	5	7	3	2	4	**1**	9

Rules of Thumb

- Lower tip speed *does not* equal lower sound
- Improved efficiency does result in lower sound





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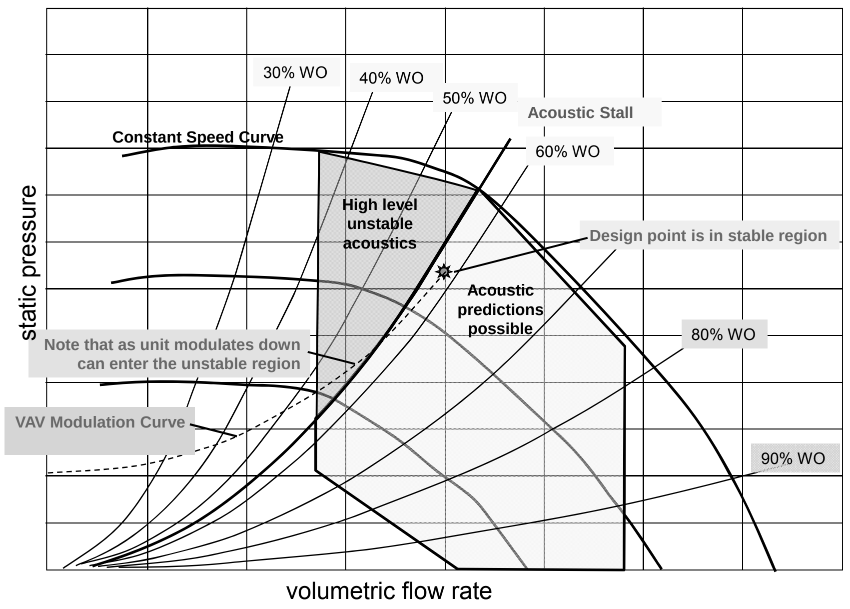

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Selection tips

- Accurate sound data is a must
- Review all fan and unit options
- Avoid “rules-of-thumb”

Fans in Air-Handling Units

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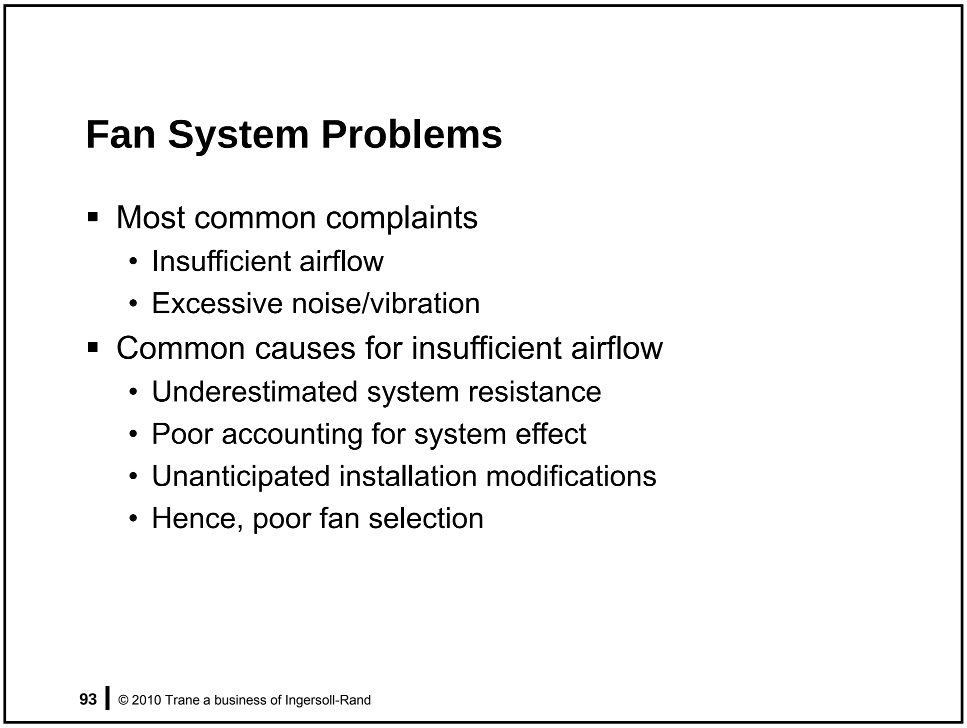
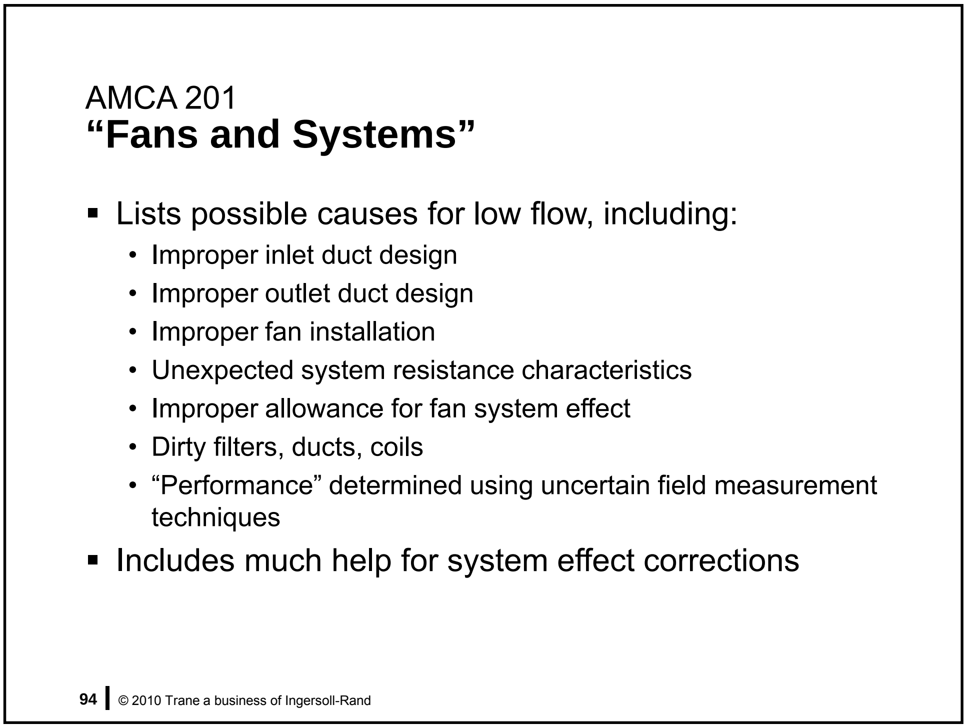
Common Problems:

Not Delivering Enough Airflow



Fan System Problems

- Most common complaints
  - Insufficient airflow
  - Excessive noise/vibration
- Common causes for insufficient airflow
  - Underestimated system resistance
  - Poor accounting for system effect
  - Unanticipated installation modifications
  - Hence, poor fan selection



AMCA 201

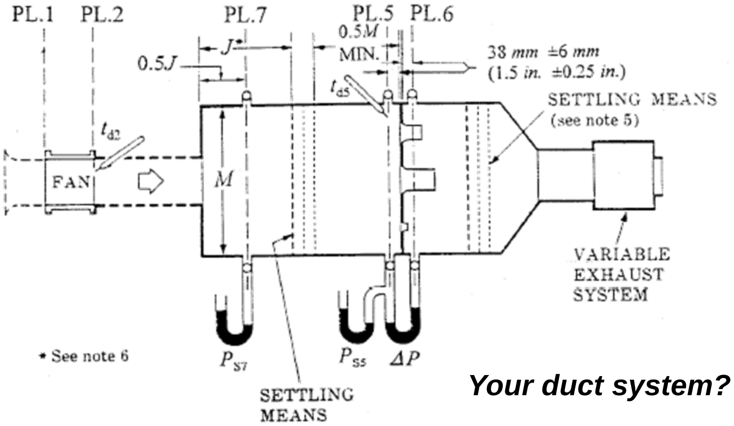
“Fans and Systems”

- 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 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

AMCA 203
“Field Performance Measurement of Fan Systems”



Your duct system?

Fans in Air-Handling Units

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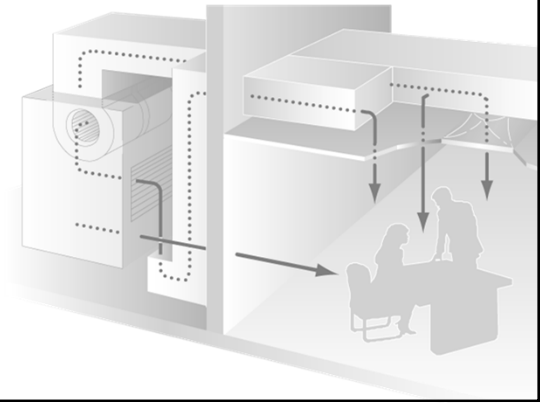
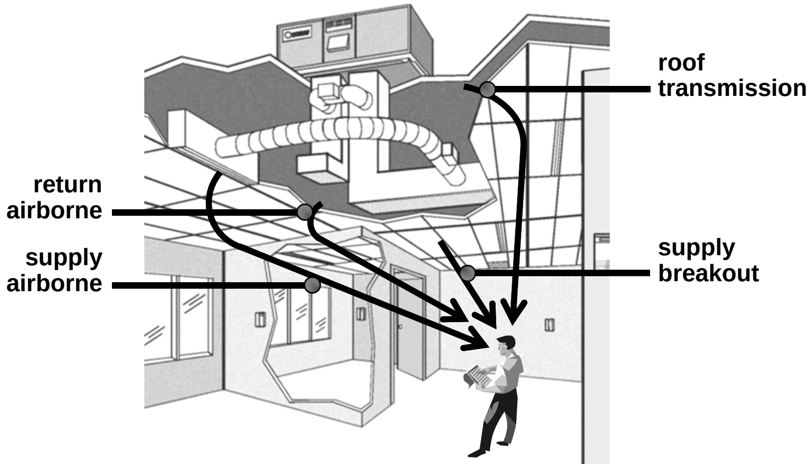
Common Problems:
Too Much Noise

Causes of Noise

- Fan / unit defect
- Acoustics ignored during selection
- Duct system flow problems



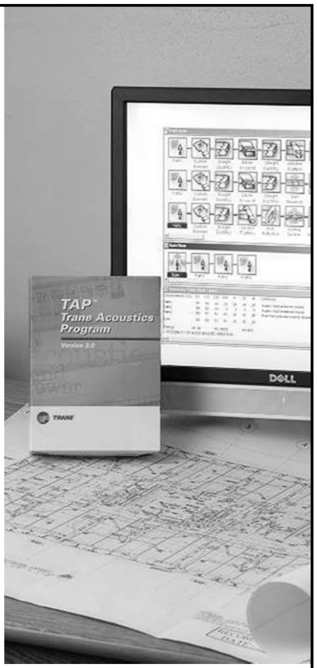
Sound Transmission Paths



Acoustical analysis: Source–path–receiver model

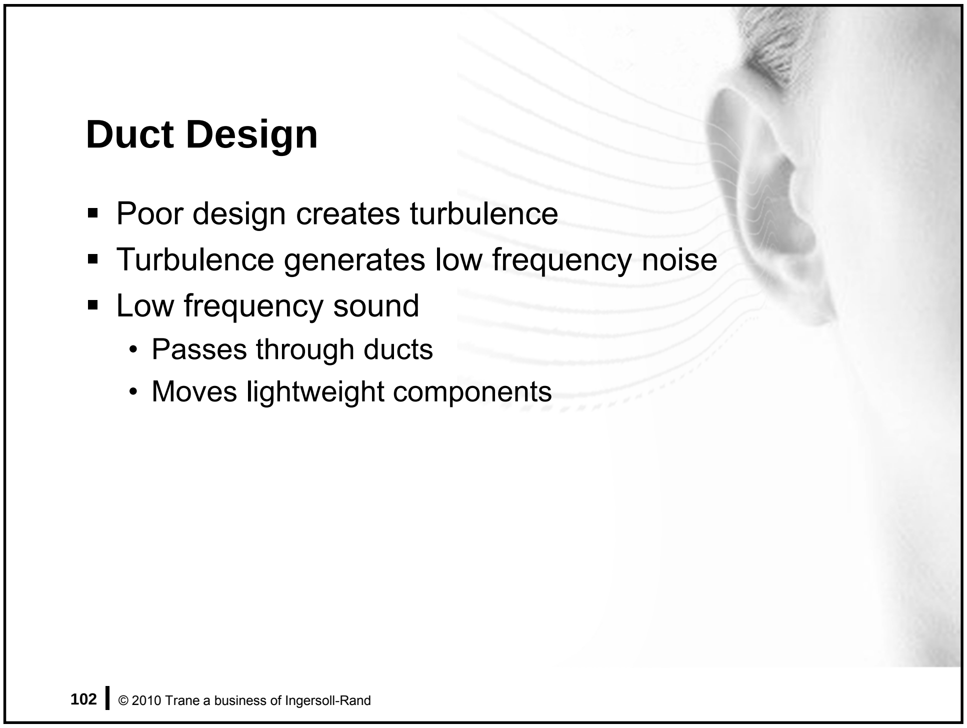
Path Analysis Tools

- Determine building acoustics
- Use to select equipment



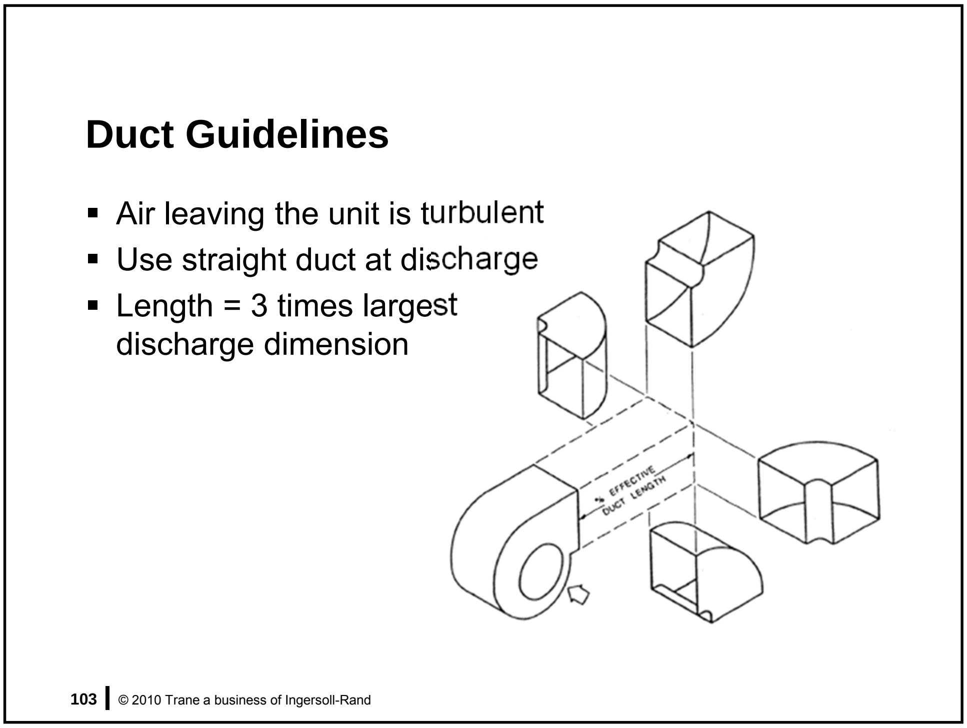
Duct Design

- ASHRAE algorithms
- Available for common duct components
- Used to predict acoustic impact



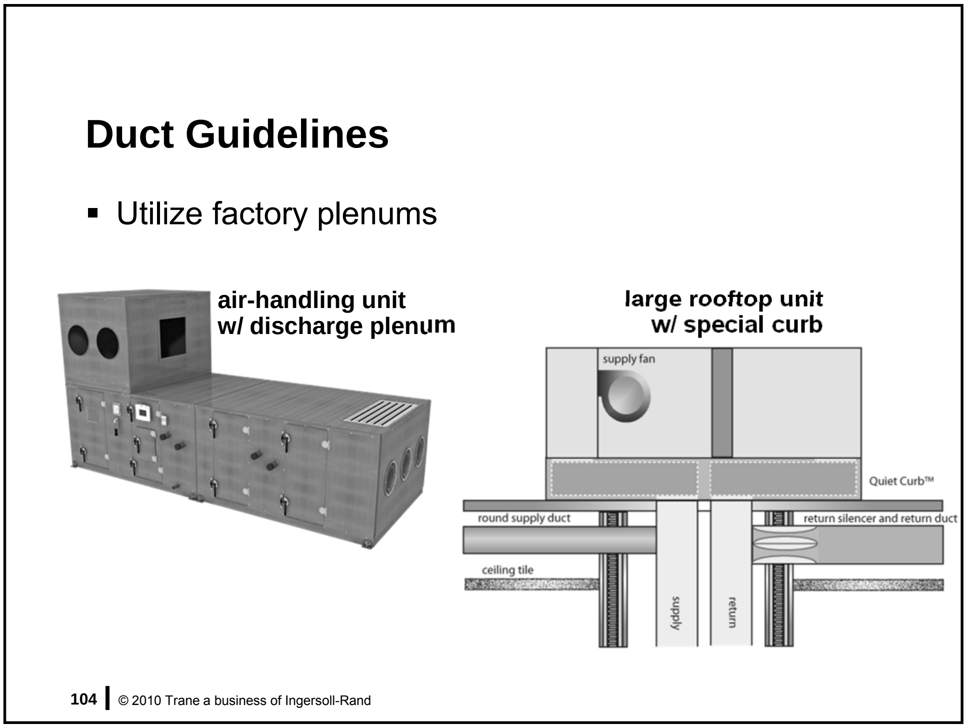
Duct Design

- Poor design creates turbulence
- Turbulence generates low frequency noise
- Low frequency sound
  - Passes through ducts
  - Moves lightweight components



Duct Guidelines

- Air leaving the unit is turbulent
- Use straight duct at discharge
- Length = 3 times largest discharge dimension

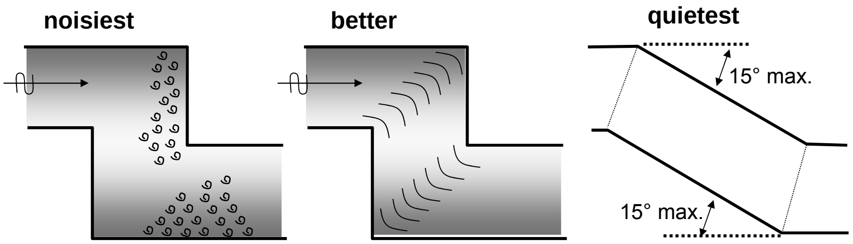


Duct Guidelines

- Utilize factory plenums

Duct Guidelines

- Avoid close coupled fittings



Source: ***A Practical Guide To Noise and Vibration Control For HVAC Systems*****, ASHRAE, 1991. Figure 1-23**

Summary

- Successful acoustics requires
  - Building analysis
  - Equipment selection
  - Duct design

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ASHRAE 90.1 Requirements

ASHRAE Standard 90.1-2007

Fan System Power Limitation

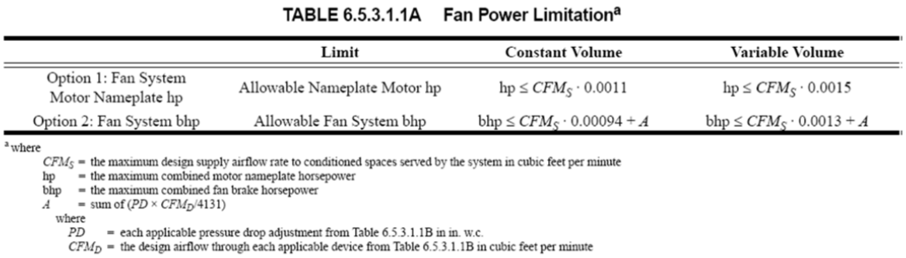


TABLE 6.5.3.1.1A Fan Power Limitation[a]

	Limit	Constant Volume	Variable Volume
Option 1: Fan System Motor Nameplate hp	Allowable Nameplate Motor hp	$\text{hp} \le CFM_S \cdot 0.0011$	$\text{hp} \le CFM_S \cdot 0.0015$
Option 2: Fan System bhp	Allowable Fan System bhp	$\text{bhp} \le CFM_S \cdot 0.00094 + A$	$\text{bhp} \le CFM_S \cdot 0.0013 + A$

[a] where

CFM_S = the maximum design supply airflow rate to conditioned spaces served by the system in cubic feet per minute

hp = the maximum combined motor nameplate horsepower

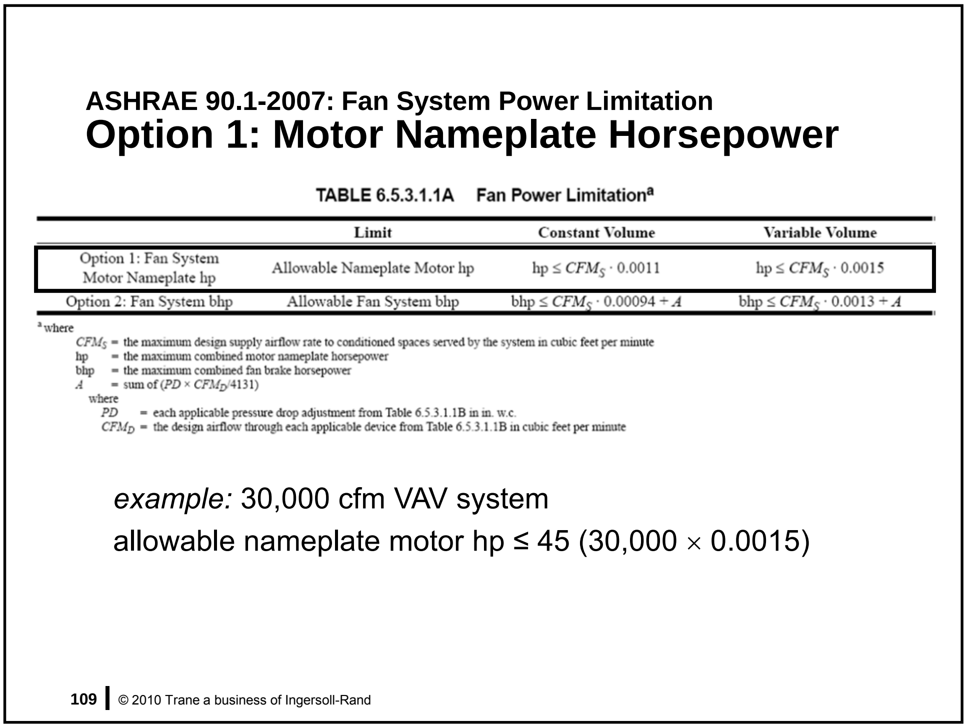
bhp = the maximum combined fan brake horsepower

A = sum of ($PD \times CFM_D/4131$)

where

PD = each applicable pressure drop adjustment from Table 6.5.3.1.1B in in. w.c.

CFM_D = the design airflow through each applicable device from Table 6.5.3.1.1B in cubic feet per minute



ASHRAE 90.1-2007: Fan System Power Limitation

Option 1: Motor Nameplate Horsepower

TABLE 6.5.3.1.1A Fan Power Limitation[a]

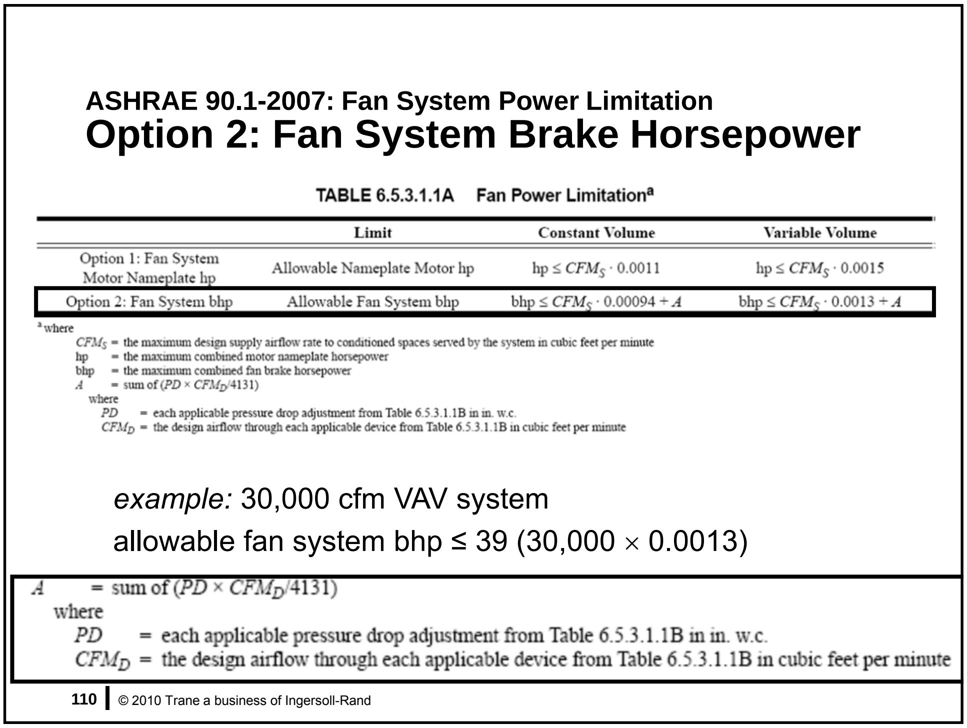
	Limit	Constant Volume	Variable Volume
Option 1: Fan System Motor Nameplate hp	Allowable Nameplate Motor hp	hp ≤ $CFM_S \cdot 0.0011$	hp ≤ $CFM_S \cdot 0.0015$
Option 2: Fan System bhp	Allowable Fan System bhp	bhp ≤ $CFM_S \cdot 0.00094 + A$	bhp ≤ $CFM_S \cdot 0.0013 + A$

[a] where

CFM_S = the maximum design supply airflow rate to conditioned spaces served by the system in cubic feet per minute
hp = the maximum combined motor nameplate horsepower
bhp = the maximum combined fan brake horsepower
A = sum of $(PD \times CFM_D/4131)$
where
PD = each applicable pressure drop adjustment from Table 6.5.3.1.1B in in. w.c.
CFM_D = the design airflow through each applicable device from Table 6.5.3.1.1B in cubic feet per minute

example: 30,000 cfm VAV system

allowable nameplate motor hp ≤ 45 (30,000 × 0.0015)



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Option 2: Fan System Brake Horsepower

TABLE 6.5.3.1.1A Fan Power Limitation[a]

	Limit	Constant Volume	Variable Volume
Option 1: Fan System Motor Nameplate hp	Allowable Nameplate Motor hp	hp ≤ $CFM_S \cdot 0.0011$	hp ≤ $CFM_S \cdot 0.0015$
Option 2: Fan System bhp	Allowable Fan System bhp	bhp ≤ $CFM_S \cdot 0.00094 + A$	bhp ≤ $CFM_S \cdot 0.0013 + A$

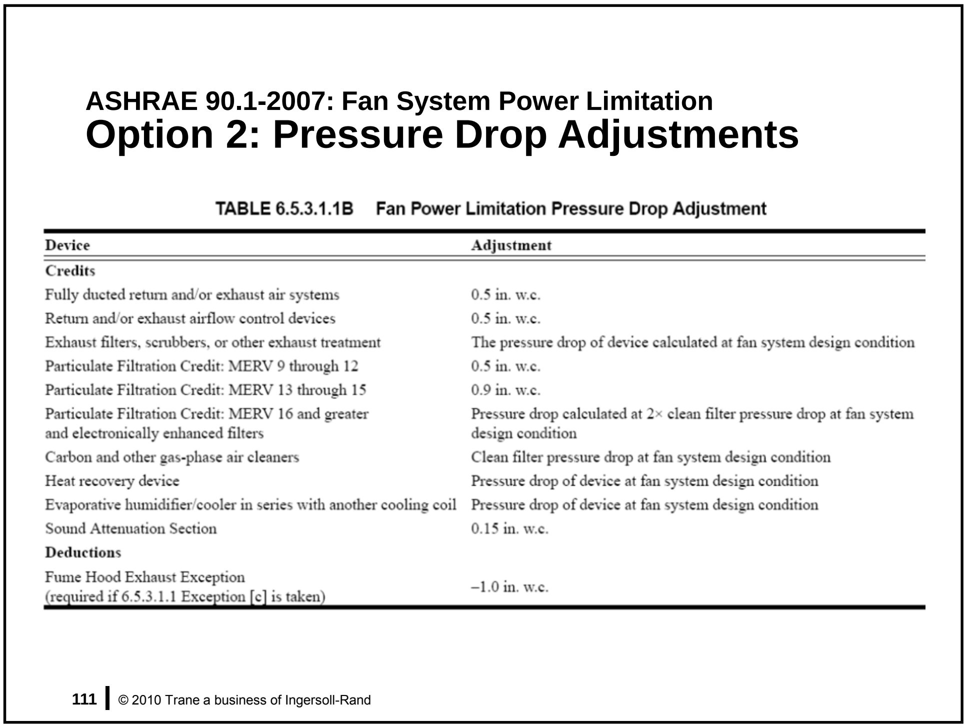
[a] where

CFM_S = the maximum design supply airflow rate to conditioned spaces served by the system in cubic feet per minute
hp = the maximum combined motor nameplate horsepower
bhp = the maximum combined fan brake horsepower
A = sum of $(PD \times CFM_D/4131)$
where
PD = each applicable pressure drop adjustment from Table 6.5.3.1.1B in in. w.c.
CFM_D = the design airflow through each applicable device from Table 6.5.3.1.1B in cubic feet per minute

example: 30,000 cfm VAV system

allowable fan system bhp ≤ 39 (30,000 × 0.0013)

A = sum of $(PD \times CFM_D/4131)$
where
PD = each applicable pressure drop adjustment from Table 6.5.3.1.1B in in. w.c.
CFM_D = the design airflow through each applicable device from Table 6.5.3.1.1B in cubic feet per minute

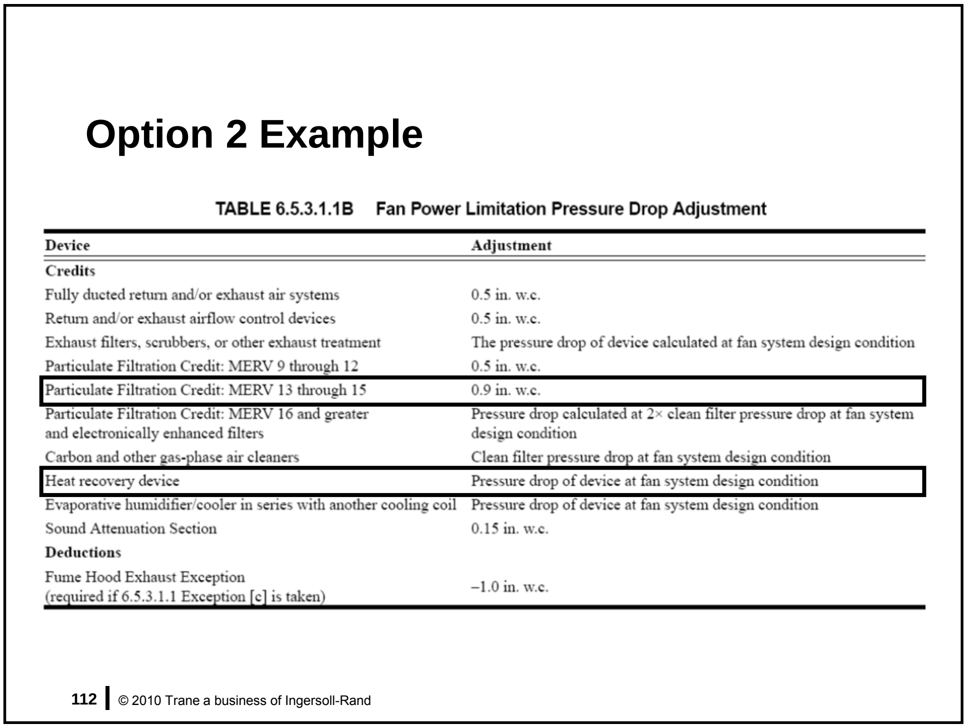


ASHRAE 90.1-2007: Fan System Power Limitation

Option 2: Pressure Drop Adjustments

TABLE 6.5.3.1.1B Fan Power Limitation Pressure Drop Adjustment

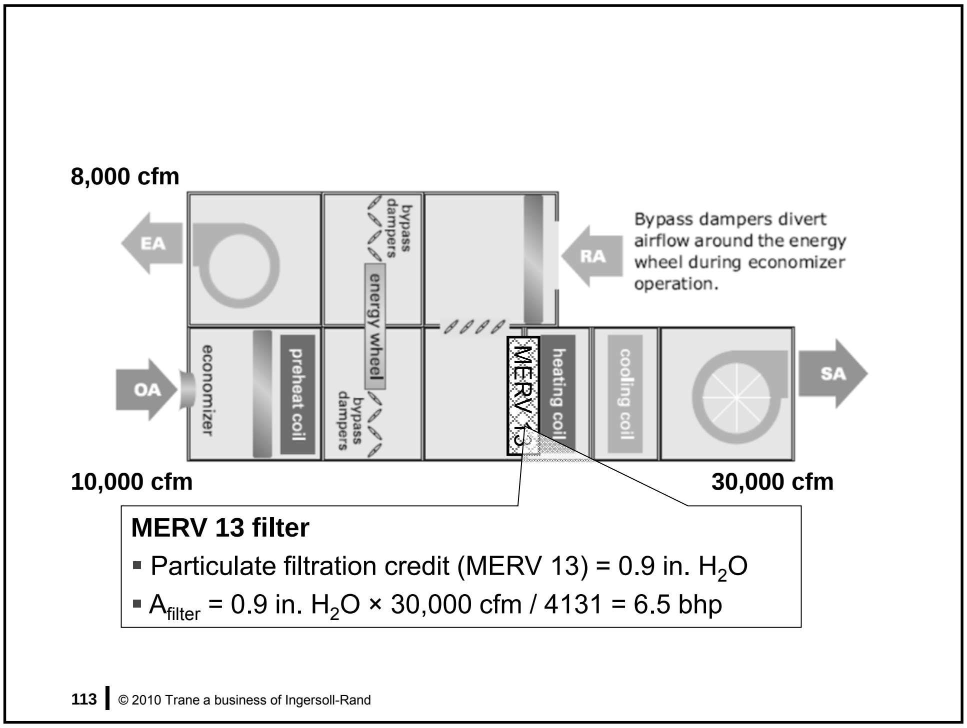
Device	Adjustment
Credits	
Fully ducted return and/or exhaust air systems	0.5 in. w.c.
Return and/or exhaust airflow control devices	0.5 in. w.c.
Exhaust filters, scrubbers, or other exhaust treatment	The pressure drop of device calculated at fan system design condition
Particulate Filtration Credit: MERV 9 through 12	0.5 in. w.c.
Particulate Filtration Credit: MERV 13 through 15	0.9 in. w.c.
Particulate Filtration Credit: MERV 16 and greater and electronically enhanced filters	Pressure drop calculated at 2× clean filter pressure drop at fan system design condition
Carbon and other gas-phase air cleaners	Clean filter pressure drop at fan system design condition
Heat recovery device	Pressure drop of device at fan system design condition
Evaporative humidifier/cooler in series with another cooling coil	Pressure drop of device at fan system design condition
Sound Attenuation Section	0.15 in. w.c.
Deductions	
Fume Hood Exhaust Exception (required if 6.5.3.1.1 Exception [c] is taken)	–1.0 in. w.c.



Option 2 Example

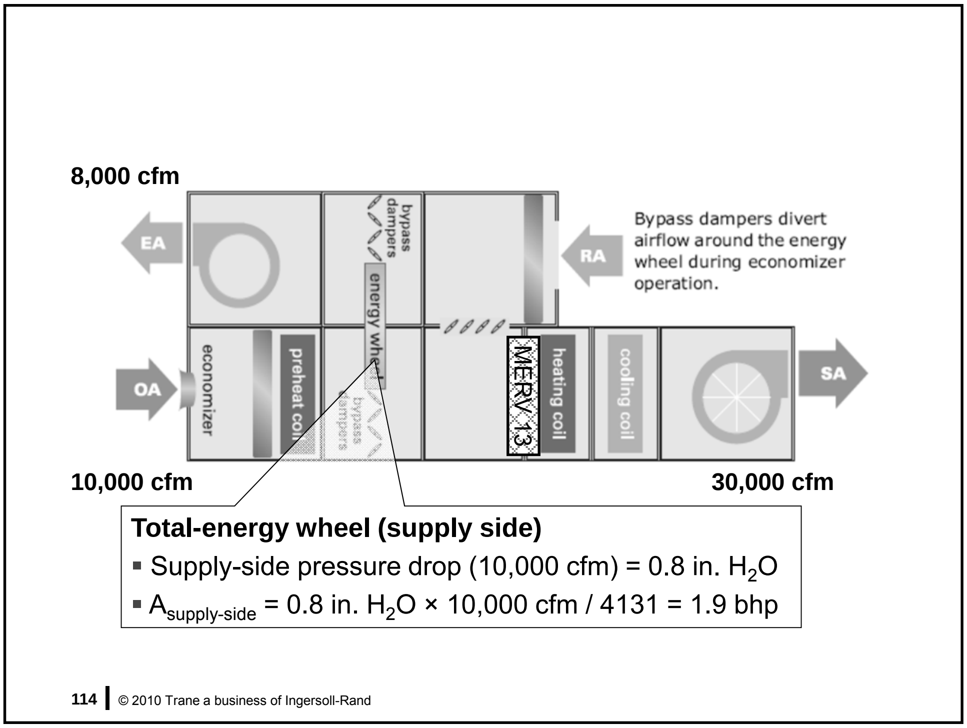
TABLE 6.5.3.1.1B Fan Power Limitation Pressure Drop Adjustment

Device	Adjustment
Credits	
Fully ducted return and/or exhaust air systems	0.5 in. w.c.
Return and/or exhaust airflow control devices	0.5 in. w.c.
Exhaust filters, scrubbers, or other exhaust treatment	The pressure drop of device calculated at fan system design condition
Particulate Filtration Credit: MERV 9 through 12	0.5 in. w.c.
Particulate Filtration Credit: MERV 13 through 15	0.9 in. w.c.
Particulate Filtration Credit: MERV 16 and greater and electronically enhanced filters	Pressure drop calculated at 2× clean filter pressure drop at fan system design condition
Carbon and other gas-phase air cleaners	Clean filter pressure drop at fan system design condition
Heat recovery device	Pressure drop of device at fan system design condition
Evaporative humidifier/cooler in series with another cooling coil	Pressure drop of device at fan system design condition
Sound Attenuation Section	0.15 in. w.c.
Deductions	
Fume Hood Exhaust Exception (required if 6.5.3.1.1 Exception [c] is taken)	–1.0 in. w.c.



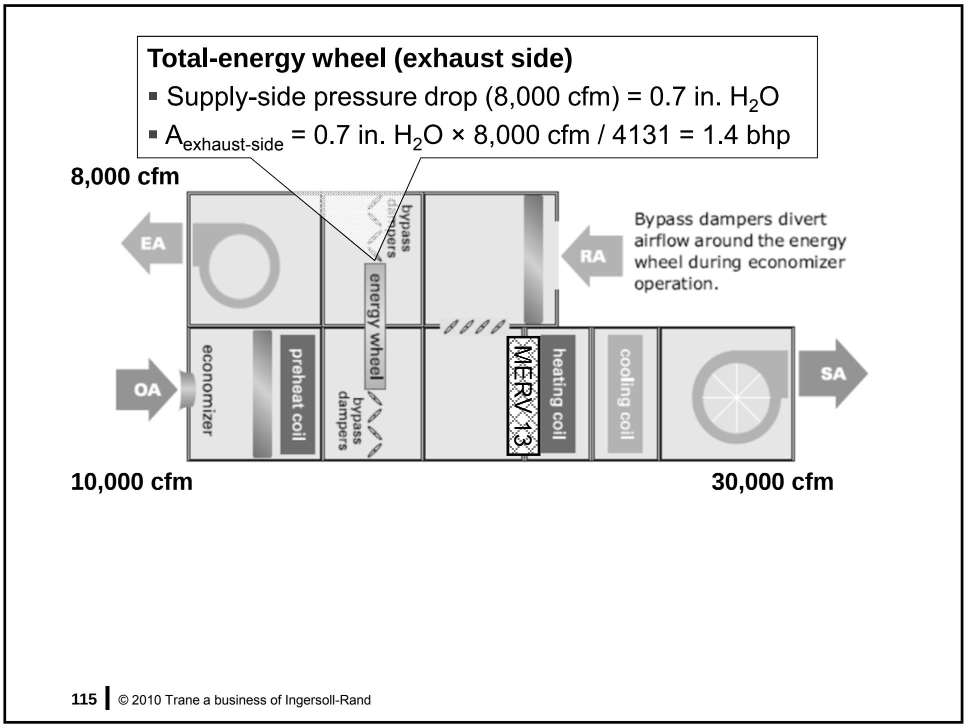
MERV 13 filter

- Particulate filtration credit (MERV 13) = 0.9 in. H_2O
- A_{filter} = 0.9 in. H_2O × 30,000 cfm / 4131 = 6.5 bhp



Total-energy wheel (supply side)

- Supply-side pressure drop (10,000 cfm) = 0.8 in. H_2O
- $A_{supply\text{-}side}$ = 0.8 in. H_2O × 10,000 cfm / 4131 = 1.9 bhp



Total-energy wheel (exhaust side)

- Supply-side pressure drop (8,000 cfm) = 0.7 in. H_2O
- $A_{exhaust\text{-}side}$ = 0.7 in. H_2O × 8,000 cfm / 4131 = 1.4 bhp

Bypass dampers divert airflow around the energy wheel during economizer operation.

Option 2 Example

MERV 13 filter

- A_{filter} = 0.9 in. H_2O × 30,000 cfm / 4131 = 6.5 bhp

Total-energy wheel

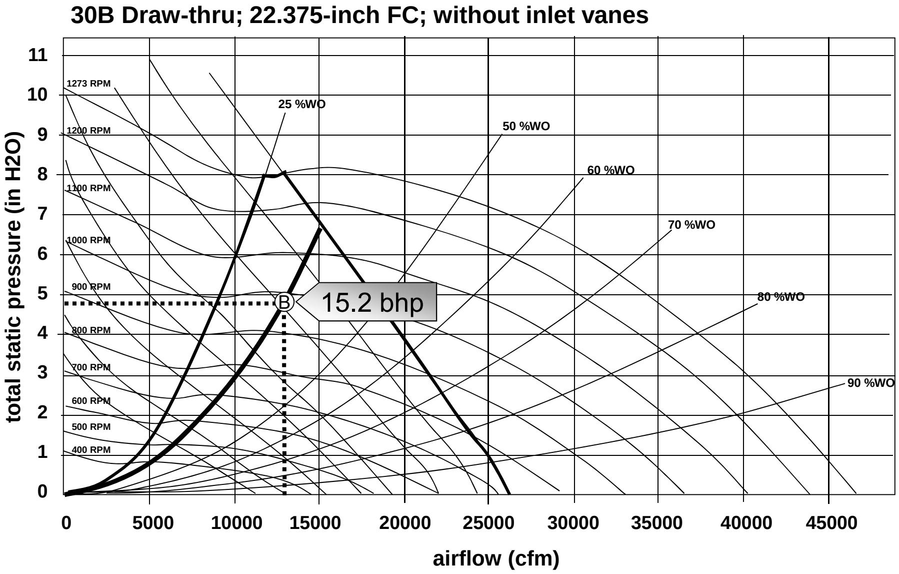
- $A_{supply\text{-}side}$ = 0.8 in. H_2O × 10,000 cfm / 4131 = 1.9 bhp
- $A_{exhaust\text{-}side}$ = 0.7 in. H_2O × 8,000 cfm / 4131 = 1.4 bhp

A = 6.5 + 1.9 + 1.4 = 9.8 bhp

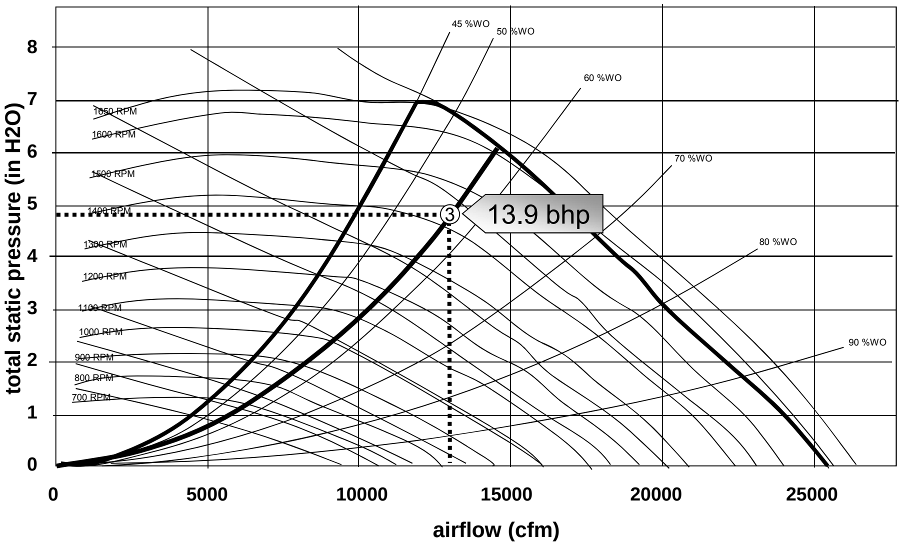
allowable fan system bhp ≤ 48.8 (30,000 × 0.0013 + 9.8)

Ways to Reduce Fan Power

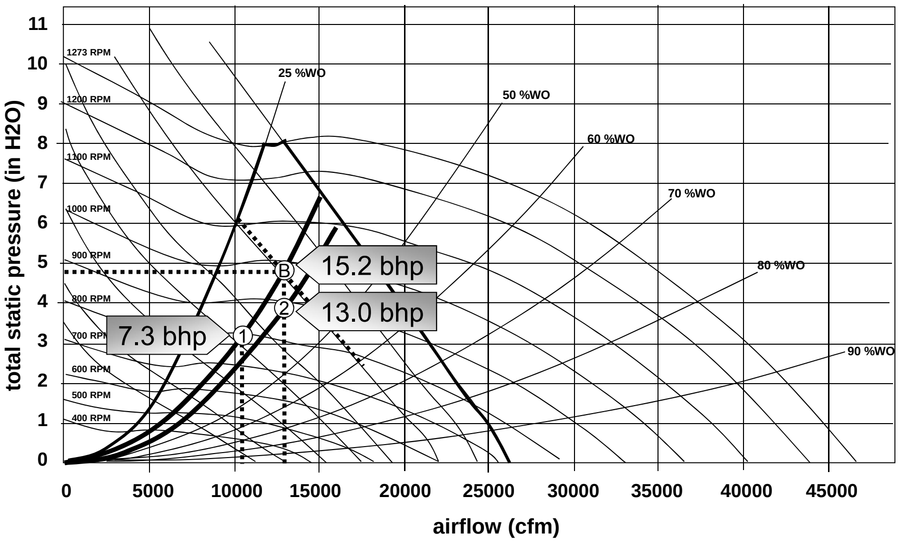
1. Reduce airflow
   - Reduce cooling loads (better envelope, fewer and better windows, more efficient lighting)
   - Colder supply-air temperature
2. Reduce airside pressure loss
   - Efficient duct fittings
   - Larger ductwork
   - Larger air-handling unit
   - Low pressure drop filters and coils
3. Select a higher-efficiency fan (if you have the choice)



D30 Draw-thru; 25-inch AF; without inlet vanes



30B Draw-thru; 22.375-inch FC; without inlet vanes



example
Ways to Reduce Fan Power

	Baseline fan selection	15.2 bhp
①	Reduce airflow (colder air)	7.3 bhp
②	Reduce airside pressure loss	13.0 bhp
③	Selecting a higher-efficiency fan	13.9 bhp
	Implement all three	5.7 bhp

summary
ASHRAE 90.1 Fan Power Limitation

- Prescriptive limits apply to sum of all fans that operate at peak design conditions
- Two options for compliance:
  - Option 1 (nameplate power) is simpler
  - Option 2 (brake horsepower) is more flexible, but be sure to make use of the adjustments
- To reduce fan power:
  - Reduce airflow (reduce loads, colder supply air)
  - Reduce airside pressure loss
  - Select a higher-efficiency fan

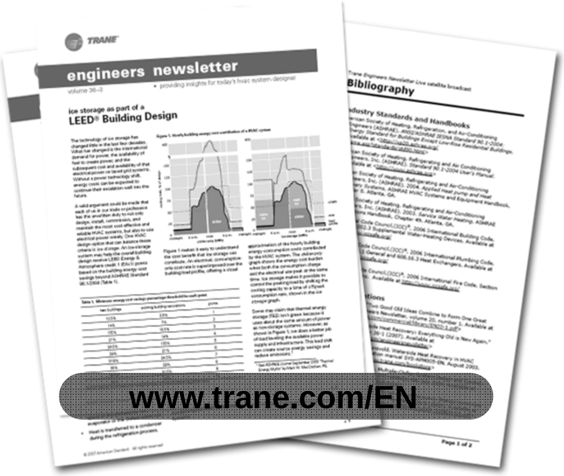
summary

Fans in Air-Handling Systems

- The right fan depends on the application, and is often based on balancing efficiency, acoustics, and cost.
- It is important to understand how the fan will interact within the system.
  - Dirty filters and wet cooling coils
  - Fan modulation in a VAV system
  - System effect
- Sound data taken in accordance with AHRI 260 provides the best indication of sound produced by the entire air-handling unit.

References for This Broadcast

Where to Learn More



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