Best Practices for Applied Rooftop Systems, Applications and Installation

Jerry Cohen
President
Jacco & Assoc.

Who is Jacco

- Established 1968
  - Hudson, Ohio
  - Columbus, Ohio
  - Toledo, Ohio
- Focused on the Engineered Environment
  - Systems Knowledgeable
    - HVAC Systems
    - Service & Maintenance
    - Parts

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Purpose Statement

The purpose of our Company is to solve our customers problems, in the most economical way, at all times optimizing the owning experience.

Full Circle Support

Who is Jacco

- Owning Experience Operations
  - –Brenda Homjak
  - –Mike Spangler
  - –Chad Russell

Who is Jacco

- Owning Experience Construction
  - –Elyse Perry
  - –Maggie Sawicki

Who is Jacco

- Owning Experience Engineering
  - Greg Drensky
  - Jerry Cohen

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Who is Jacco

- 30 Minute Design
  - –Unit Performance
  - –Drawing
  - –Weights
  - –Electrical
  - –Specifications?
  - –Sequence of Operation?
  - –Cartoon?
  - –Narrative?

Who is Jacco

2015 Seminars

Seminars	Instructor	Date
Psychrometrics	JKC	14-Jan
The Refrigeration Cycle	JKC	11-Feb
Energy Recovery	GAD	11-Mar
Applied Rooftop Systems	JKC	8-Apr
VRF Design & Installation	GAD	13-May
Geothermal Systems	GAD	10-Jun
Chilled Beam, Radiant Cooling & DOAS	JKC	12-Aug
Vertical Market Systems	GAD	9-Sep
Building Pressure & Air Flow Measurement	GAD	14-Oct
Controlling HVAC Systems - Sequence of Operations	JKC	11-Nov

Agenda

Define and relate in practical terms the following components:

- Supply, Return & Exhaust Fans
- Cooling & Heating Options
- Temperature Control Options
- Filtration Options
- Cabinet Options
- Psychrometrics of Motor Heat
- Best Installation Practices

Agenda

Explain common Applied Rooftop applications including:

- Variable Air Volume
- Single Zone Variable Air Volume
- Constant Air Volume

Questions for You

- Is a DX Rooftop as Efficient as a Chilled Water system?

- What Constant are all Rooftops Designed Around?

Fans or Compressors?

5%	of the unit's operation requires one hundred-percent mechanical cooling capacity,
20%	of the unit's operation is in heating mode,
30%	of the unit's operation is in an "unoccupied" mode that requires ten-percent mechanical cooling capacity
45%	of the time the machine is at a part load requiring an average fifty-percent mechanical cooling capacity.
100%	

Fan and Compressor Power Consumption

Peak Cooling Mode	Btu/hr	Watt	Horsepower
Compressor Input Power	69,130	20,260	27.2
Backward Curved Plenum Fan Power	23,465	6,876	9.2
Heating Mode	**Btu/hr**	**Watt**	**Horsepower**
Compressor Input Power	0	0	0
Backward Curved Plenum Fan Power	23,465	6,876	9.2
Unoccupied Mode	**Btu/hr**	**Watt**	**Horsepower**
Compressor Input Power	6,913	2,026	2.7
Backward Curved Plenum Fan Power	23,465	6,876	9.2
Part Load	**Btu/hr**	**Watt**	**Horsepower**
Compressor Input Power	34,565	10,130	13.6
Backward Curved Plenum Fan Power	23,465	6,876	9.2

Weighted Average Operating Fan and Compressor Power Consumption

Part Load	Btu/hr	Watt	Horsepower
Compressor Input Power	**21,085**	6,179	8.3
Backward Curved Plenum Fan Power	**23,465**	6,876	9.2

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Fan System Effect

- Fan *System Effects* occur because of the difference in inlet and outlet conditions under laboratory test conditions and the inlet and outlet conditions as the fan is installed in the system.

Fan System Effect

- Centrifugal and axial fans are usually tested with an outlet duct. Propeller fans are normally tested in the wall of a chamber or plenum. Power roof ventilators (PRV) are tested mounted on a curb exhausting from the test chamber.
- The System Effect includes **only** the effect of the system configuration on the fan's performance.
- ANSI/AMCA 210 specifies an outlet duct that is no greater than 105% or less than 95% of the fan outlet area. It also requires that the slope of the transition elements be no greater than 15° for converging elements or greater than 7° for diverging elements.
- Fan performance can be greatly affected by nonuniform or swirling inlet flow. Fan rating and catalog performance is typically obtained with unobstructed inlet flow. Any disruption to the inlet airflow will reduce a fan's performance. Restricted fan inlets located close to walls, obstructions or restrictions caused by a plenum or cabinet will also decrease the performance of a fan and add to the System Effect.

Fan System Effect

- Fans within plenums and cabinets or next to walls should be located so that air may flow unobstructed into the inlets. Fan performance is reduced if the space between the fan inlet and the enclosure is too restrictive. It is common practice to allow at least one-half impeller diameter between an enclosure wall and the fan inlet. Adjacent inlets of multiple double width centrifugal fans located in a common enclosure should be at least one impeller diameter apart if optimum performance is to be expected.

- Factory Supplied Accessories that have a System Effect.
  - Bearing and supports in fan inlet
  - Drive guards obstructing fan inlet
  - Belt tube in axial fan inlet or outlet
  - Inlet box
  - Inlet box dampers
  - Variable inlet vane (VIV)
  - Discharge dampers

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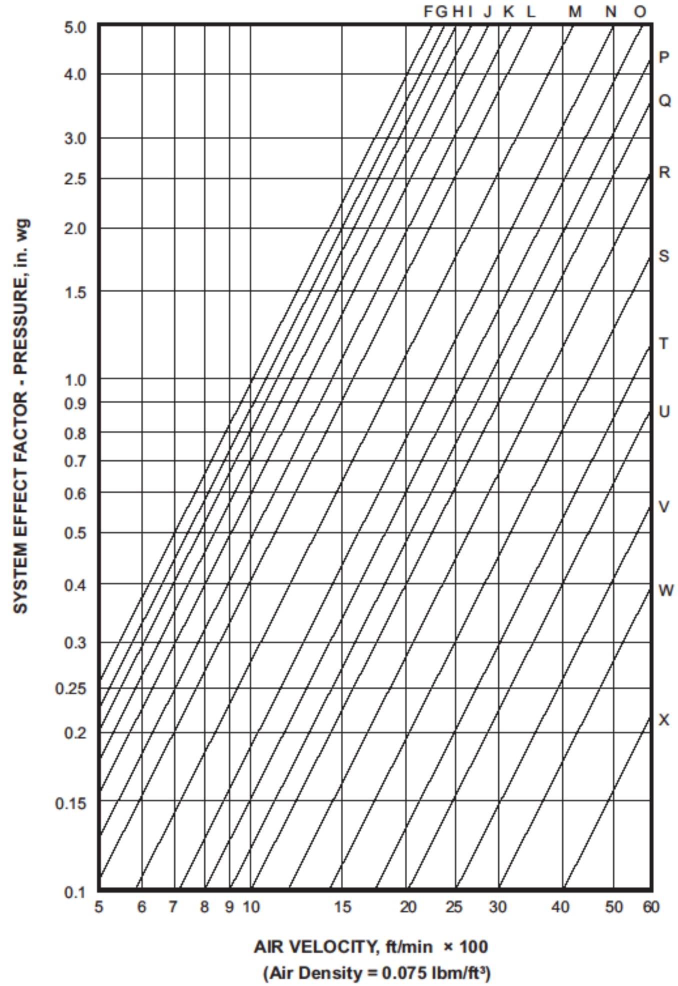
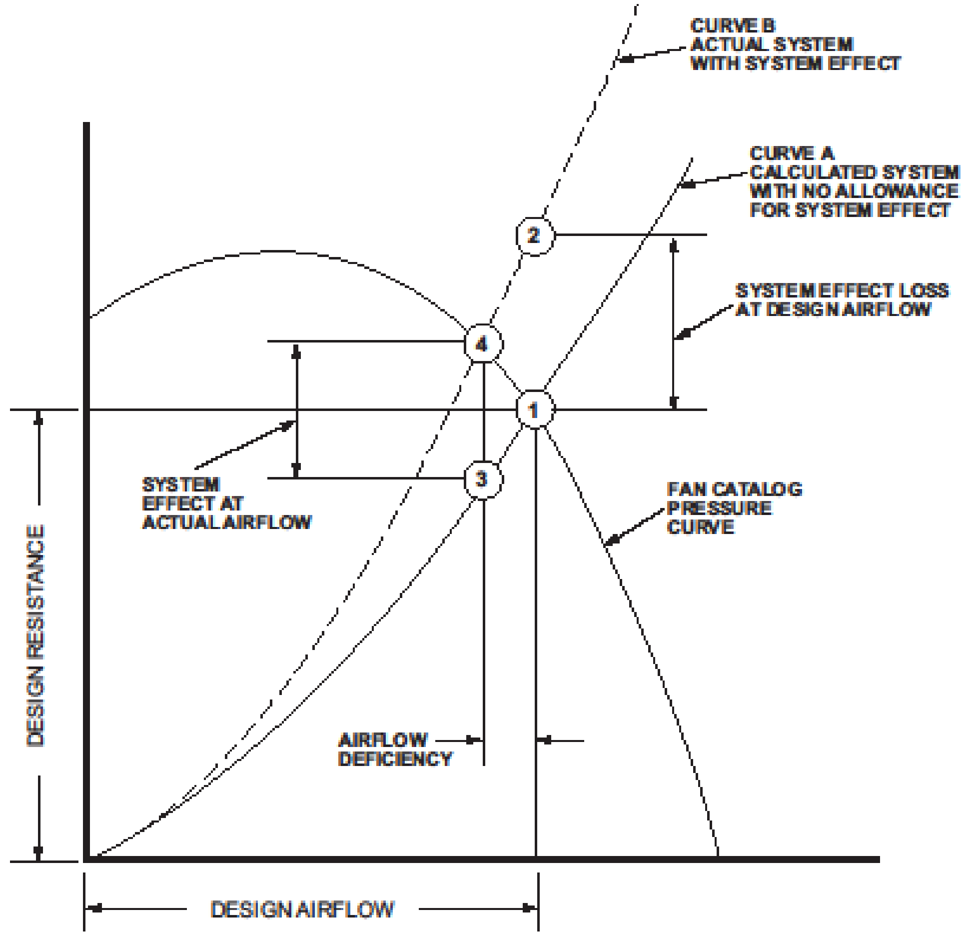
Fan System Effect

- Total Pressure = Static Pressure + Velocity Pressure
  - Static Regain Converts Velocity Pressure to Static Pressure
- System Effect is Velocity Dependent
- You CAN NOT Measure System Effect
- You CAN Calculate System Effect, called System Effect Factor

19 Fan System Effect Curves



Overcome Fan System Effect



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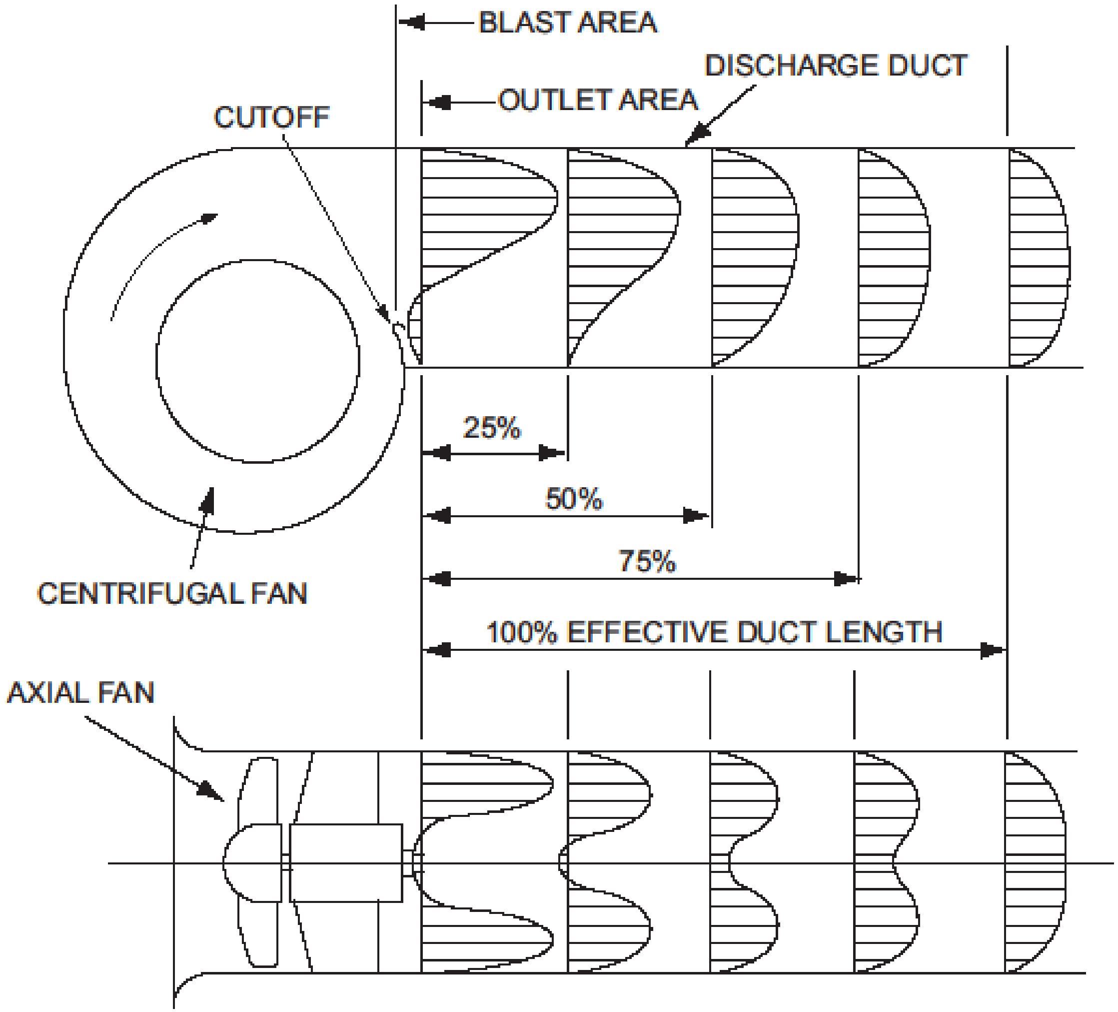
Outlet Requirements per AMCA

- If the outlet velocity is less than 2,500 fpm: 100 percent-effective duct length = 2.5 x Duct diameter
- If the outlet velocity is more than 2,500 fpm: 100 percent-effective duct length = fpm/1000 x Duct diameter

Outlet Conditions



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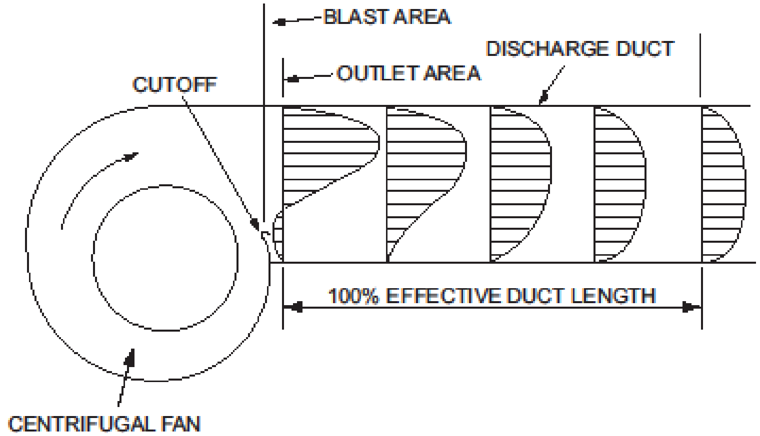
Centrifugal Fan Outlet Conditions



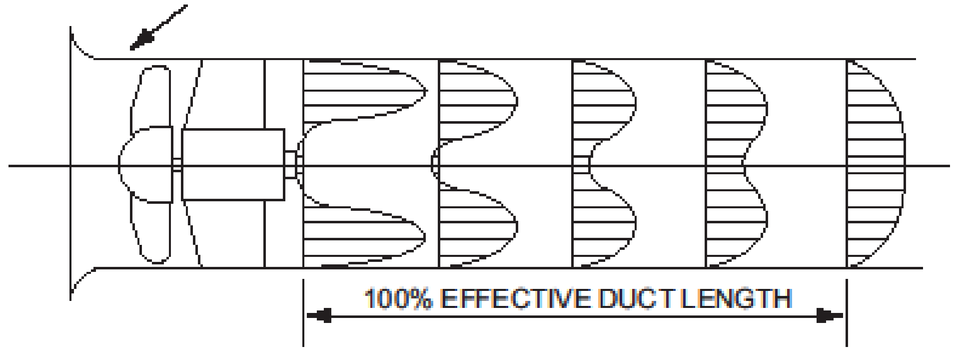
To calculate 100% duct length, assume a minimum of 2½ duct diameters for 2500 fpm or less. Add 1 duct diameter for each additional 1000 fpm.

EXAMPLE: 5000 fpm = 5 equivalent duct diameters. If the duct is rectangular with side dimensions *a* and *b*, the equivalent duct diameter is equal to $(4ab/\pi)^{0.5}$.

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Pressure Recovery	0%	50%	80%	90%	100%
Blast Area / Outlet Area	System Effect Curve				
0.4	P	R-S	U	W	—
0.5	P	R-S	U	W	—
0.6	R-S	S-T	U-V	W-X	—
0.7	S	U	W-X	—	—
0.8	T-U	V-W	X	—	—
0.9	V-W	W-X	—	—	—
1.0	—	—	—	—	—

Determine *SEF* by using Figure 7.1

Axial Fan Outlet Conditions



To calculate 100% duct length, assume a minimum of 2½ duct diameters for 12.7 m/s (2500 fpm) or less. Add 1 duct diameter for each additional 5.08 m/s (1000 fpm).

EXAMPLE: 25.4 m/s (5000 fpm) = 5 equivalent duct diameters

	No Duct	12% Effective Duct	25% Effective Duct	50 % Effective Duct	100% Effective Duct
Tubeaxial Fan	---	---	---	---	---
Vaneaxial Fan	U	V	W	---	---

Determine *SEF* by using Figure 7.1

Figure 8.2 - System Effect Curves for Outlet Ducts - Axial Fans

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Inlet Requirements per AMCA

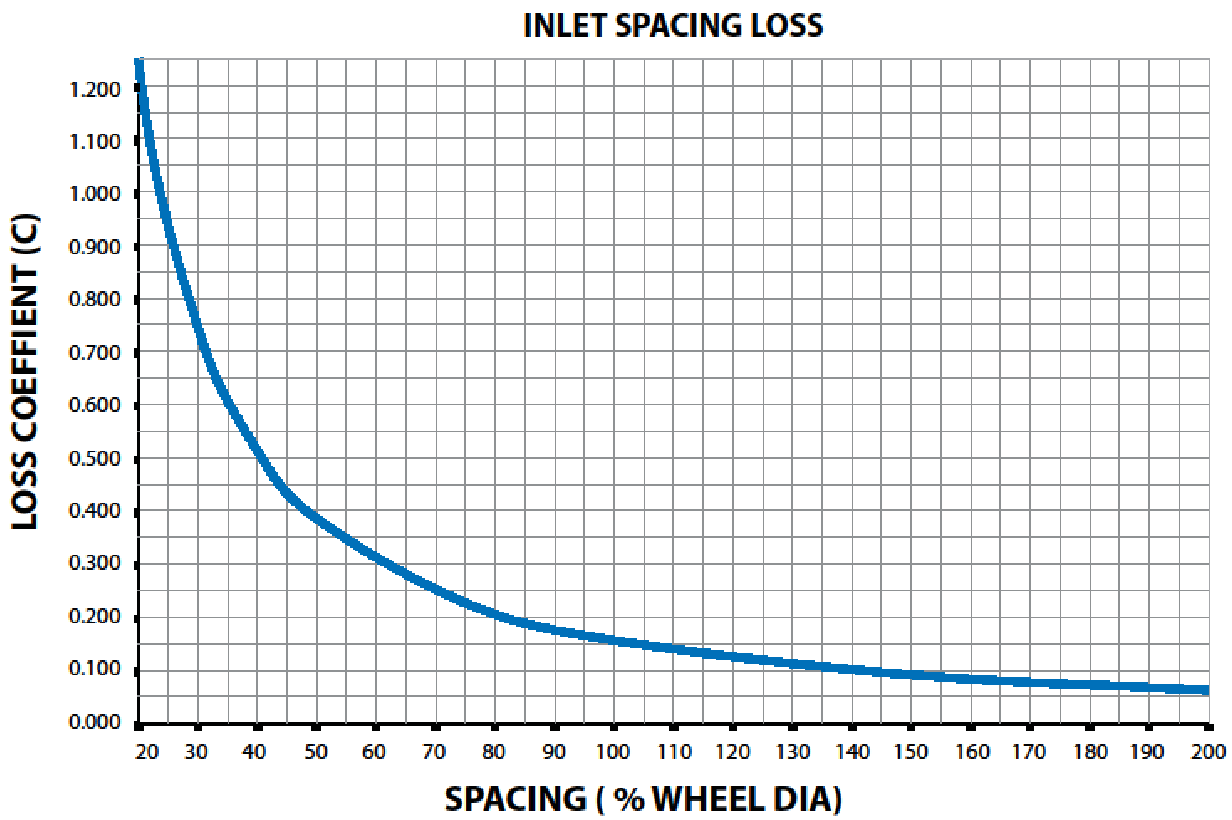

Fig. 27: Inlet Spacing Loss Coefficient

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Centrifugal Fan Inlet Conditions

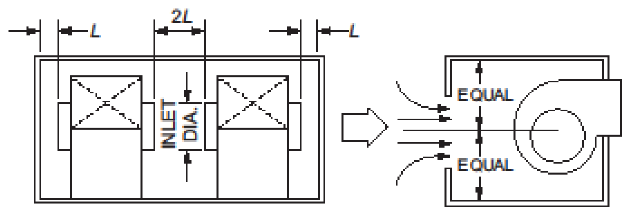

Figure 9.11A - Fans and Plenum

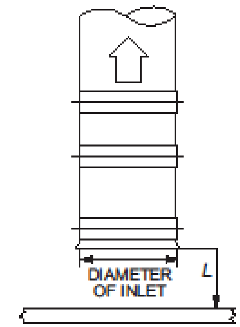

Figure 9.11B - Axial Fan Near Wall

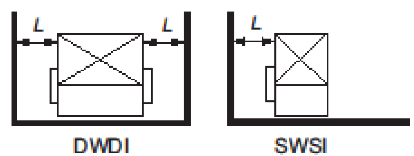

Figure 9.11C - Centrifugal Fan Near Wall(s)

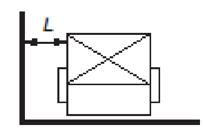

Figure 9.11D - DWDI Fan Near Wall on One Side

L - DISTANCE INLET TO WALL	For Figures 9.11A, B & C SYSTEM EFFECT CURVES	For Figures 9.11D SYSTEM EFFECT CURVES
0.75 x DIA. OF INLET	V-W	X
0.50 x DIA. OF INLET	U	V-W
0.40 x DIA. OF INLET	T	V-W
0.30 x DIA. OF INLET	S	U

Determine SEF by calculating inlet velocity and using Figure 7.1

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Inlet Conditions

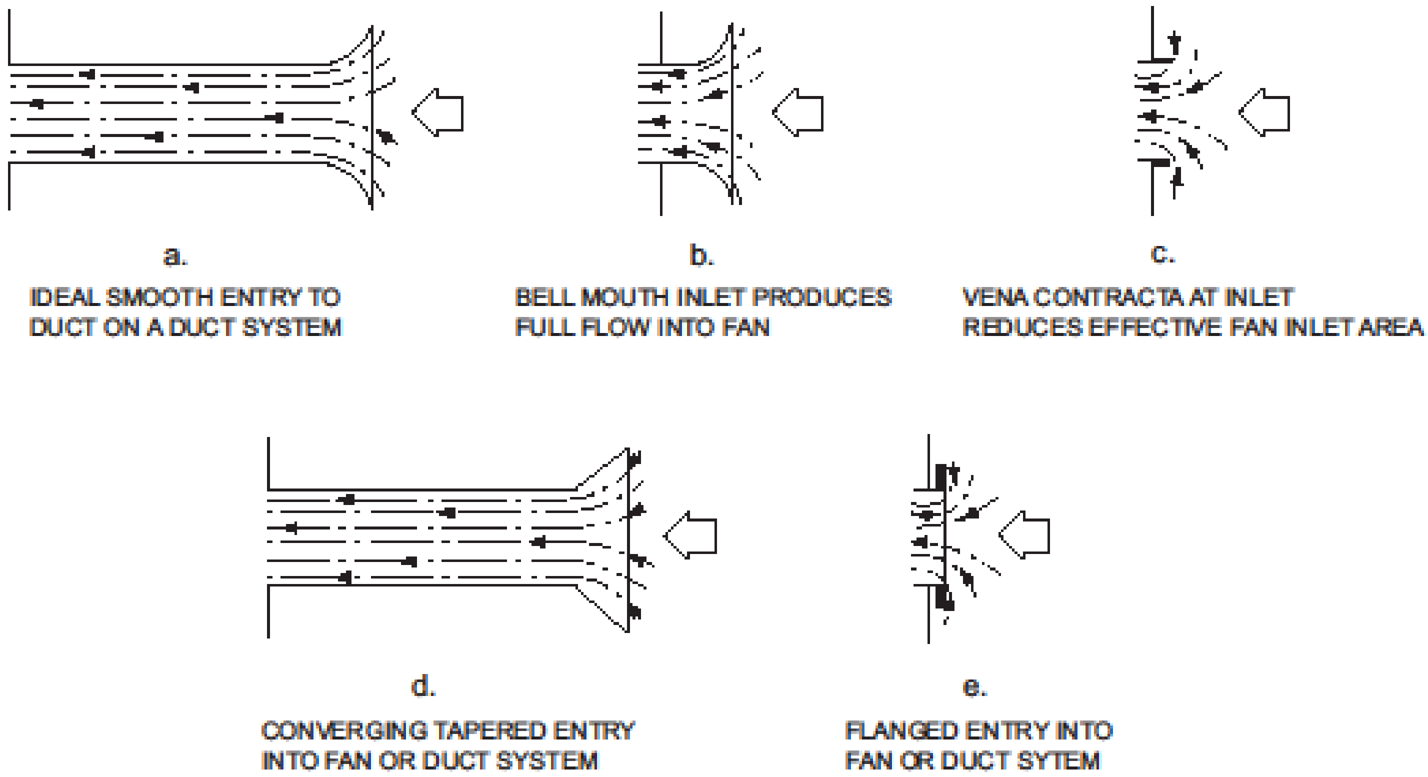

Figure 9.1 Typical Inlet Connections for Centrifugal and Axial Fans

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Belts & Sheaves

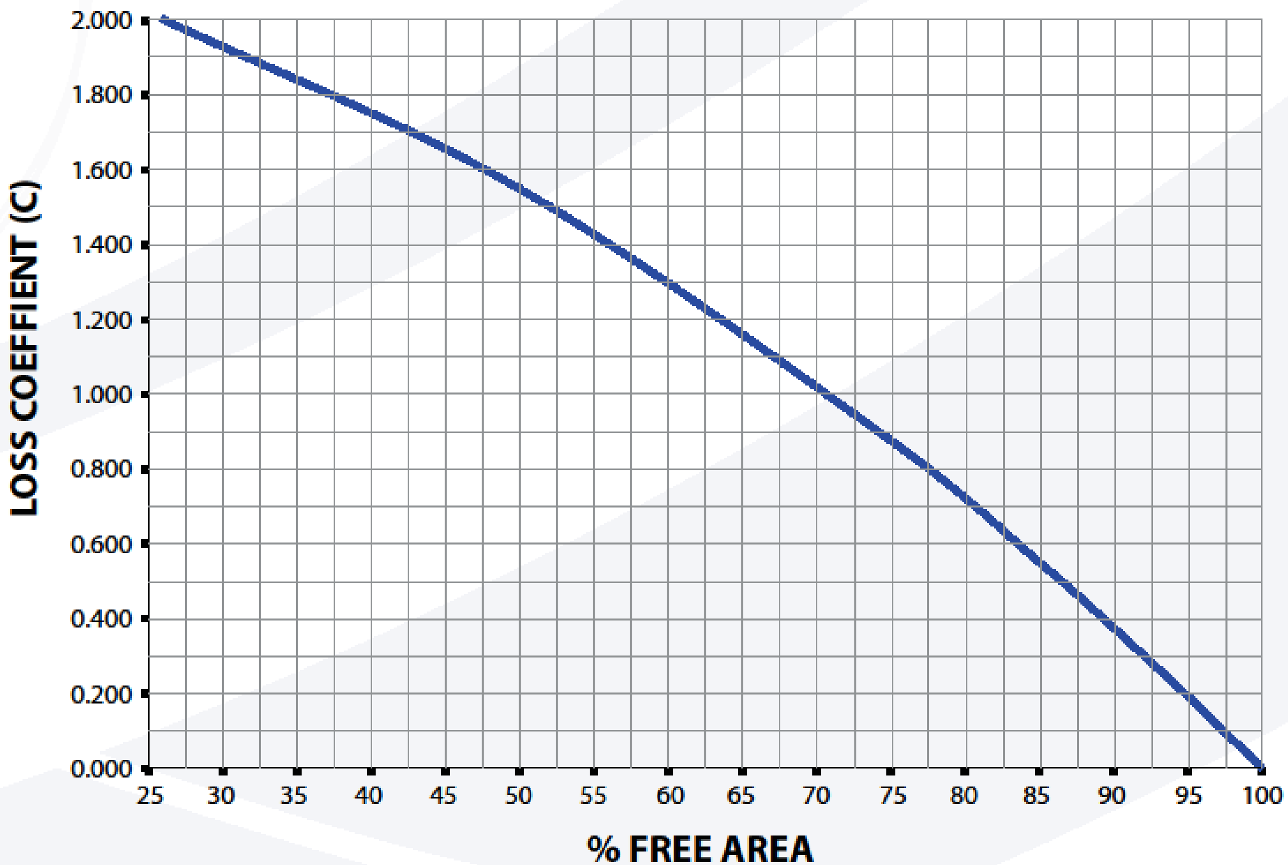

Fig. 29: Inlet Free Area Reduction Loss

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Discharge Elbows and Tees

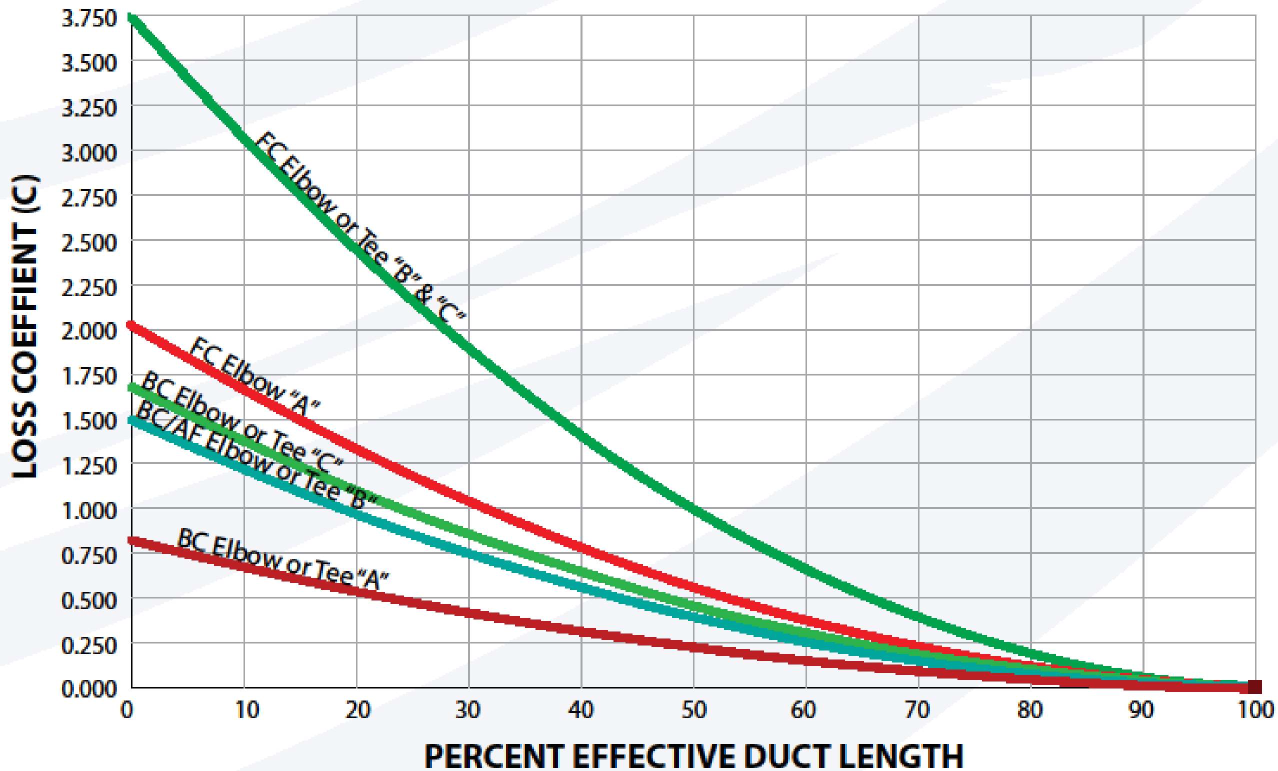


Fig. 31: Housed Discharge Elbows and Tee

Unducted Discharge Losses

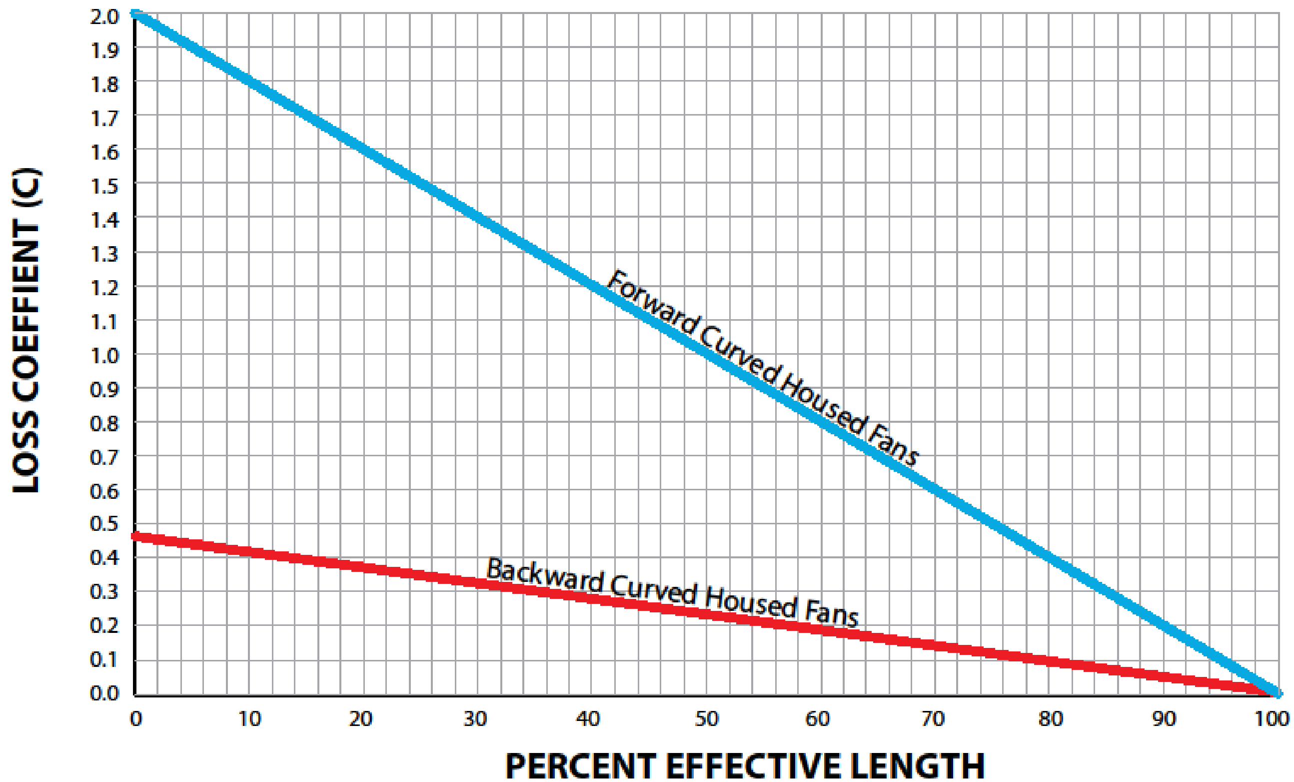
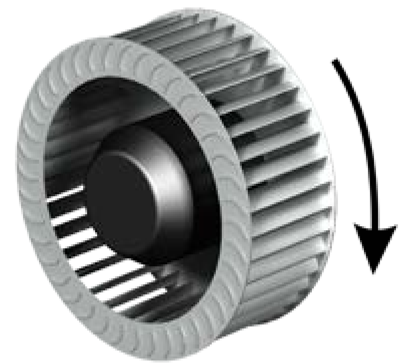
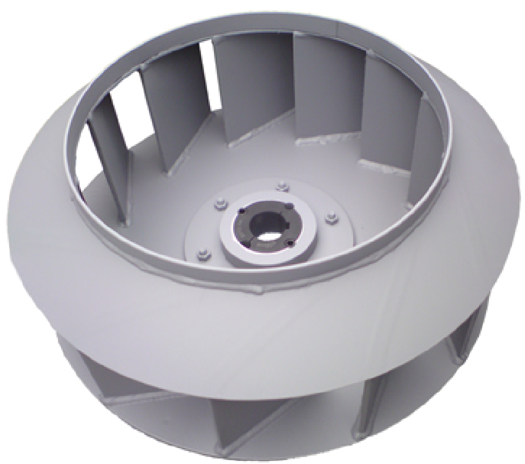


Fig. 32: House Fan Unducted Discharge Losses

Fans

- FC – Low Static, Lowest Efficiency
- BI – High Efficient, High Static
- AF – High Efficient, High Static
- Class I, II & III





Class I, II & III

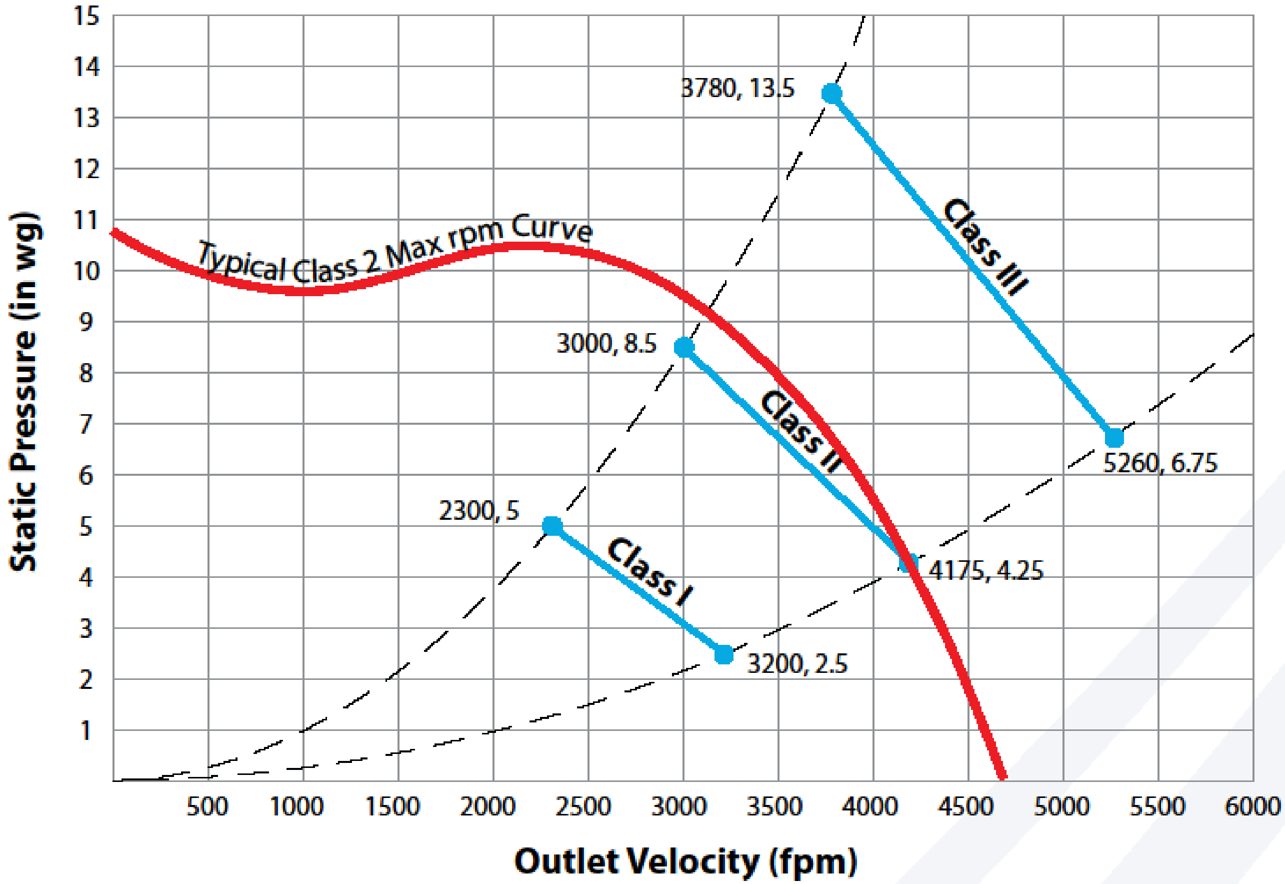


Fig. 17: AMCA Fan Class

SWSI Fans

- Exhaust Fans
- Regeneration Fans
- Previous System Effect Applies

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DWDI Fans

- Supply & Return Fans
- Previous System Effect Applies

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Vane Axial Fans

- High Static
- High Volume
- Very High Efficiency
- Supply & Return Fans
- Previous System Effect Applies

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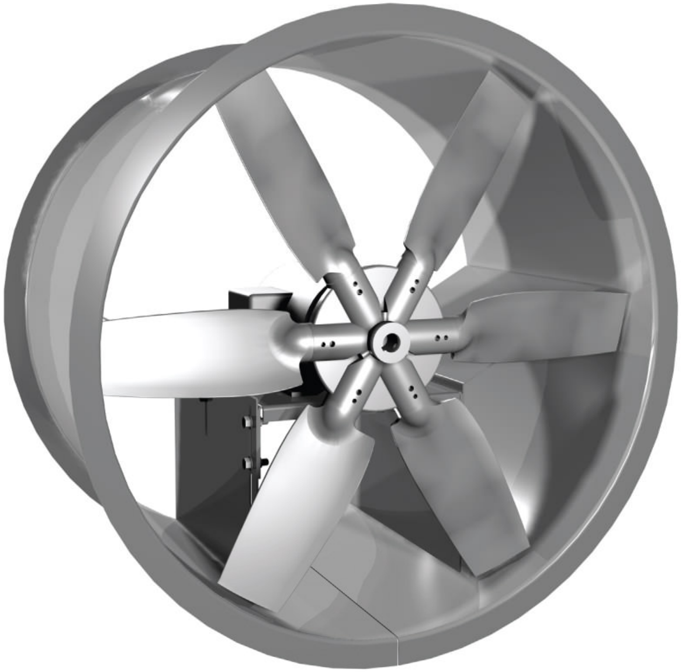
Propeller Fans

- Low Static
- High Volume
- Exhaust Fans
- Previous System Effect Does Not Apply
  - Velocity Pressure



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Plenum Fans

- Typically BI or AF
- Supply, Return & Exhaust Fans
- Previous System Effect Does Not Apply

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Plenum Fans

- Virtually no Velocity Pressure
  - Not Pushed Against the Housing Creating Unbalanced Outlet Velocity Profile
  - Motor is Out of the Way, No System Effect

Plenum Fan Inlet Conditions

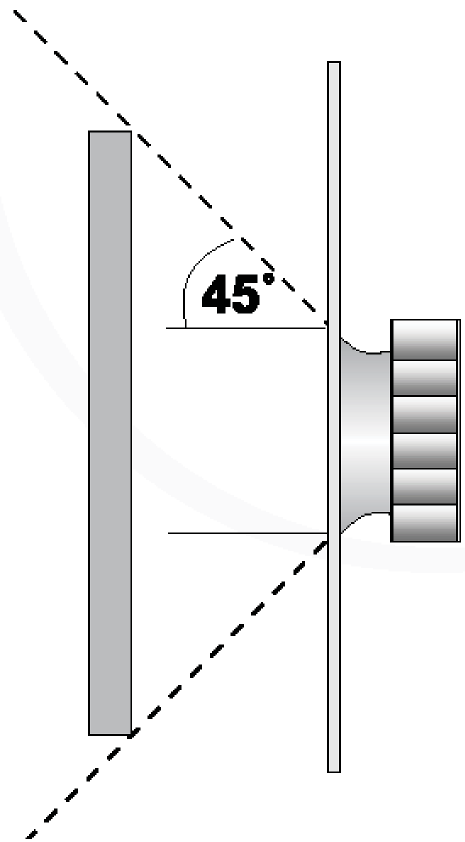

Fig. 28: Airstream Approach Angle

Plenum Fan Outlet Conditions

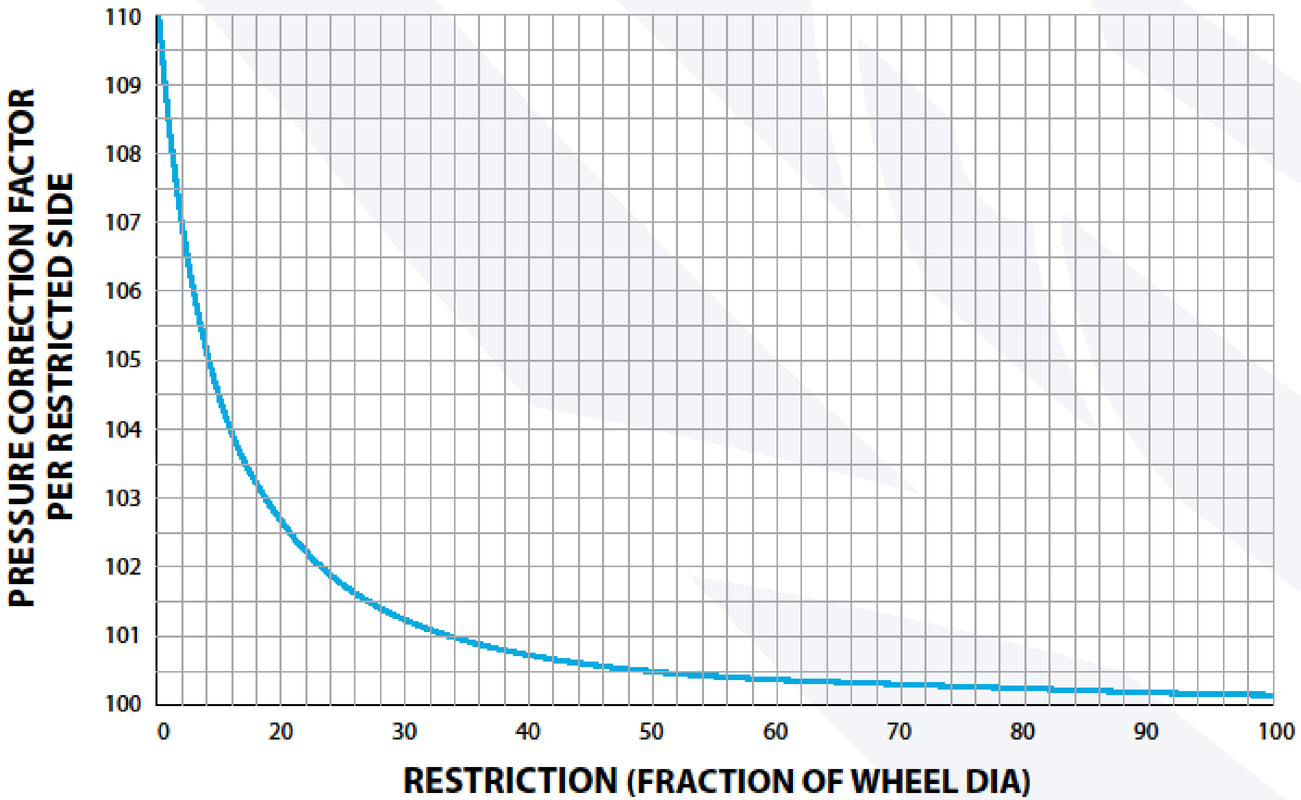

Fig. 35: Plenum Fan cfm Correction for Side Restriction

Belt Driven Fans

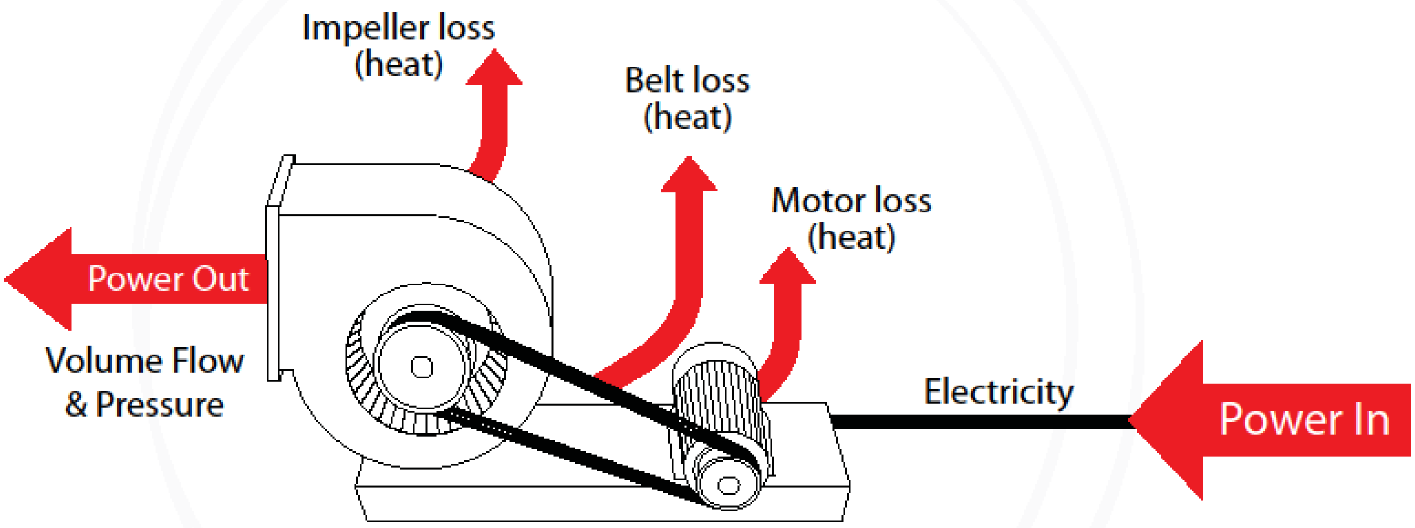

Fig. 36: Belt Driven Fan System

Direct vs. Belt Driven Fans

- New Belts
  - Peak Efficiency 90-95%
- Worn Belts
  - 85-90% Efficient

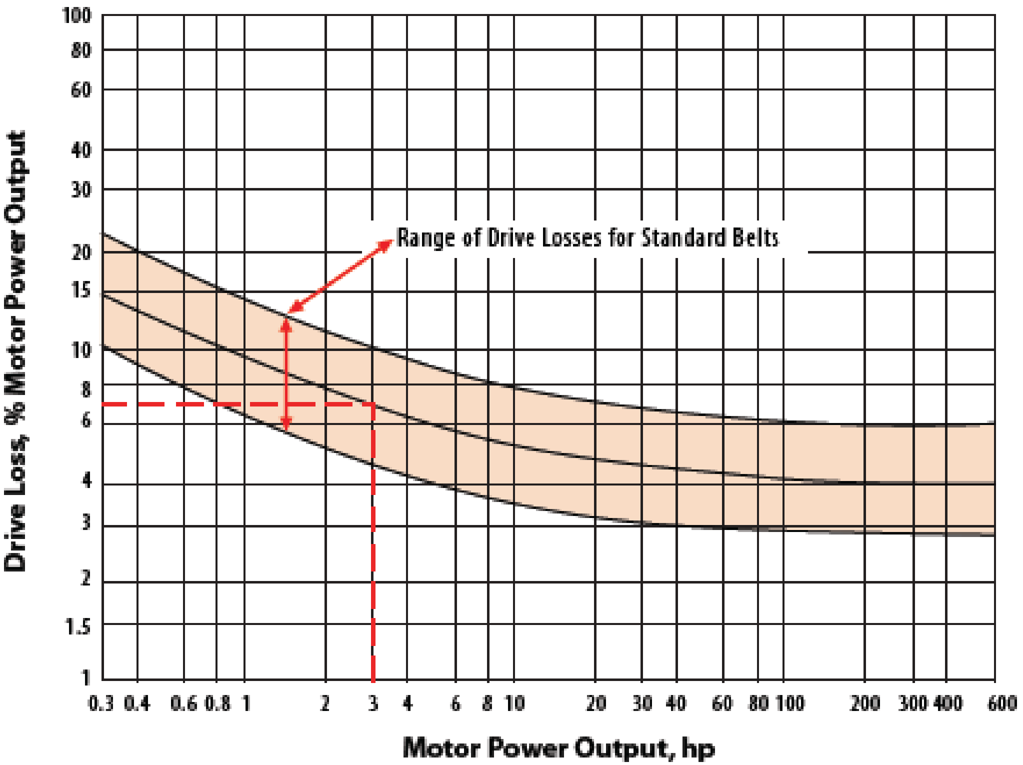

Figure 1: Average Belt Drive Losses vs. Motor Power Output
(AMCA publication 203-90)

Direct Driven Fans

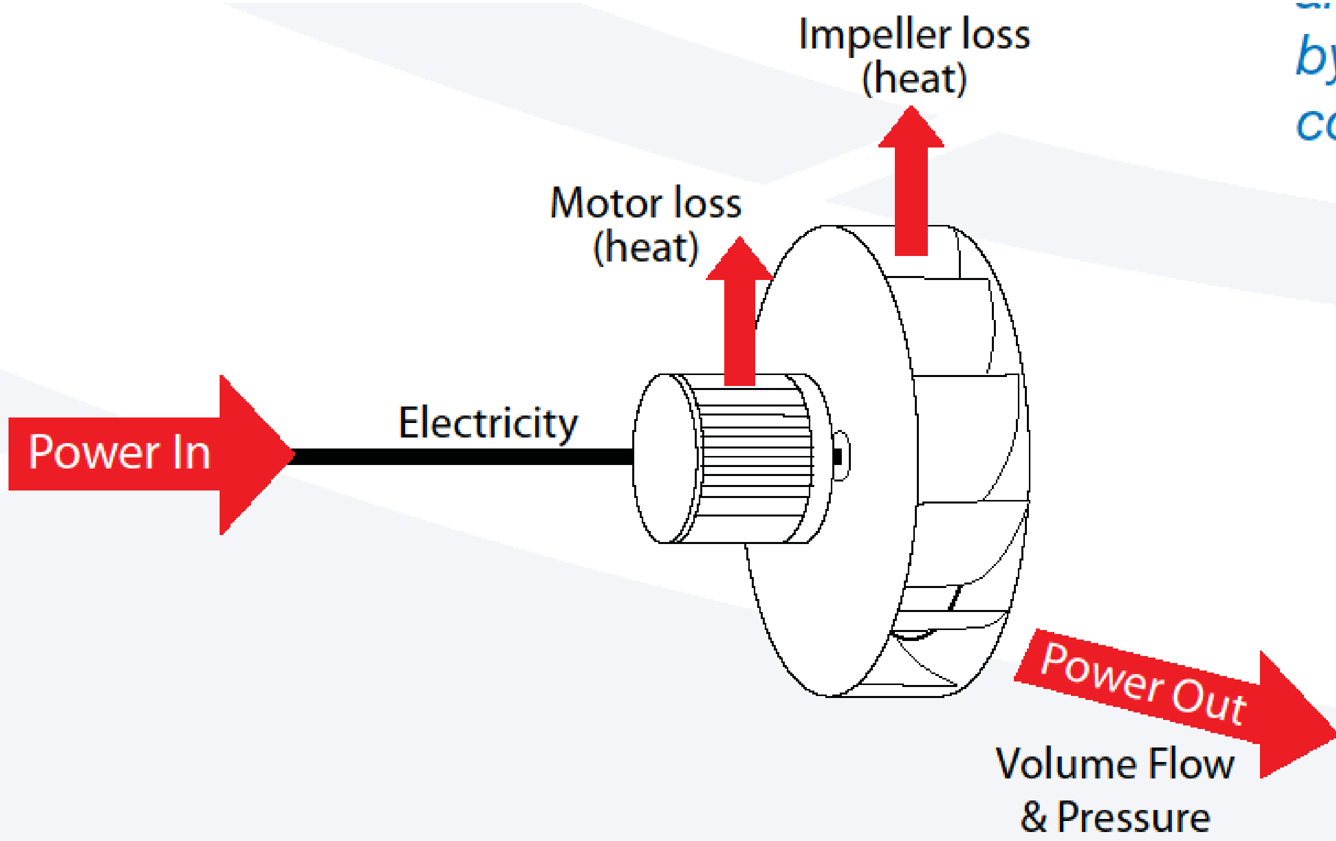

Fig. 37: Direct Drive Fan System

What Fan Would You Choose?

Calculated Application Efficiency

	Motor Efficiency		Belt Efficiency		Fan Efficiency		System Effects		Total System Efficiency
Belt-Driven, Housed, Forward Curved Total Efficiency =	(0.90)	•	(0.87)	•	(0.60)	•	(0.70)	=	**33%**
Belt-Driven, Housed, Backward Curved Total Efficiency =	(0.90)	•	(0.87)	•	(0.75)	•	(0.80)	=	**47%**
Direct Drive, Unhoused Backward Curved, Total Efficiency =	(0.90)	•	(1.00)	•	(0.70)	•	(1.00)	=	**63%**

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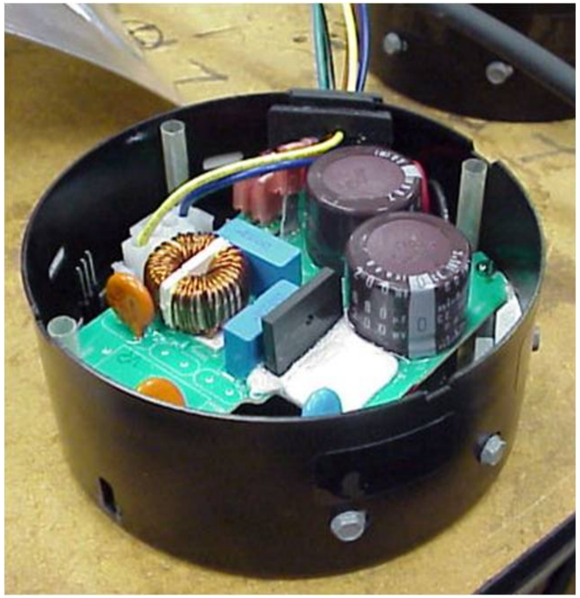
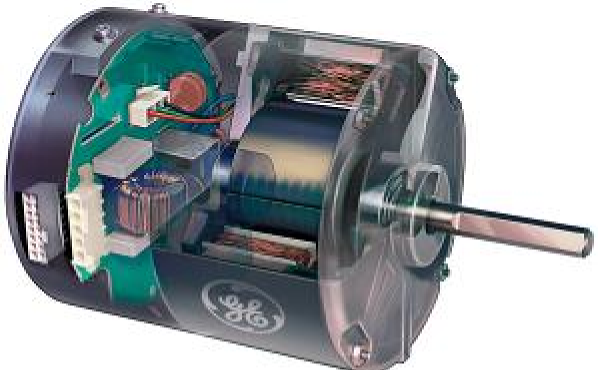
ECM Motors & VFD's





ECM Motors & VFD’s

- ECM = AC to DC Speed Control
- VFD = AC to DC to AC Speed Control
- Soft Start
- Balancing Tool

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Return Fans or Exhaust Fans

- Assume 6400 CFM, 15 Ton Unit
  - 1” Supply ESP & .50” Return ESP
- Exhaust Supply Fan Sized for 1.5” ESP
- Exhaust Fan Sized for .5” ESP
  - 7.5 HP SF & 3 HP EF
- Return Supply Fan Sized for 1” ESP
- Return Fan Sized for .5” ESP
  - 5 HP SF & 2 HP RF

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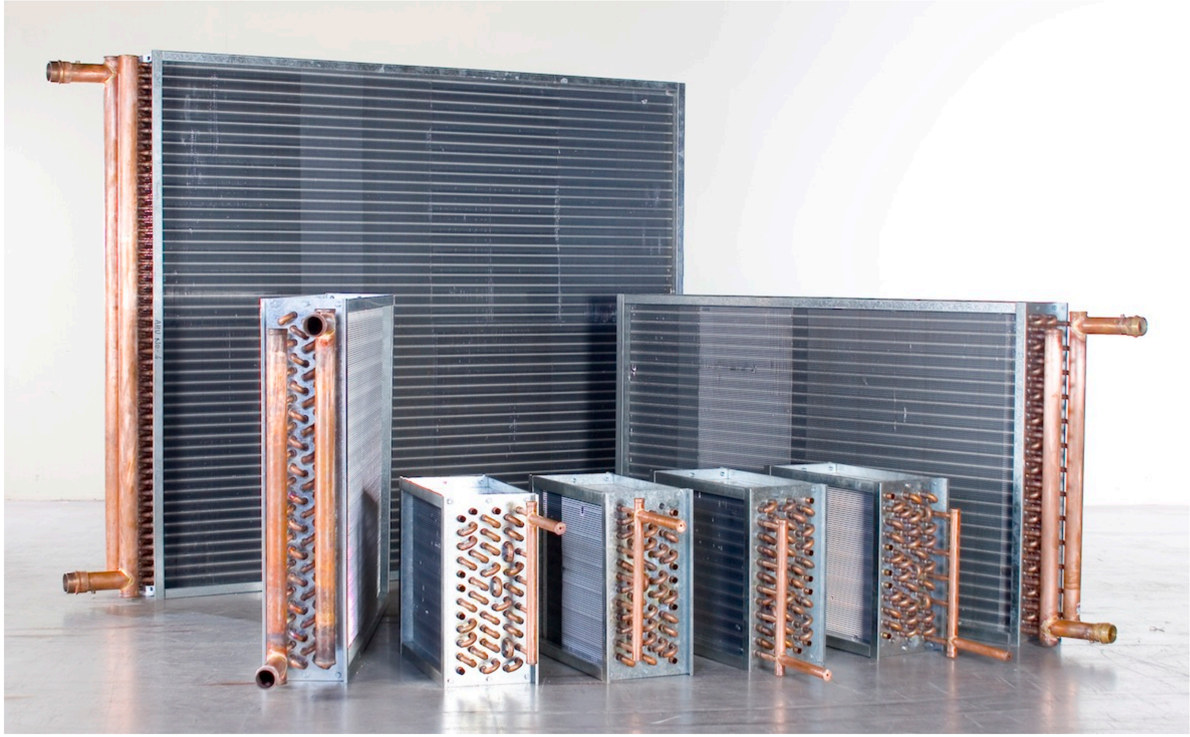
Cooling Options

- Chilled Water
- Direct Expansion

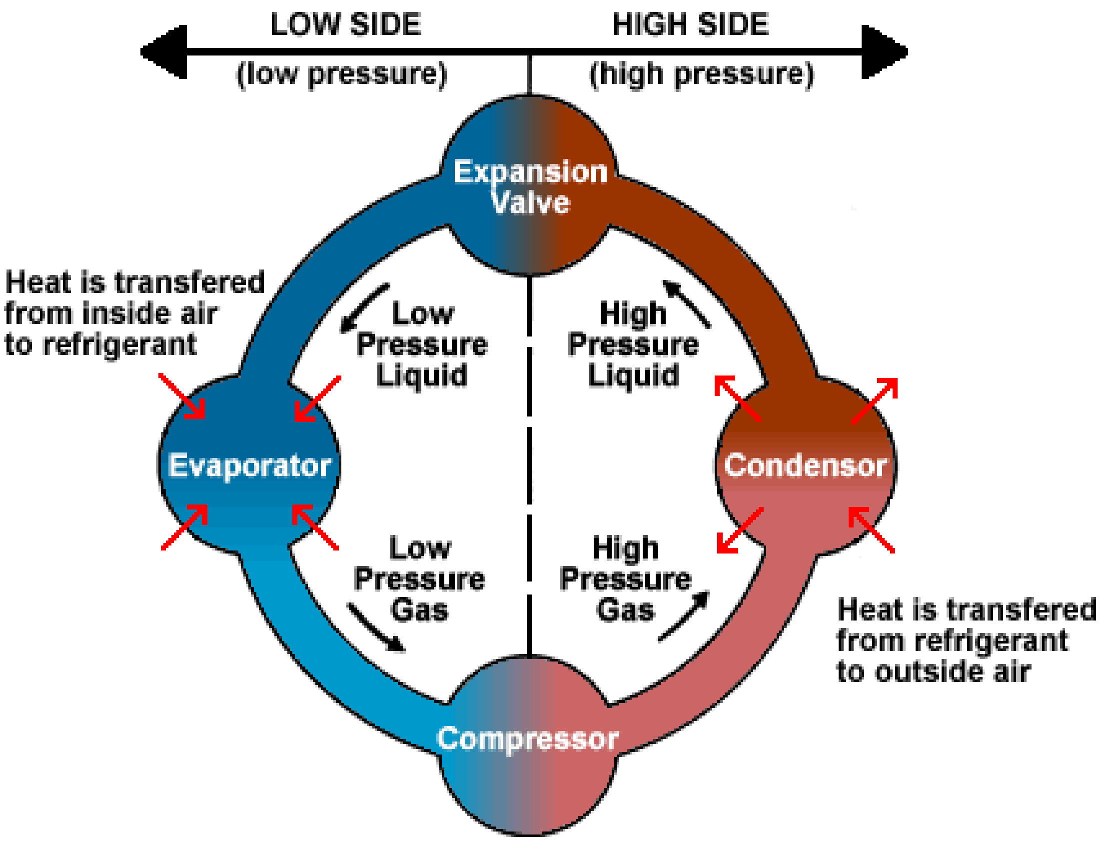
Aaon Evaporative Condensing Chiller

Chilled Water Coils

- Multiple Coil Options



Direct Expansion



Direct Expansion

- Air Source & Water Source
- Multiple Coil Options
- Expansion Devices
  - Thermal Expansion Valve
  - Electronic Expansion Valve

High Capacity DX Coils

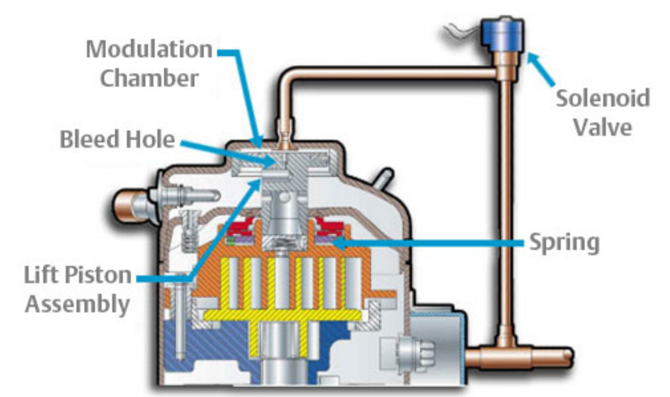
- Increased Efficiency
- Increased Dehumidification

Micro Channel Condenser Coils

- Reduced Refrigerant Amounts By 40%
- Think Car Radiator

Compressors

- Single Stage Compressors
- Multiple Staging with Multiple Compressors
- Modulating Compressors
  - – Scroll – Digital & VFD
  - – Screw - VFD
  - – Centrifugal - Magnetic
- Or Hot Gas Bypass



Digital Scroll Compressor

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Head Pressure Control

- Variable Speed Condenser Fan Provides Energy Savings
  - Variable Speed Compressors
  - Fluctuating Ambient Conditions
- Similar to Cooling Tower with VFD's

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Heating Options

- Steam Heat
- Hot Water Heat
- Electric Heat
- Gas Heat
- Heat Pump & Hybrid Heat

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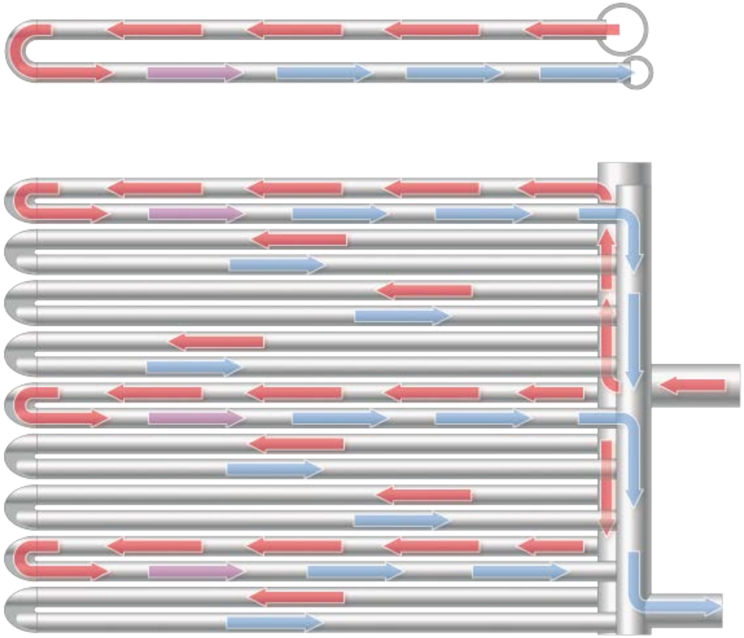
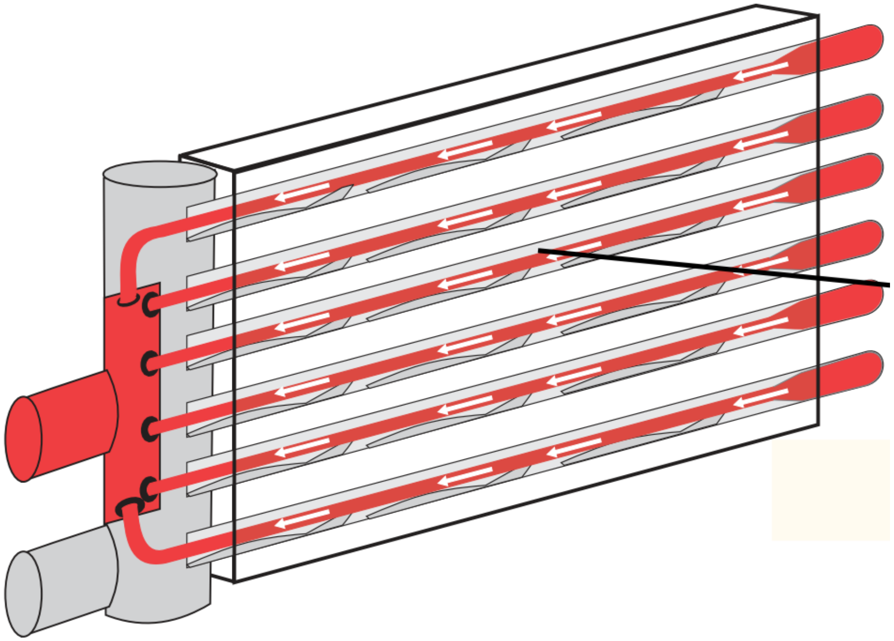
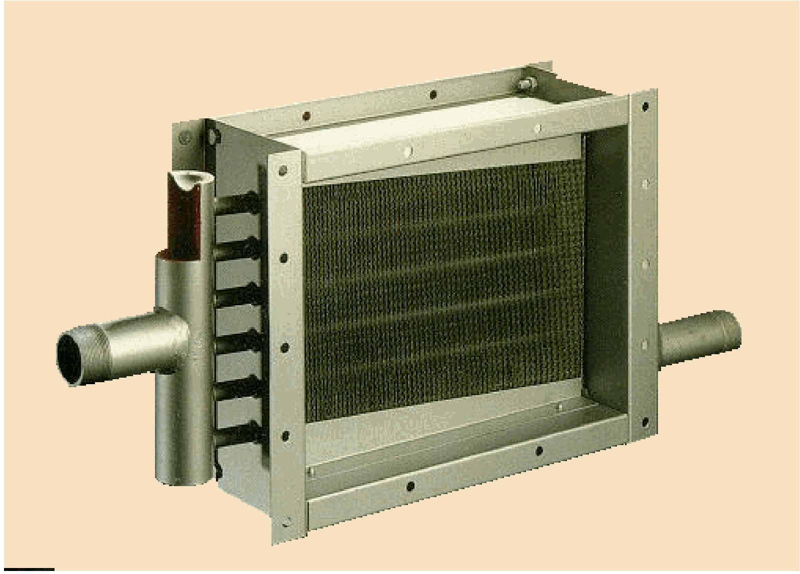
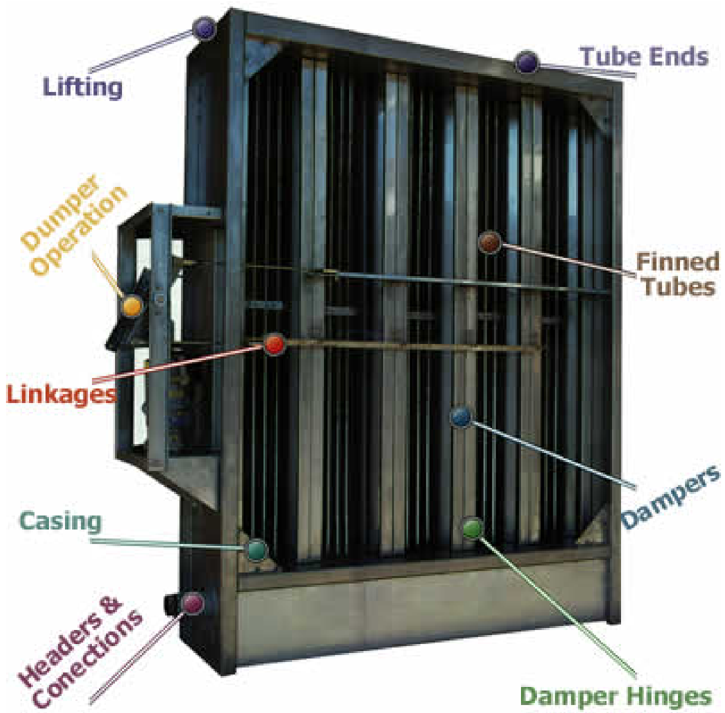
Steam Heat





Steam Heat



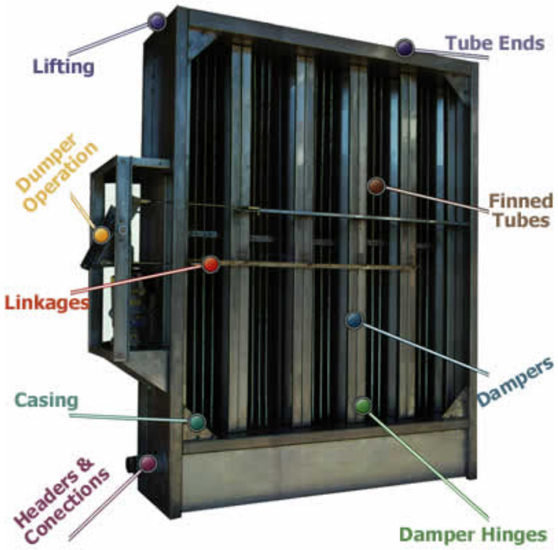


Lifting
Tube Ends
Dumper Operation
Finned Tubes
Linkages
Dampers
Casing
Headers & Conections
Damper Hinges

Steam Heat

- Single Coil for Normal Heating
- Single Coil for 100% OA
- Two Coils for 100% OA (PH & RH)
- F&BP for 100% OA
  - Internal, External & Integral F&BP

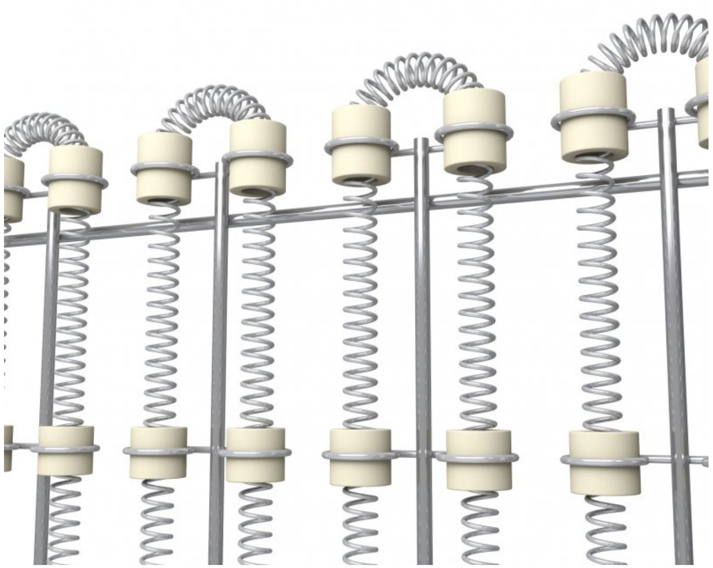
Hot Water Heat



Hot Water Heat

- Single Coil for Normal Heating
- Single Coil for 100% OA
- Two Coils for 100% OA
- F&BP for 100% OA
  - Internal, External & Integral F&BP

Electric Heat



Electric Heat

- Open Wire & Fin Tubular Element
- Multiple Stages with Contactors
- Modulating with SCR Controls

Gas Heat

- Staging
- Modulating

Gas Heat

- 1 Stage
  - 40 Degree TR = 40 Degree Minimum TR
- 2 Stage
  - 40 Degree TR = 20 Degree Minimum TR
- 4 Stage
  - 40 Degree TR = 10 Degree Minimum TR

Gas Heat

- 3:1 Modulation
  - 90 Degree TR = 30 Degree Minimum TR
- 5:1 Modulation
  - 90 Degree TR = 18 Degree Minimum TR
- 10:1 Modulation
  - 90 Degree TR = 9 Degree Minimum TR
- 20:1 Modulation
  - 90 Degree TR = 4.5 Degree Minimum TR

Hybrid Heat

- Primary Air Source Heat Pump
- Primary Water Source Heat Pump
- Secondary Gas, HW, Steam or Electric

Hybrid Heat for 100% OA

- Infinite TR
- 3:1 Modulation
  - 90 Degree TR = 1 Degree Minimum TR
- 5:1 Modulation
  - 90 Degree TR = 1 Degree Minimum TR
- 10:1 Modulation
  - 90 Degree TR = 1 Degree Minimum TR

Temperature Controls

- Factory Analog
- Factory Digital
  - BACnet or LON Compatibility
- Factory Mounted DDC by Others
- Field Mounted DDC by Others
  - Isolation Relays

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Filtration Options

- MERV 7 or 8
- MERV 13, 14 or 15
  - 4” or 12”
- Clogged Filter Switch
- Magnehelic Gauge

Filter Loading

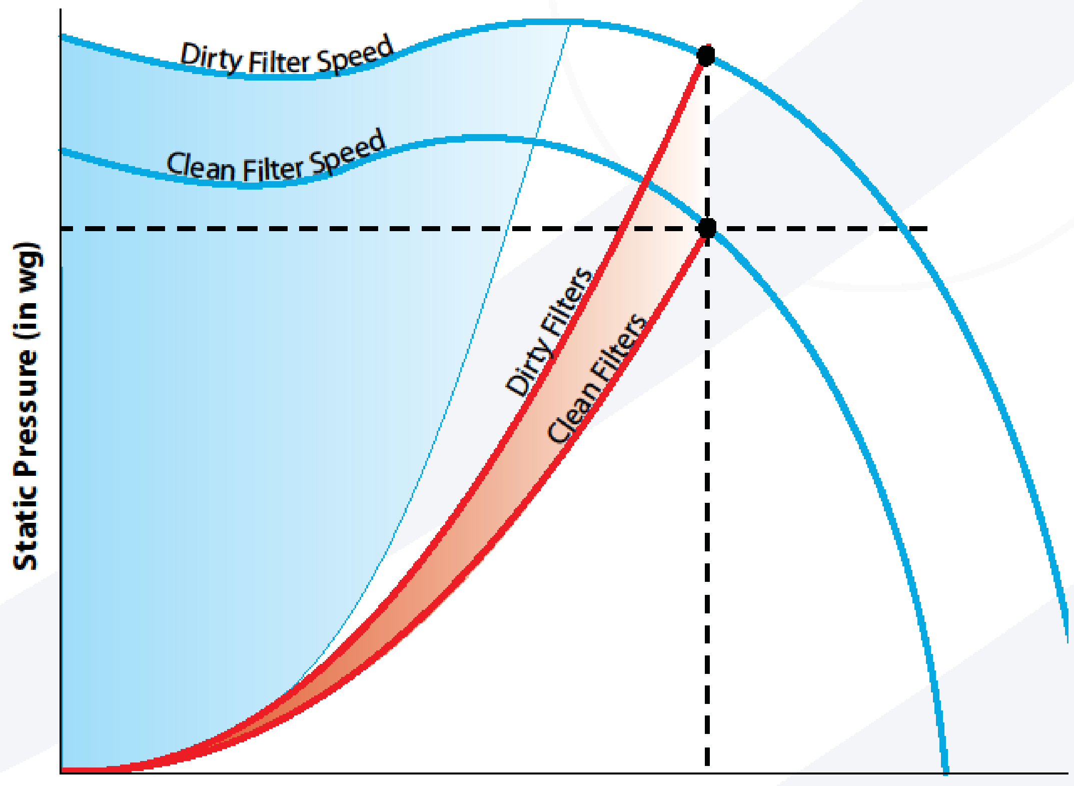

Fig. 23: VAV System with Filter Loading

Cabinet Construction

- 2500 Hour Salt Spray Testing per ASTM B 117-95
- 2” Double Wall Panels
- R-13 Foam Insulation
- Full Thermal Break
- Galvanized or Stainless Steel Construction
- Stainless Steel Piano Hinges and Corrosion Resistant Lockable Handles
- Sloped Stainless Steel Drain Pan

2,500 Hour Salt Spray Test

- ASTM B 117-95 Testing Procedure
- 5% Salt Spray & Fog Atmosphere
- Stopped At First Visible Sign Of Deterioration
- Can Be Custom Color

Aaon

8 year old custom unit with 1,000 Hr. Salt Spray Test

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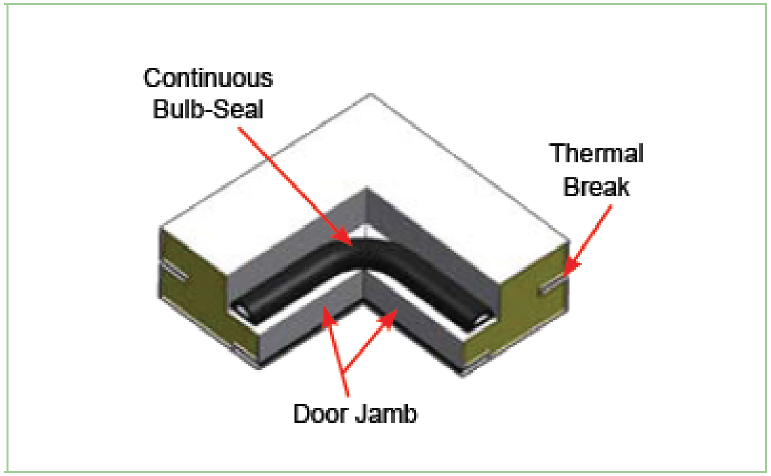
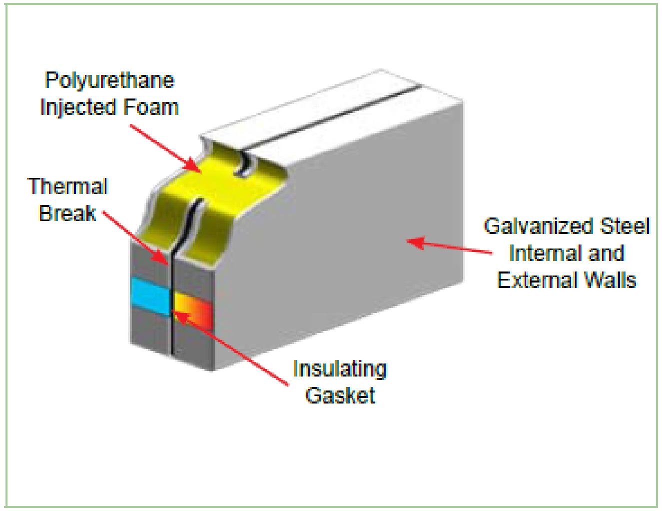
Double Wall Foam Panel Construction

- Thermal Resistance & Break
- Air Seals
- Rigidity
- Maintainability
- Indoor Air Quality
- Equipment Life
- Energy Savings



AAON Rigid Polyurethane Foam Panels



AAON Rigid Polyurethane Foam Panels

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Double Wall Foam Panel Energy Savings

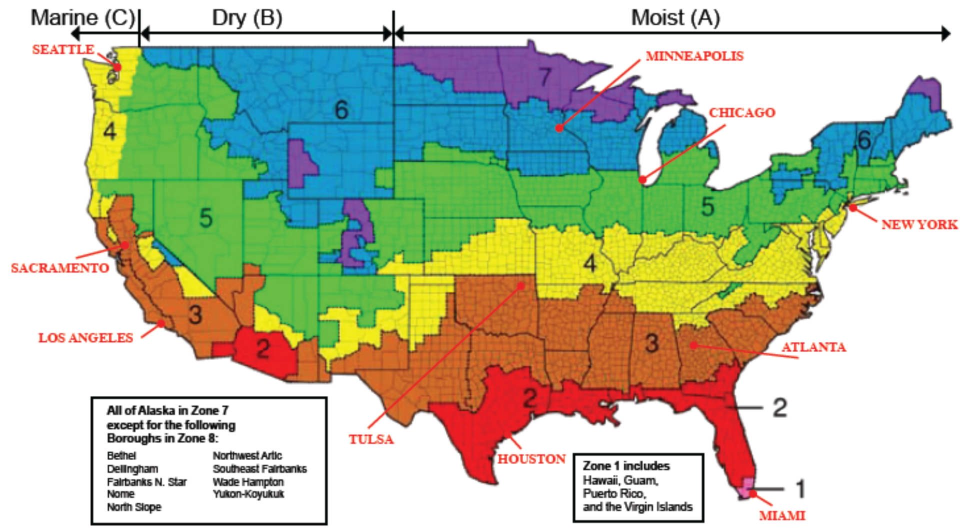

Figure 8: ASHRAE Climate Zones

Heating Savings

	Nominal Tons							
	5	10	20	35	75	125	175	210
Atlanta	$91	$170	$310	$553	$1,142	$1,722	$2,353	$2,794
Chicago	$154	$287	$522	$931	$1,924	$2,985	$4,078	$4,843
Houston	–	–	–	–	–	–	–	–
Los Angeles	–	–	–	–	–	–	–	–
Miami	–	–	–	–	–	–	–	–
Minneapolis	$177	$331	$603	$1,074	$2,221	$3,446	$4,707	$5,590
New York	$130	$242	$440	$784	$1,622	$2,516	$3,437	$4,081
Sacramento	$107	$200	$364	$649	$1,342	$2,084	$2,846	$3,380
Seattle	$146	$273	$497	$886	$1,833	$2,844	$3,885	$4,613
Tulsa	$105	$196	$356	$635	$1,313	$2,037	$2,783	$3,305

Table 13: Estimated Heating Savings from AAON Rigid Polyurethane Foam Cabinet
($0.12/kWh and $1.20/therm)

Cooling Savings

	Nominal Tons							
	5	10	20	35	75	125	175	210
Atlanta	$151	$295	$574	$1,009	$2,139	$3,496	$4,861	$5,818
Chicago	$74	$144	$279	$491	$1,040	$1,693	$2,351	$2,812
Houston	$278	$544	$1,058	$1,861	$3,946	$6,442	$8,958	$10,719
Los Angeles	$46	$91	$177	$311	$662	$1,088	$1,516	$1,816
Miami	$394	$769	$1,493	$2,628	$5,569	$9,089	$12,635	$15,117
Minneapolis	$67	$130	$253	$444	$941	$1,534	$2,133	$2,552
New York	$82	$159	$308	$542	$1,147	$1,867	$2,593	$3,101
Sacramento	$56	$106	$198	$350	$731	$1,158	$1,158	$1,898
Seattle	$14	$27	$51	$89	$187	$302	$418	$500
Tulsa	$166	$324	$625	$1,100	$2,327	$3,781	$5,249	$6,277

Table 12: Estimated Cooling Savings from AAON Rigid Polyurethane Foam Cabinet
($0.12/kWh and $1.20/therm)

Psychrometrics of Motor Heat

- Draw Through
- Blow Through

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Applications – Blow Through

- Large VAV systems
- High sensible loads
- Higher efficiency requirements
- Sound sensitive applications

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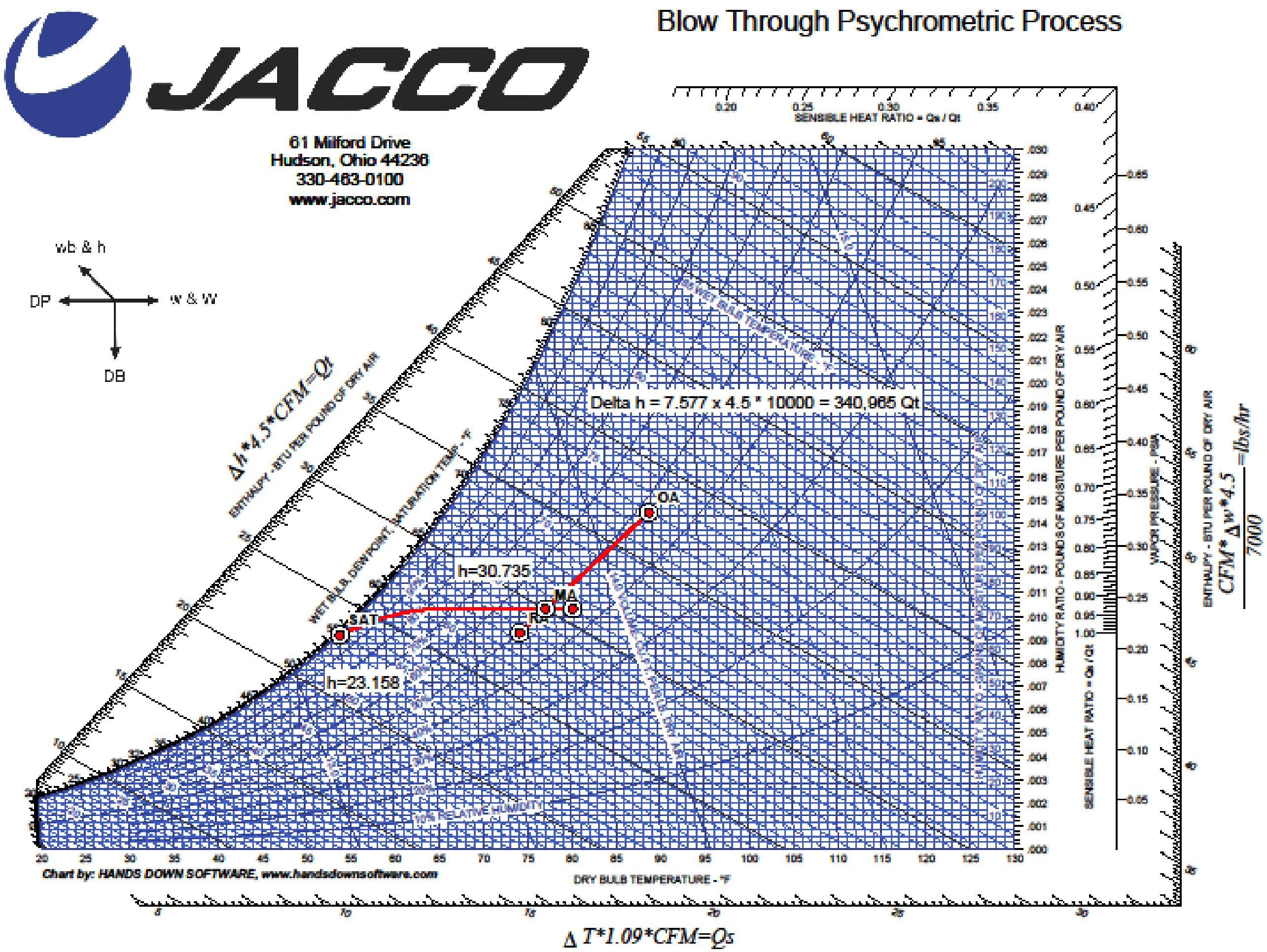
Blow Through

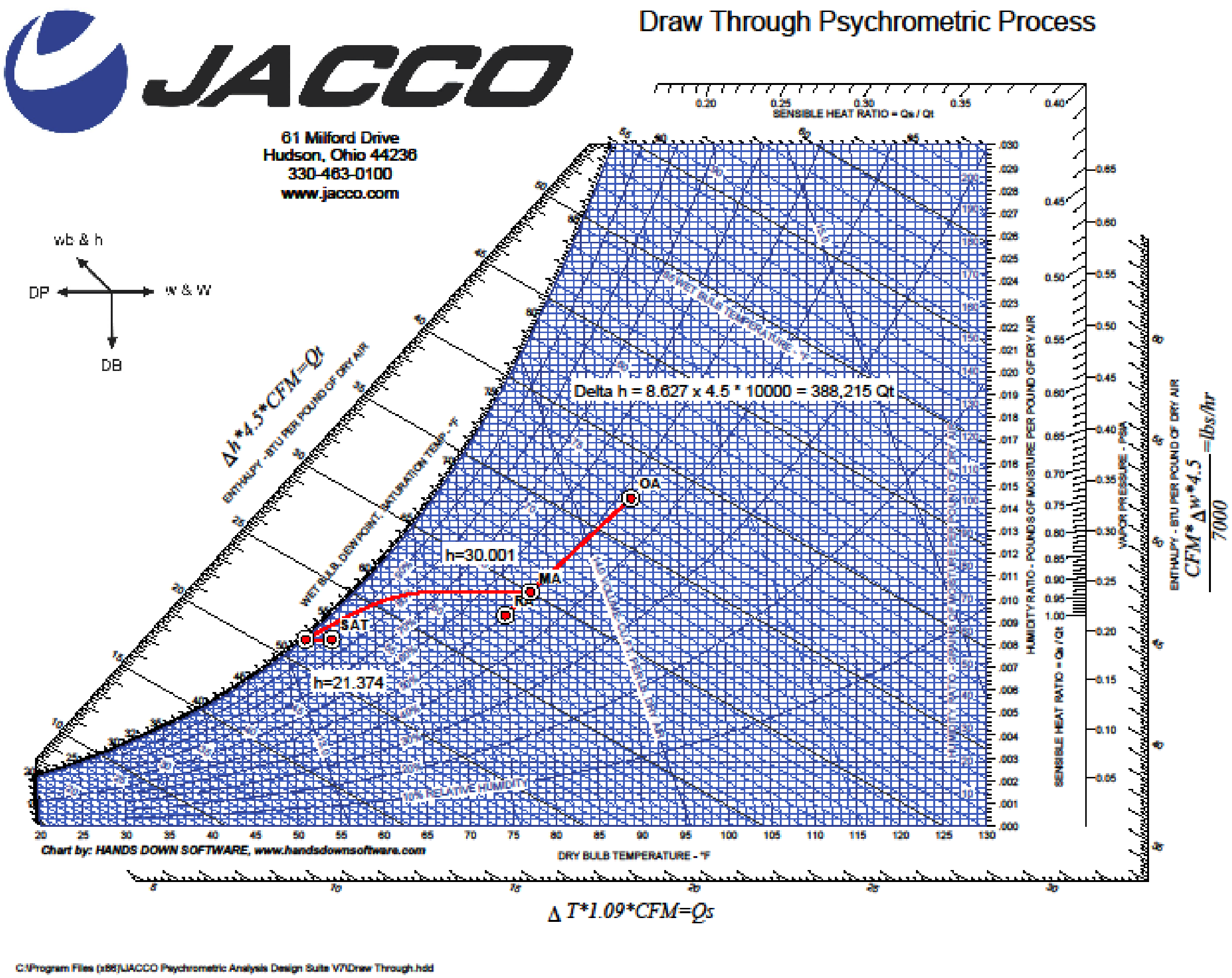


C:\Program Files (x86)\JACCO Psychrometric Analysis Design Suite V7\Draw Through.hdd

Applications – Draw Through

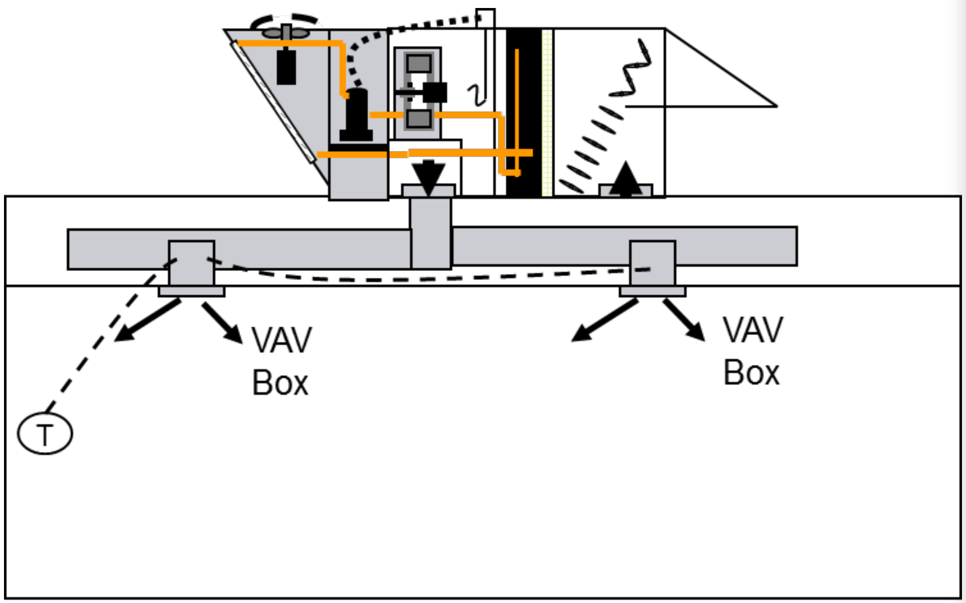
- Compact space requirements
- High latent loads
  - Pools
  - Underfloor or Displacement
- Initial cost constraints

Draw Through



Traditional VAV Systems

- Traditional VAV systems feed multiple zones from one unit
- Supply airflow changes to maintain supply duct pressure
- Unit capacity changes to maintain supply air temperature



Minimum VAV Flow

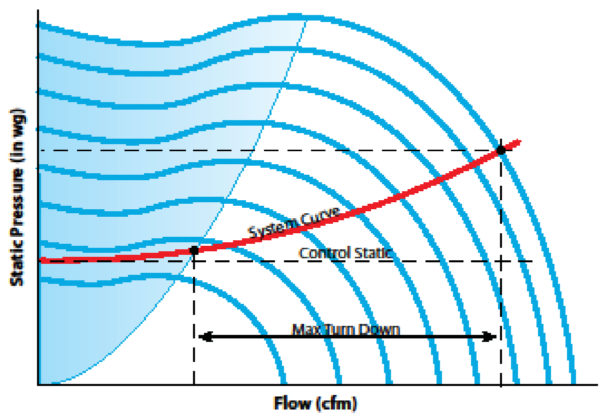


Fig. 25: VAV System Selected Further Right of Peak Static

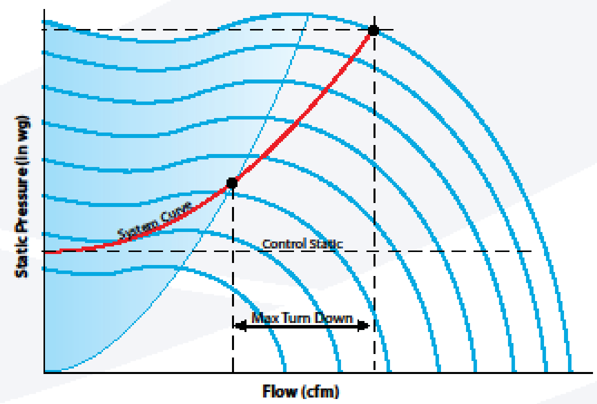
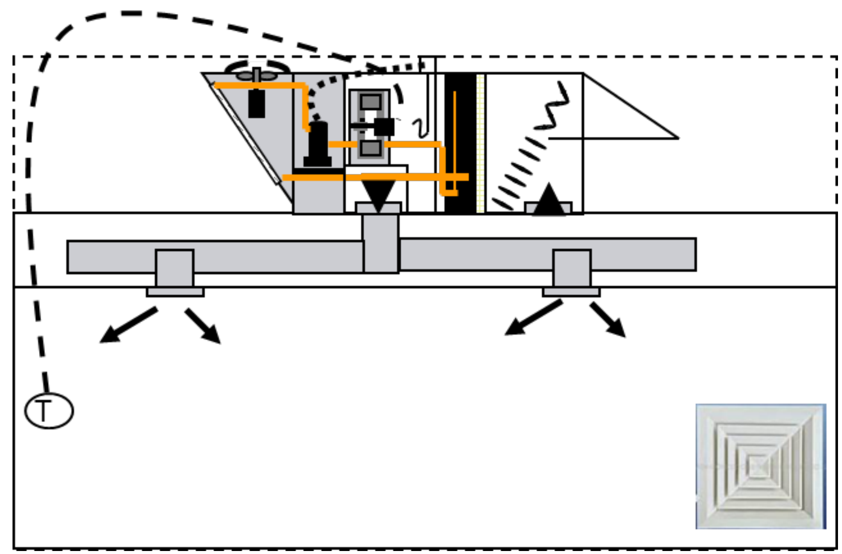


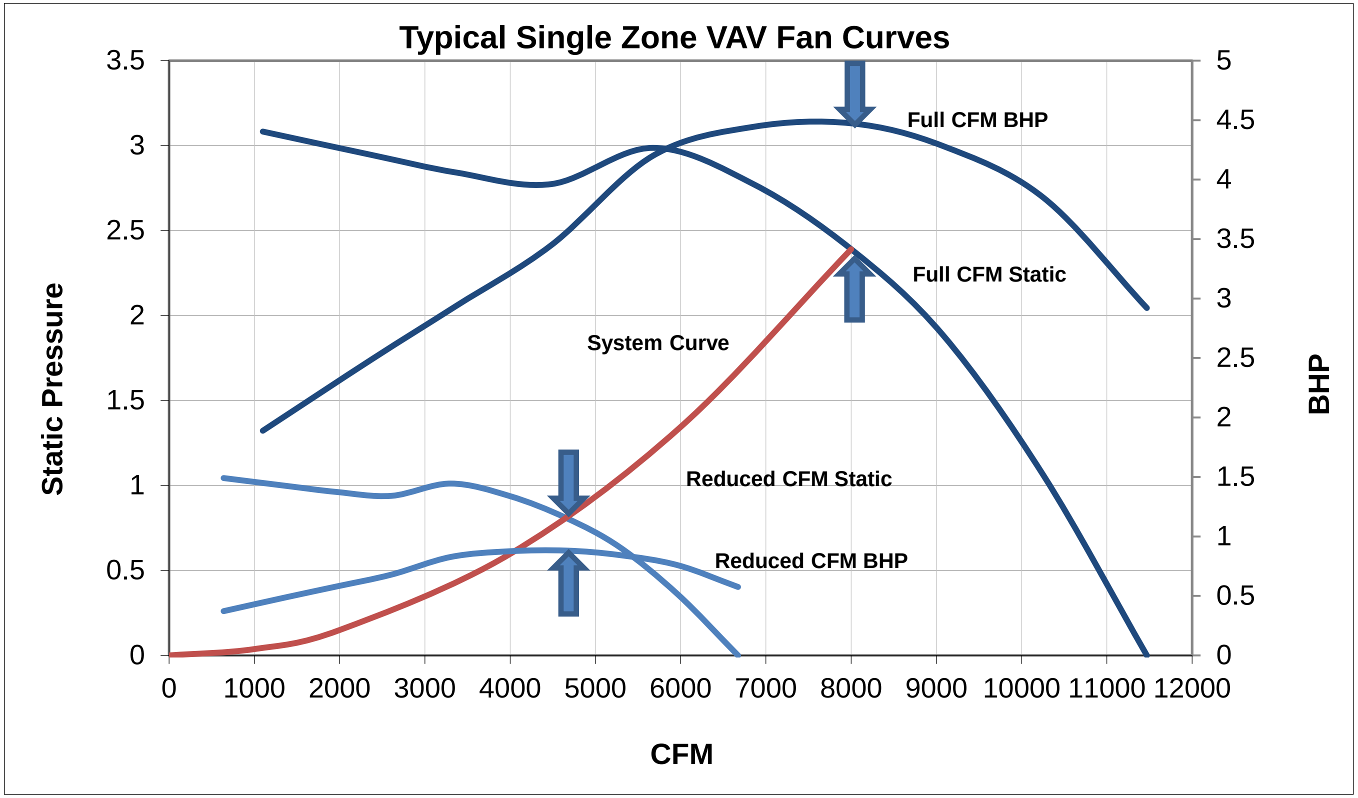
Fig. 24: VAV System Selected Close to Peak Static & Efficency

Single Zone VAV Systems

- Single Zone VAV systems serve one zone.
- Airflow changes based on space load
- Unit capacity changes to maintain supply air temperature
- SAT set point can be reset to maintain humidity control (if reheat available)
- VAV boxes not required

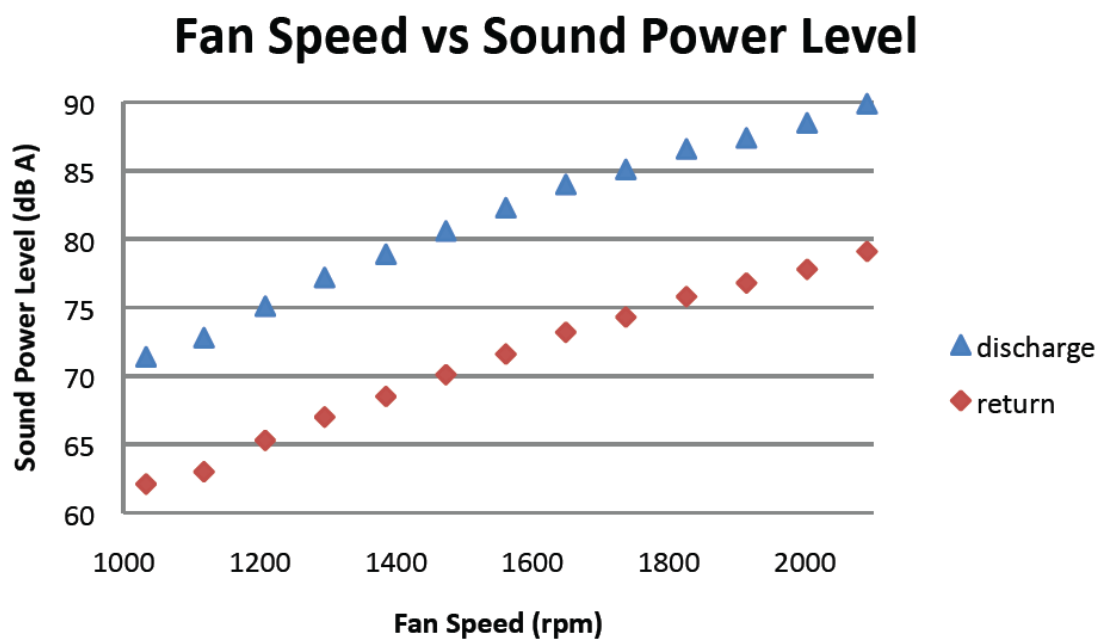


SZVAV Fan Energy Savings

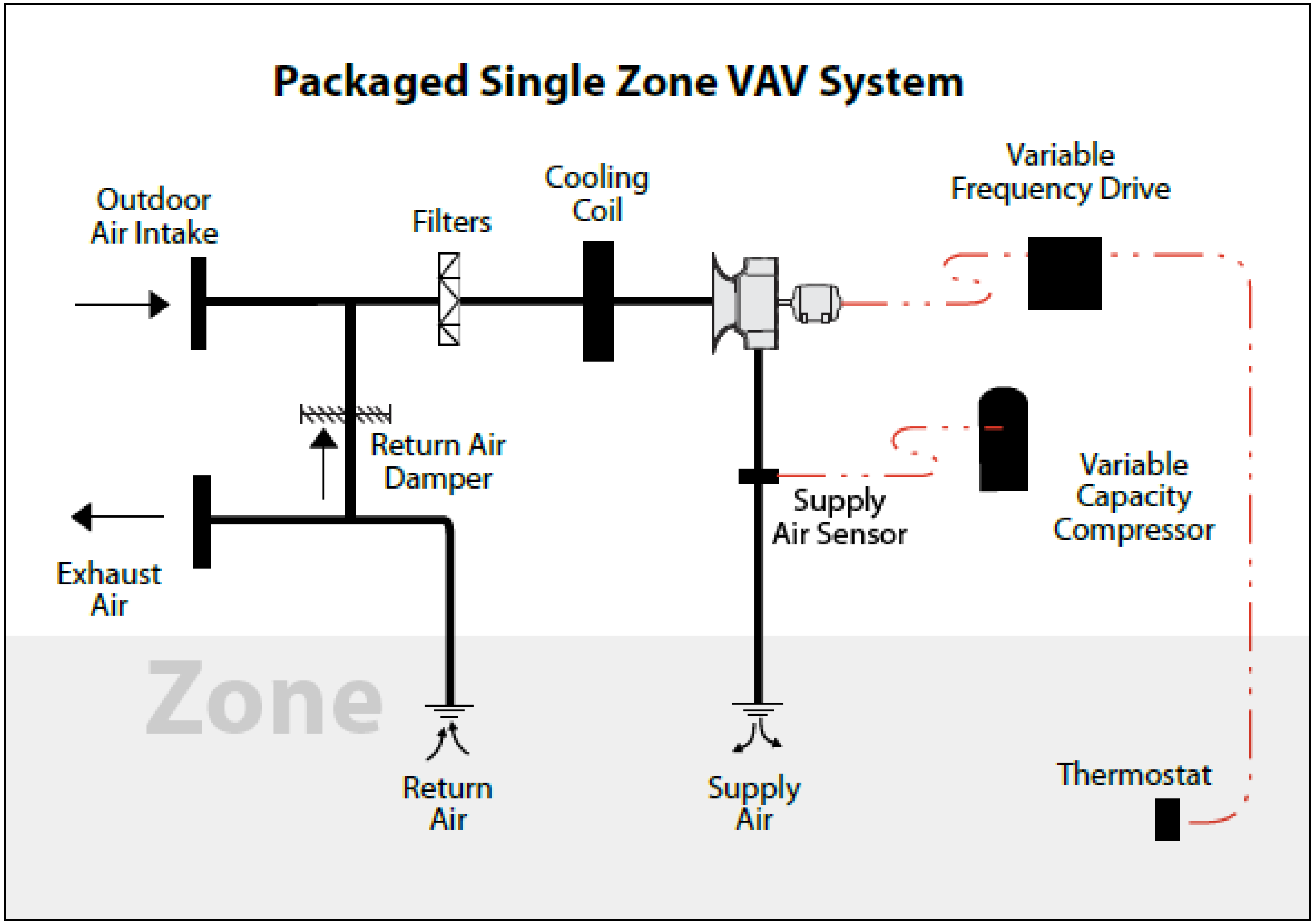


SZVAV Sound Benefit

Another benefit to airflow reduction is the reduction in fan noise due to change in speed



Single Zone VAV Controls



Best Installation Practices

- Location
- Clearance
- Sound
- Isolation
  - Spring or Rubber in Shear (RIS)

Acoustical Considerations

Proper unit placement is critical to reducing transmitted sound levels from the unit to the building. Do not locate units directly above areas such as: **offices, conference rooms, executive office areas, and classrooms**. Instead, ideal locations to consider are: **corridors, utility rooms, toilets, or other areas** where higher sound levels directly below the units are acceptable.

1. Never cantilever the compressor side of the unit. A structural cross member or full perimeter roof curb, supported by roof structural members, must support this side of the unit.
2. Locate the unit's center of gravity close to or over column or main support beam.
3. If the roof structure is very light, replace roof joists by a structural shape in the critical areas described above.
4. If several units are to be placed on one span, stagger them to reduce deflection over that span.
5. Use the quietest fans available!!!!

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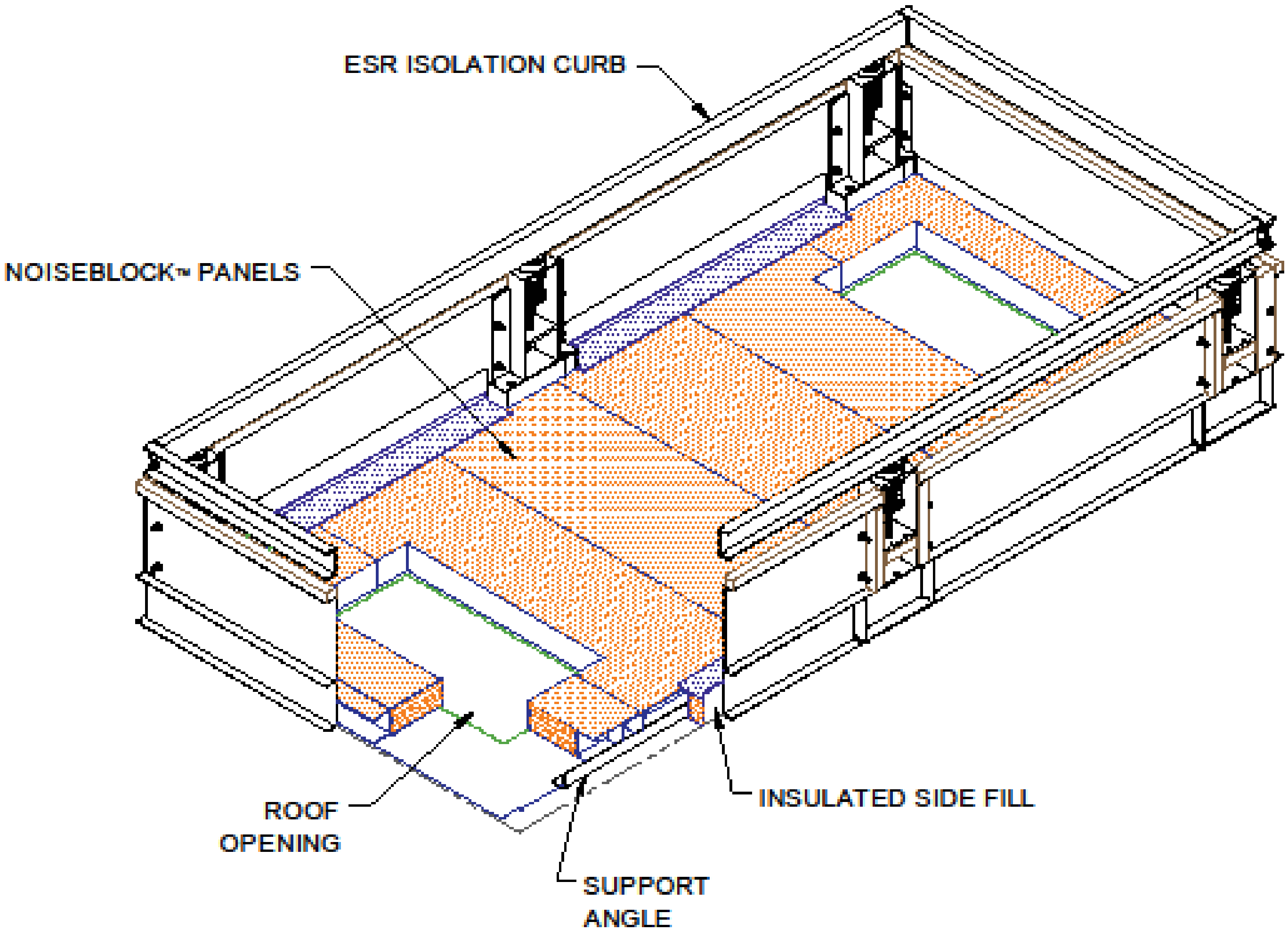
Acoustical Curbs



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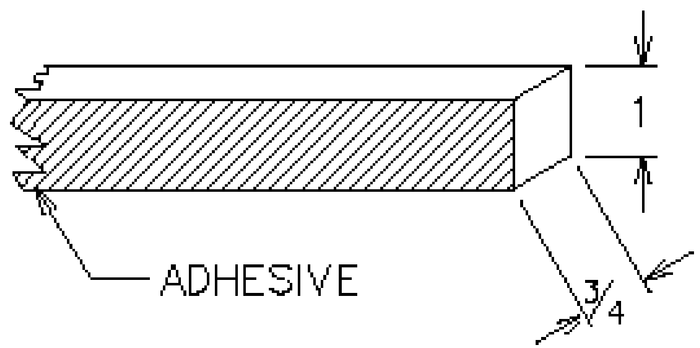
Poor Man Acoustical Curb

SPECIFICATIONS:

NOMINAL SIZE: 1" W X ¾" H

MATERIAL: NEOPRENE TYPE SCE 42 (CLOSED CELL)

TREATMENT: PRESSURE SENSITIVE ADHESIVE ON 1" SIDE



Clearance Requirements

Follow the recommended unit clearances to assure adequate serviceability, maximum capacity, and peak operating efficiency.

1. Do the clearances available allow for major service work, such as changing compressors or coils?
2. Do the clearances available allow for proper outside air intake, exhaust air removal, and condenser airflow?
3. If screening around the unit is used, is there a possibility of air recirculation from the exhaust to the outside air intake or from condenser exhaust to condenser intake.

When two or more units are placed side by side, increase the distance between the units to twice the recommended single unit clearance. Stagger the units for these two reasons:

1. To reduce span deflection if more than one unit is placed on a single span.
2. To assure proper exhaust air diffusion before contact with the adjacent unit's outside air intake.

Duct Design

A well-designed duct system is essential to meet the rated capacities of the unit .

1. Satisfactory air distribution throughout the system requires an unrestricted and uniform airflow from the unit discharge duct.

2. When job conditions dictate installation of elbows near the unit outlet, using turning vanes may reduce capacity loss and static pressure loss.

3. Plenum return duct design should incorporate multiple turns before return air openings.

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AAON Rooftops

- Rooftop Package Units
  - 2 to 300 tons in 7 cabinet sizes
    - Air Cooled Condensing
    - Water Cooled Condensing
    - Evaporative Condensing
    - Geothermal
- Rooftop air handling units – 800 to 70,000 cfm

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The RQ Series Rooftop

- 2 through 6 Tons
- A Different Choice

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The RN Series Rooftop

- 6 through 140 Tons
- Large Capacity, Small Footprint, Lightweight

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The RL Series Rooftops

- 40 through 300 Tons

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RL Series
Evaporative Condensing

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AAON Rooftop Heat Pumps

- Air Source to 40 tons
- Water Source/ Geothermal to 140 tons +++
- 100% Outside Air Units
- Supplemental Heat (Electric, Gas, Hot Water, Steam)
  - Auxiliary Heating
  - Emergency Heating

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Some AAON Rooftop Applications

- Pool Units
- Make Up Air Units
- Tight Humidity Control Units
- Tight Temperature Control Units
- Heat Recovery Units
  - Coming Soon Plate Heat Exchangers
- Coming Soon Horizontal Duct Connections

Seismic Certification Compliance

- RQ & RN Rooftop Units (2-30 tons)
- IBC-2000
- IBC-2003
- IBC-2006
- IBC-2009
- IBC-2012

Tulsa, OK Facility

- 1.3 M sq. ft. on 54 acres
- 1,160 employees
- Rooftop package units
- Rooftop air handlers
- Chillers/Boiler/Pumping packages
- Large condensing units
- Large air handlers

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Longview, TX Facility

- AAON Coil Products, Inc. was founded in 1991 with the acquisition of Coils Plus, Inc. of Longview, Texas.
- 251,000 sq. ft.
- 25 acres
- 394 employees
- Coils, condensing units, air handlers, residential products

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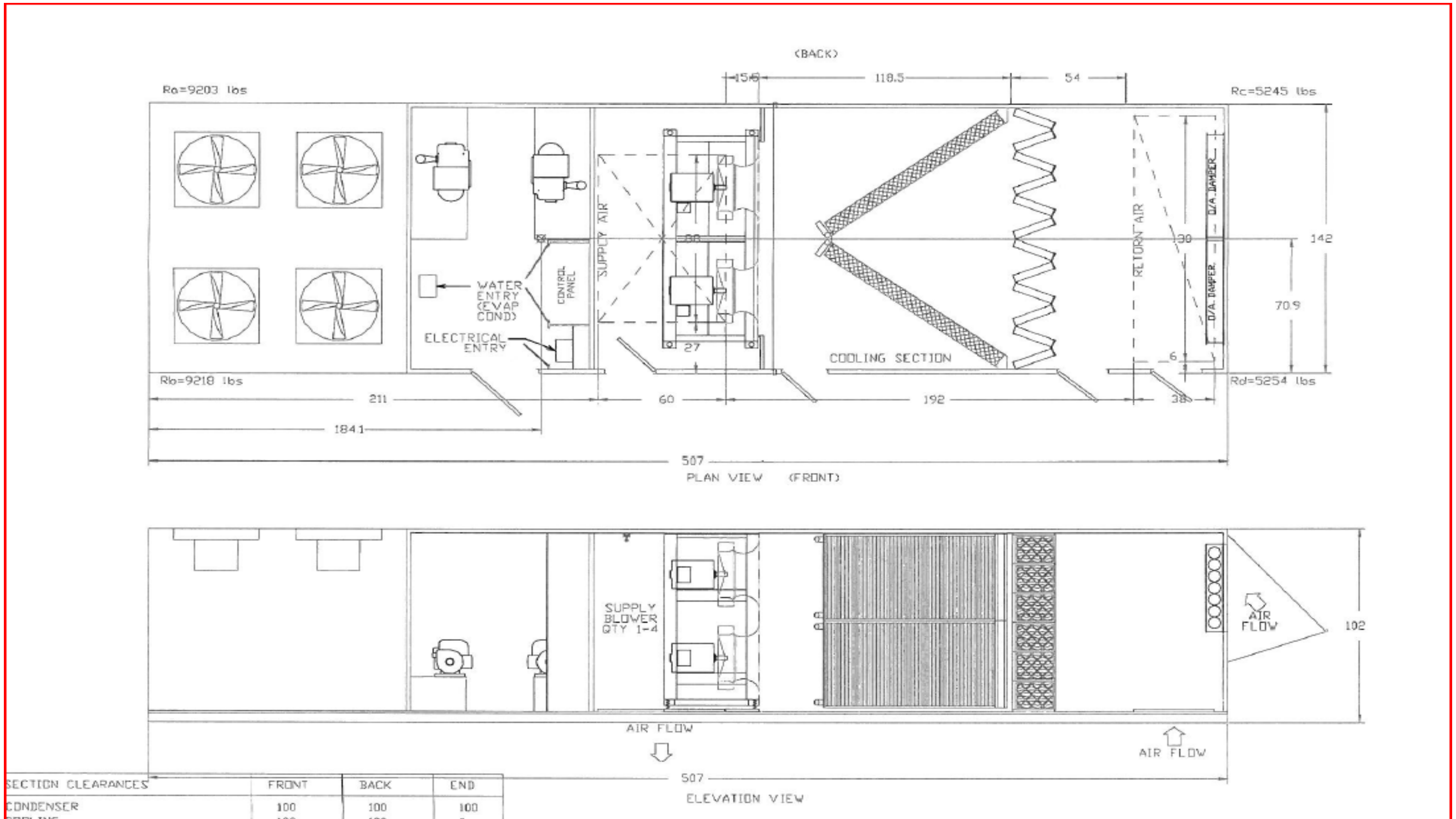
Aaon Rooftops w/ Oil Free Magnetic Bearing Centrifugal Compressors - from 90-300 Tons



Question for You

- Is a DX Rooftop as Efficient as a Chilled Water System?

Thank You