

AIRFLOW IN A SYSTEM

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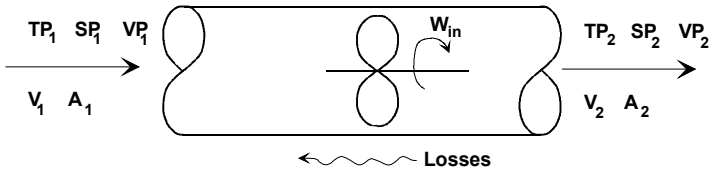
Learning Objectives

- Components of Air and Power Performance
- Fan and System Curves
- Surge and Stall
- Fan and System Relationships
- Fan Ratings
- Fan Laws
- Air Density
- Fans in Series and Parallel
- Sample Selections
- Questions

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Components of Airflow

The purpose of a fan is to add energy to the air. To move air from one point to another.



- TP = Total Pressure
- SP = Static Pressure
- VP = Velocity Pressure
- V = Velocity
- A = Area

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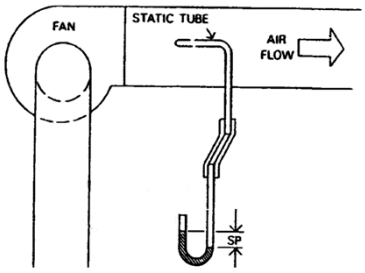
Fan Air Volume Flow

- The volume of air moved by the fan is usually expressed in cubic feet per minute (CFM) or cubic meter per second (CMS)
  - ACFM = volume of air moved by the fan at any air density
  - SCFM = volume of air moved by moved by the fan at standard air density (0.075 lbm./ft.^{3} or 1.2 kg/m^3)
- Measured using pitot tube (VP), calibrated nozzle or venturi
- Air Volume = Velocity (V) * Area (A)

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Static Pressure

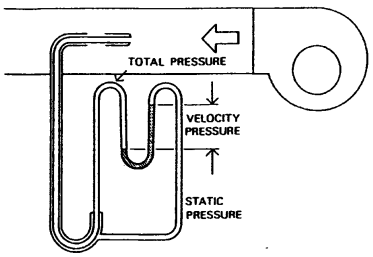
- Static Pressure (SP)
  - The pressure exerted by air on a surface at rest
  - Commonly used to specify fan performance
  - SP = total pressure (TP) - velocity pressure (VP)



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Velocity Pressure

- Fan Velocity Pressure (VP)
  - The pressure corresponding to the fan outlet velocity
  - The kinetic energy per unit volume of flowing air
  - VP = total pressure (TP) - static pressure (SP)



$VP = \rho (V / 1098)^2$

ρ = Density in $lb./ft^3$

V = Velocity in FPM

if standard air:

$VP = (V / 4009)^2$

Airflow (CFM) = V * Area (ft^2)

$VP = 0.5 \rho V^2$

ρ = Density in Kg/m^3

V = Velocity in MPS

if standard air:

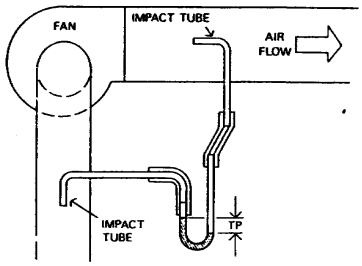
$VP = (V / 1.3)^2$

Airflow (CMS) = V * Area (m^2)

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Total Pressure

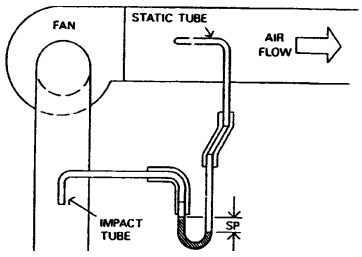
- Fan Total Pressure (FTP)
  - Total mechanical energy added to the air by the fan
  - FTP = TP at outlet - TP at inlet
  - TP = static pressure (SP) + velocity pressure (VP)



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Fan Static Pressure

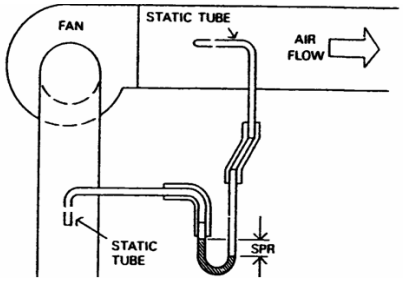
- Fan Static Pressure (FSP)
  - Used for ratings
  - The fan total pressure less the fan velocity pressure
  - FSP = SP at outlet - TP at inlet
  - SP = total pressure (TP) - velocity pressure (VP)



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Static Pressure Rise

- Static Pressure Rise (SPR, Heavy Industrial Fans)
  SPR = SP at outlet - SP at inlet



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Fan Power

- **Air Horsepower (AHP)**
  - Assuming 100% efficiency, the horsepower required to move a given volume of air against given pressure (IP units)
  - Static AHP = (CFM x SP)/**6343** or Static AKW = (CMS x SP)/**1000**
  - Total AHP = (CFM x TP) / **6343** or Total AKW = (CMS x TP)/**1000**

- **Brake Horsepower (BHP)**
  - The actual horsepower a fan requires
  - Determined by testing
  - BHP > AHP, because a fan is not 100% efficient
  - In addition to fan power input, BHP may include power absorbed by V-belt drives, accessories, etc.

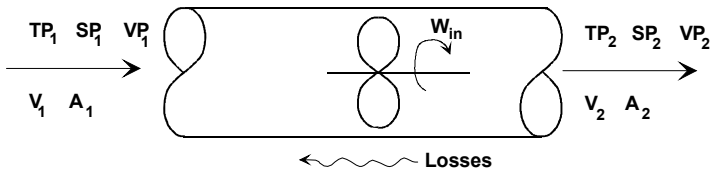
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Fan Efficiency

- **Static Efficiency (SE)**
  - Ratio of fan power output to the fan power input
  - Uses SP, which does not include kinetic energy
  - SE = AHP_{SP} / BHP
  - SE = (CFM x SP)/(**6343**xBHP) or SE = (CMS x SP)/(1000xKW)
- **Mechanical Efficiency (ME) or Total Efficiency, (TE)**
  - Ratio of total fan power output to the fan power input
  - Uses TP, which includes kinetic energy
  - TE = AHP_{TP} / BHP
  - TE = (CFM x TP)/(**6343**xBHP) or TE = (CMS x TP)/(1000 x KW)

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Relating Fan Power, Flow, Pressure



Energy equation

$TP_1 + W_{in} = TP_2 + Losses$

and since TP = SP + VP

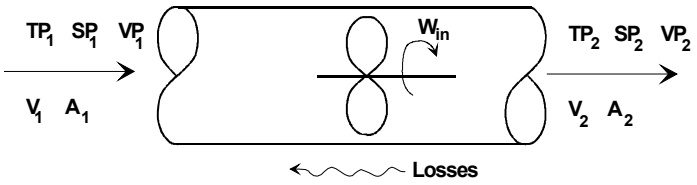
$VP_1 + SP_1 + W_{in} = VP_2 + SP_2 + Losses$

W_{in} is shaft power transferred to the air

Losses are due to fan inefficiencies and flow losses

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Relating Fan Power, Flow, Pressure



Fan Airflow (CFM)

CFM is based on continuity

$CFM = V_1 A_1 = V_2 A_2$

Fan TP (iwc)

$TP_F = TP_2 - TP_1$

$TP_F = SP_2 + VP_2 - (SP_1 + VP_1)$

Fan SP (iwc)

$SP_F = TP_F - VP_2$

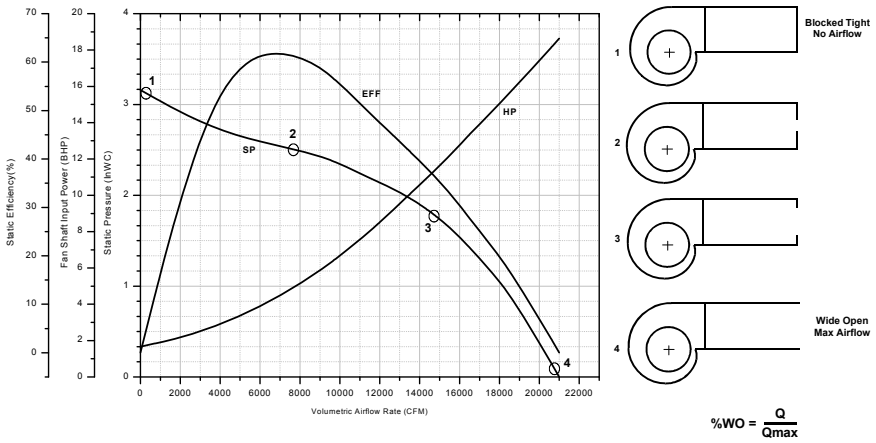
$SP_F = SP_2 + VP_2 - (SP_1 + VP_1) - VP_2$

$SP_F = SP_2 - (SP_1 + VP_1)$

$SP_F = SP_2 - TP_1$

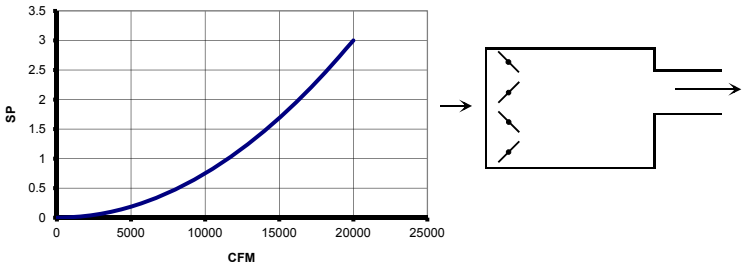
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Fan Performance Curve



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System Curve



Pressure Loss = ff (V^X) where X = 2 for turbulent flow

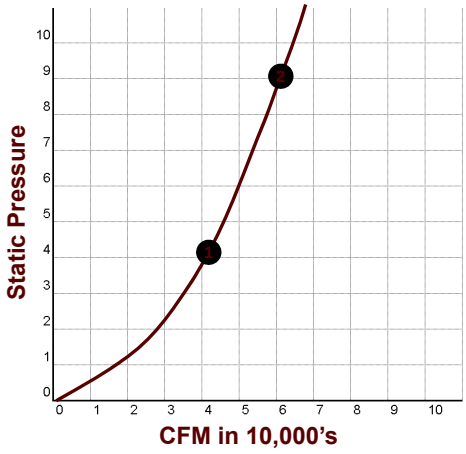
Pressure Loss ~ SP ~ V^2 ~ Q^2

$SP_{Loss} = kQ^2$ System curve ? $k = \frac{SP}{Q^2}$

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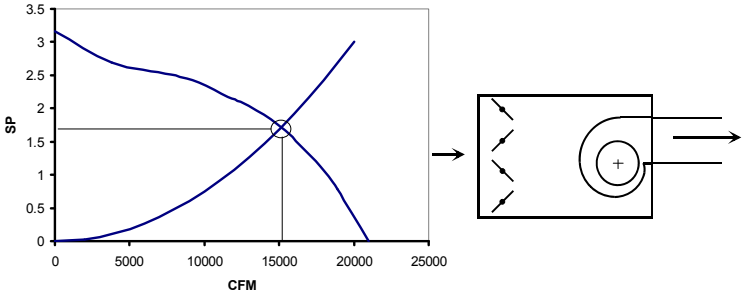
System Resistance Curve - Example

- Point 1:
  - 40,000 CFM
  - 4 in. wg. SP
- Point 2:
  - 60,000 CFM
  - 9 in. wg. SP



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Fan Operating Point

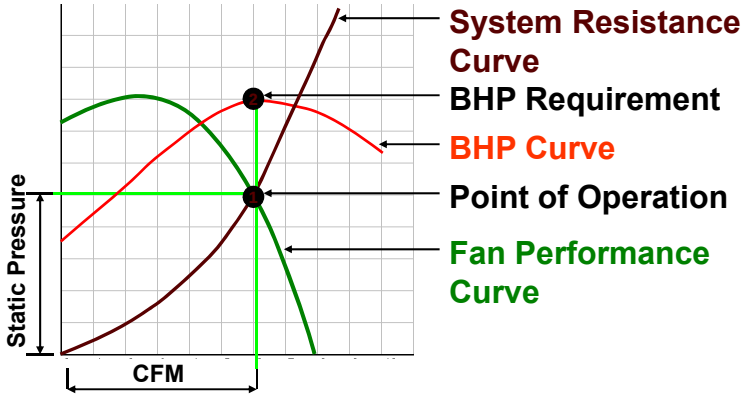


The intersection of the fan performance curve and the system curve is the fan operating point

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Fan & System – Point of Operation

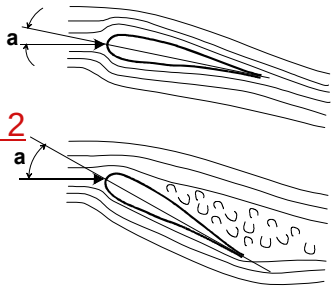
- A fan will operate at the point where the system resistance curve intersects the fan curve

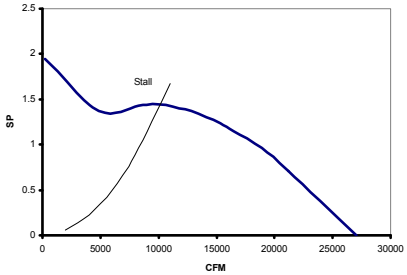


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Fan Stall

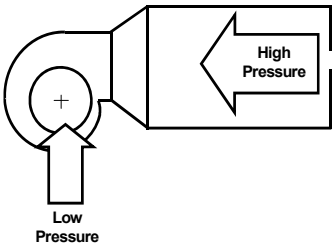
- Aerodynamic effect
- Fan unstable operation
- Less severe than surge
- Movie Clip 1 Movie Clip 2

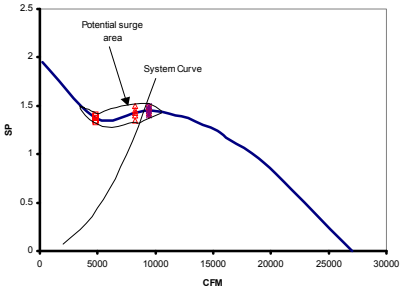




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Fan Surge

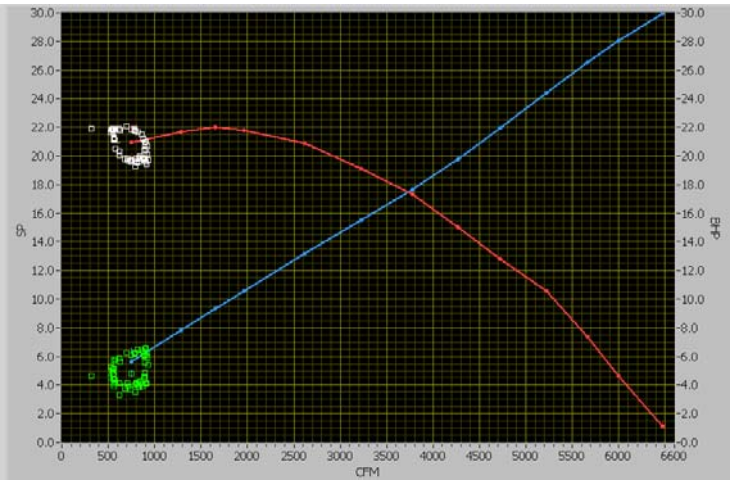




- Interaction with system
- Fan operation is very unstable
- Pulsating flow at inlet and discharge
- Movie clips: Normal Surge

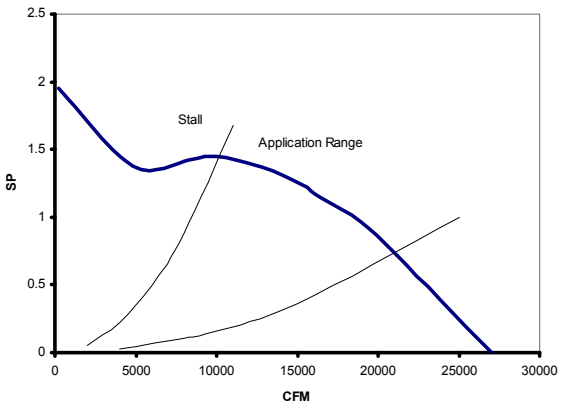
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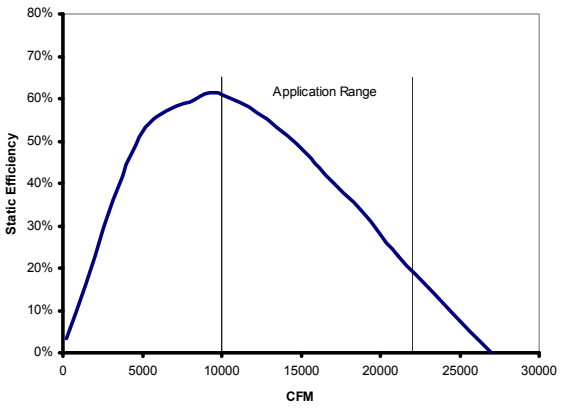
Fan Surge



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Fan Useful Application Range

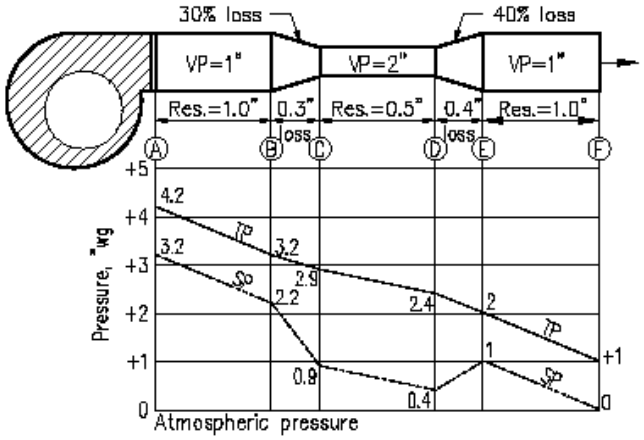




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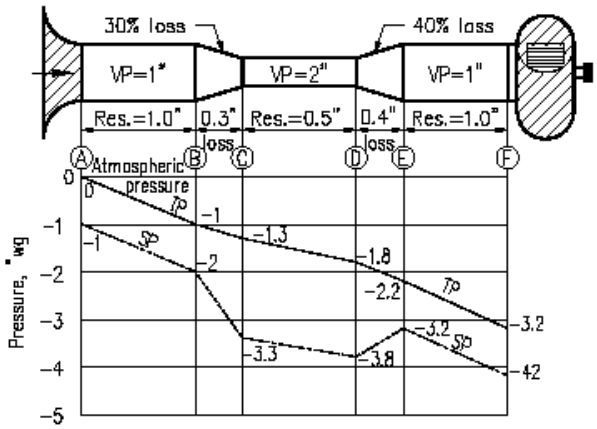
Fan & System Pressure Relationships

- Duct located on discharge side of fan

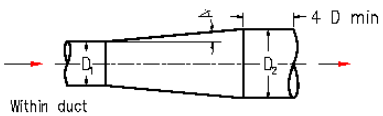


Fan & System Pressure Relationships

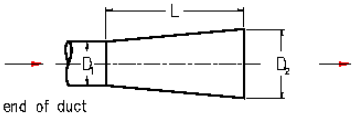
- Duct located on suction side of fan



SP Regain for Expansions



Within duct



At end of duct

Regain (R), fraction of VP difference					
Taper angle degrees	Diameter ratios D_2/D_1				
	1.25:1	1.5:1	1.75:1	2:1	2.5:1
3 1/2	0.92	0.88	0.84	0.81	0.75
5	0.88	0.84	0.80	0.76	0.68
10	0.85	0.76	0.70	0.63	0.53
15	0.83	0.70	0.62	0.55	0.43
20	0.81	0.67	0.57	0.48	0.43
25	0.80	0.65	0.53	0.44	0.28
30	0.79	0.63	0.51	0.41	0.25
Abrupt 90	0.77	0.62	0.50	0.40	0.25
Where: $SP_2 = SP_1 + R(VP_1 - VP_2)$					

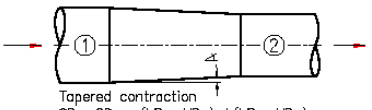
Regain (R), fraction of inlet VP						
Taper length to inlet diam L/D	Diameter ratios D_2/D_1					
	1.2:1	1.3:1	1.4:1	1.5:1	1.6:1	1.7:1
1.0:1	0.37	0.39	0.38	0.35	0.31	0.27
1.5:1	0.39	0.46	0.47	0.46	0.44	0.41
2.0:1	0.42	0.49	0.52	0.52	0.51	0.49
3.0:1	0.44	0.52	0.57	0.59	0.60	0.59
4.0:1	0.45	0.55	0.60	0.63	0.63	0.64
5.0:1	0.47	0.56	0.62	0.65	0.66	0.68
7.5:1	0.48	0.58	0.64	0.68	0.70	0.72
Where: $SP_1 = SP_2 - R(VP_1)$*						

*When SP_2 =0 (atmosphere) SP_1 will be (−)

The regain (R) will only be 70% of value shown above when expansion follows a disturbance or elbow (including a fan) by less than 5 duct diameters.

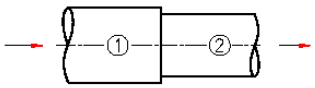
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SP Losses for Contractions



Tapered contraction
$SP_2 = SP_1 - (VP_2 - VP_1) - L(VP_2 - VP_1)$

Taper angle degrees	L(loss)
5	0.05
10	0.06
15	0.08
20	0.10
25	0.11
30	0.13
45	0.20
60	0.30
over 60	Abrupt contraction



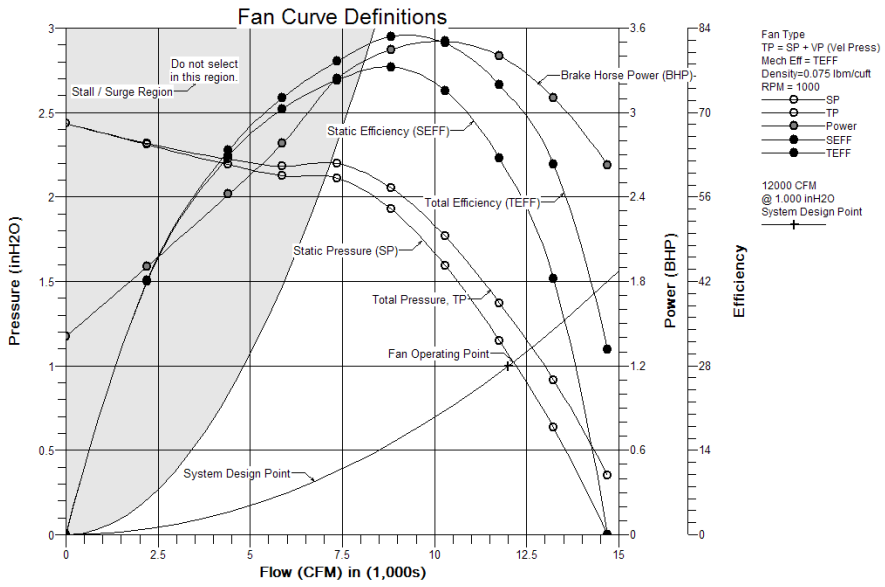
Abrupt contraction
$SP_2 = SP_1 - (VP_2 - VP_1) - K(VP_2)$

Ratio A_2/A_1	K
0.1	0.48
0.2	0.46
0.3	0.42
0.4	0.37
0.4	0.32
0.6	0.26
0.7	0.20

A= duct area, ft^2

Note:
In calculating SP for expansion or contraction use algebraic signs: VP is (+), and usually SP is (+) in discharge duct from fan, and SP is (−) in inlet duct to fan.

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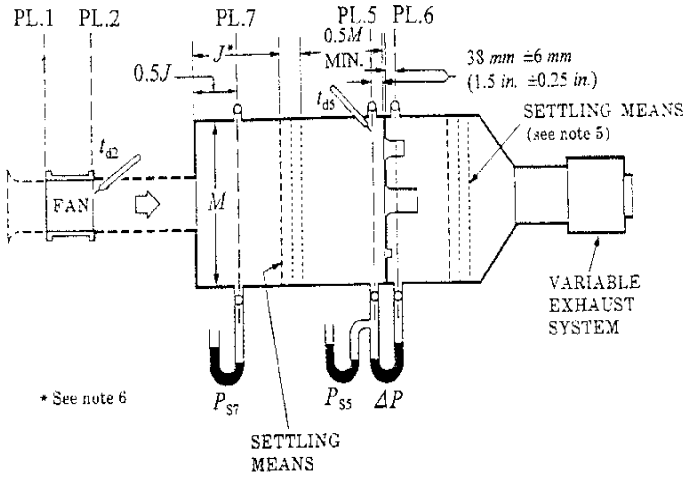
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Performance Lab Test Setup:
AMCA 210 Fig 12 Air
AMCA 300 Inlet Sound



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AMCA 210 Fig 12 Test Setup Multi Nozzel Chamber



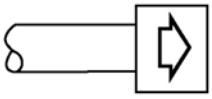
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Fan Testing and Rating

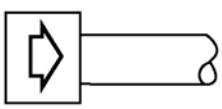
- Fans are tested in setups that simulate installations
  - Four standard installation types:



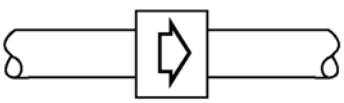
Type A: Free Inlet, Free Outlet



Type C: Ducted Inlet, Free Outlet



Type B: Free Inlet, Ducted Outlet

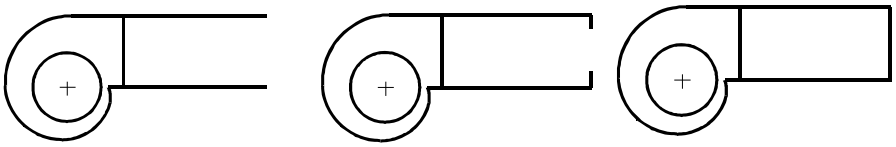


Type D: Ducted Inlet, Ducted Outlet

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Fan Testing and Rating

- Fans are tested at a given speed (RPM) per AMCA std 210-16.
- Volumetric flow rate (CFM or CMS) is measured at:
  - No resistance (wide open volume or free delivery),
  - Complete resistance (shut off), and
  - Everything in between



No Resistance Complete Resistance

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Fan Testing and Rating

- Not all fan sizes are tested for rating. AMCA std 211 provides the CRP for air performance
  - By use of equations called the fan laws test data is used to calculate performance of a larger fan that is geometrically similar
  - Not practical to test a fan at every speed
  - By use of the fan laws it is possible to predict fan performance at various speeds
- Fans are tested & rated at standard air density
  - Fan performance can be predicted at varying air densities by using the fan laws

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Fan Laws

- Dynamic Similitude
- Effect of RPM Change, Law 1
- Effect of Fan Diameter Change, Law 2
- Effects of Fan Diameter and RPM Change, Law 3
- Density Effects, Law 4
- Summary of Fan Laws
- Standard Air

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Dynamic Similitude

- Flow systems are considered to be dynamically similar if:
  - they are geometrically similar
  - the forces acting in one system are in the same ratio to each other as similar forces in the second system
- Geometric similarity
  - all physical dimensions must have the same linear ratio,
  - all surface roughness properties must scale,
  - for fans, the number of blades must be equal,
  - all angles, flow directions and orientations must be maintained.
- Dynamic similarity
  - the dimensionless analysis parameters (e.g. Re number) are the same for both systems
- **Fan Laws are only applicable for dynamically similar systems**

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Effect of RPM Change, Law 1

$$\frac{CFM_1}{CFM_2} = \frac{RPM_1}{RPM_2}$$

$$\frac{SP_1}{SP_2} = \left[\frac{RPM_1}{RPM_2}\right]^2$$

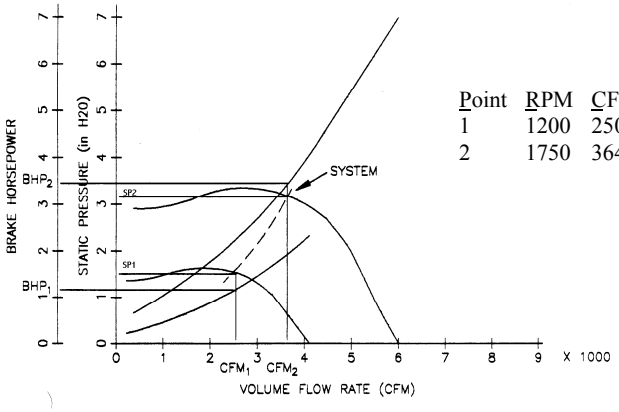
$$\frac{BHP_1}{BHP_2} = \left[\frac{RPM_1}{RPM_2}\right]^3$$

Efficiency will not change.

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Effect of RPM Change, Law 1: Along system resistance curve

- Illustration of Fan Law 1



Point	RPM	CFM	SP	BHP
1	1200	2500	1.45	1.10
2	1750	3646	3.08	3.41

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Effect of Fan Diameter Change, Law 2

$$\frac{CFM_1}{CFM_2} = \left[\frac{D_1}{D_2}\right]^3$$

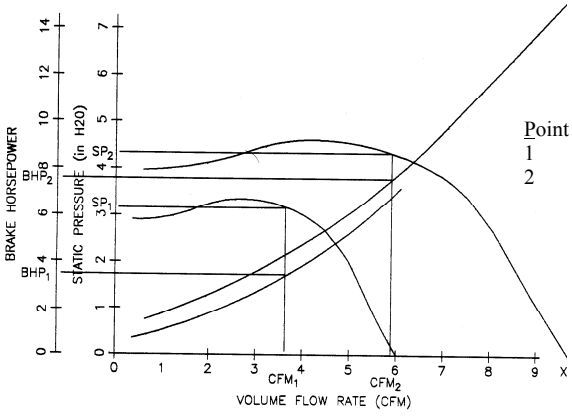
$$\frac{SP_1}{SP_2} = \left[\frac{D_1}{D_2}\right]^2$$

$$\frac{BHP_1}{BHP_2} = \left[\frac{D_1}{D_2}\right]^5$$

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Effect of Fan Diameter Change, Law 2: NOT along system resistance curve

- Illustration of Fan Law 2



Point	Dia	CFM	SP	BHP
1	10.5	3646	3.08	3.41
2	12.25	5789	4.19	7.37

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Effects of Fan Diameter and RPM Change, Law 3

$$\frac{CFM_1}{CFM_2} = \left[\frac{RPM_1}{RPM_2}\right]\left[\frac{D_1}{D_2}\right]^3$$

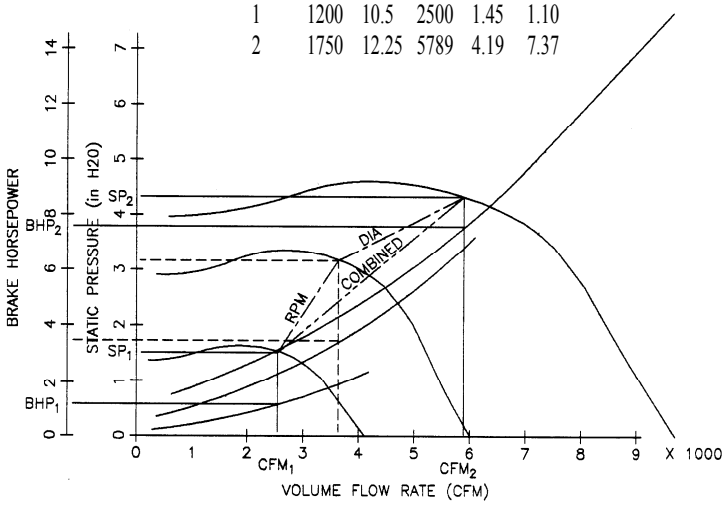
$$\frac{SP_1}{SP_2} = \left[\frac{RPM_1}{RPM_2}\right]^2\left[\frac{D_1}{D_2}\right]^2$$

$$\frac{BHP_1}{BHP_2} = \left[\frac{RPM_1}{RPM_2}\right]^3\left[\frac{D_1}{D_2}\right]^5$$

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Effects of Fan Diameter and RPM Change Law 3: NOT along system resistance curve

Point	RPM	Dia	CFM	SP	BHP
1	1200	10.5	2500	1.45	1.10
2	1750	12.25	5789	4.19	7.37



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Density Effects, Law 4

Fan volume in CFM will not change with density: $CFM_1 = CFM_2$

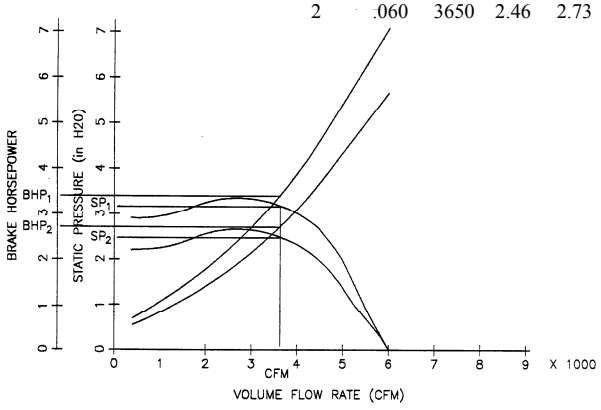
SP and BHP will vary in direct proportion to density: $\frac{SP_1}{SP_2} = \frac{BHP_1}{BHP_2} = \frac{\rho_1}{\rho_2}$

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Density Effects, Law 4

- Illustration of Fan Law 4

Point	ρ	CFM	SP	HP	Temp	Elevation
1	.075	3650	3.08	3.41	70 F	0 feet
2	.060	3650	2.46	2.73	203 F	6000 feet



Point	ρ	CFM	SP	HP	Temp	Elevation
1	.075	3650	3.08	3.41	70 F	0 feet
2	.060	3650	2.46	2.73	203 F	6000 feet

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Summary of Fan Laws

$$\frac{CFM_1}{CFM_2} = \left[\frac{D_1}{D_2}\right]^3 \left[\frac{RPM_1}{RPM_2}\right]$$

$$\frac{SP_1}{SP_2} = \left[\frac{D_1}{D_2}\right]^2 \left[\frac{RPM_1}{RPM_2}\right]^2 \left[\frac{\rho_1}{\rho_2}\right]$$

$$\frac{BHP_1}{BHP_2} = \left[\frac{D_1}{D_2}\right]^5 \left[\frac{RPM_1}{RPM_2}\right]^3 \left[\frac{\rho_1}{\rho_2}\right]$$

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Standard Air Density

- Standard air density = 0.075 lb./ft^3 or 1.2 kg/m^3
  - Air temperature of 70ºF (21ºC)
  - Altitude at 0 ft. elevation (sea level)
  - Barometric pressure of 29.92" Hg (101 kPA)
  - Specific volume of 13.33 ft^3/lb or 0.78m^3/kg

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ACFM vs. SCFM

- ACFM = actual cubic feet per minute
  - Represents the actual conditions of the job, not corrected to standard density conditions
  - Represents the volume of gas flowing anywhere in the system independent of air density
- SCFM = standard cubic feet per minute
  - Volume that gas would occupy if at standard density
  - Selecting a fan when SCFM is specified requires us to calculate the ACFM
  - SCFM is usually identified by specifying lbs./hour (mass flow rate)

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Temperature Effect

- Temperatures other than 70ºF affect density
- At temperatures above 70ºF, air density is less (lighter air), thus SP and BHP are less
- At temperatures below 70ºF, air density is greater (heavier air), thus SP and BHP are greater



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Altitude Effect

- Altitudes other than sea level affect density
- At altitudes above sea level, air density is less (lighter air), thus SP and BHP are less
- At altitudes below sea level, air density is greater (heavier air), thus SP and BHP are more



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Density Effects on Fan Performance

- The reduction in SP is proportional to the reduction in BHP, therefore the overall fan efficiency will remain unchanged

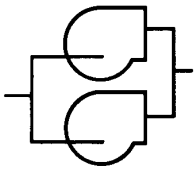
$$\text{Total Efficiency} = \frac{\text{CFM (TP x } d_1/d_2)}{\mathbf{6343}\ \text{(BHP x } d_1/d_2)} = \frac{\text{CFM (TP)}}{\mathbf{6343}\ \text{(BHP)}}$$

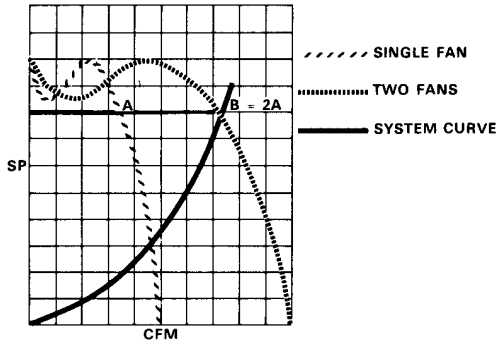




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Fans in Parallel

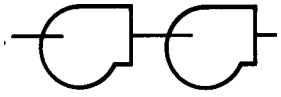


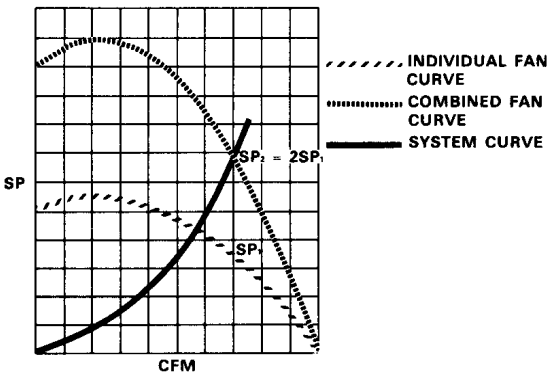


$SP_2 = SP_1$ $CFM_2 = 2\ CFM_1$ $BHP_2 = 2\ BHP_1$

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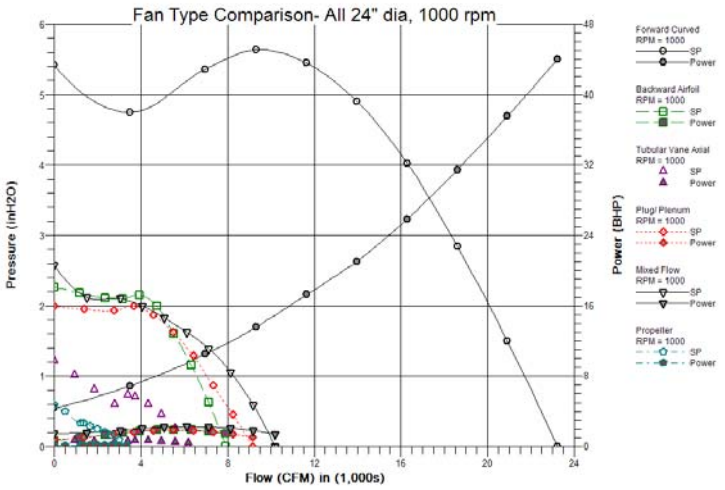
Fans in Series



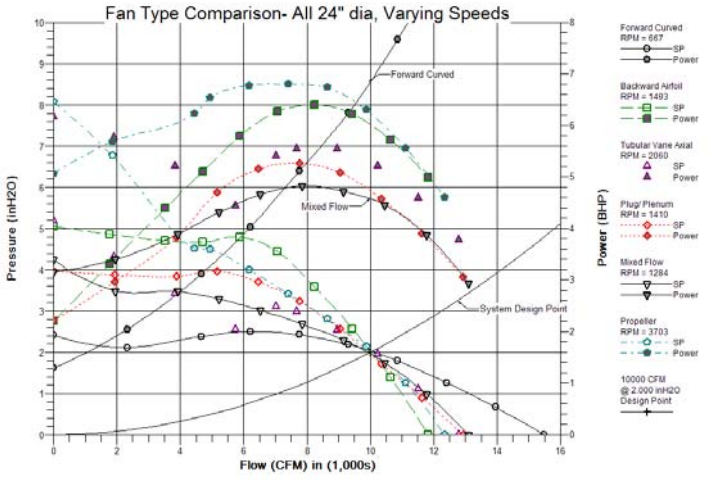


$SP_2 = 2\ SP_1$ $CFM_2 = CFM_1$ $BHP_2 = 2\ BHP_1$

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Fan Type Comparison- All 24" dia, 1000 rpm
Pressure (inH2O)
Power (BHP)
Flow (CFM) in (1,000s)
Forward Curved
RPM = 1000
SP
Power
Backward Airfoil
RPM = 1000
SP
Power
Tubular Vane Axial
RPM = 1000
SP
Power
Plug/ Plenum
RPM = 1000
SP
Power
Mixed Flow
RPM = 1000
SP
Power
Propeller
RPM = 1000
SP
Power

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Fan Type Comparison- All 24" dia, Varying Speeds
Pressure (inH2O)
Power (BHP)
Flow (CFM) in (1,000s)
Forward Curved
Mixed Flow
System Design Point
Forward Curved
RPM = 667
SP
Power
Backward Airfoil
RPM = 1493
SP
Power
Tubular Vane Axial
RPM = 2060
SP
Power
Plug/ Plenum
RPM = 1410
SP
Power
Mixed Flow
RPM = 1284
SP
Power
Propeller
RPM = 3703
SP
Power
10000 CFM
@ 2.000 inH2O
Design Point

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All fans selected at peak SE (Static Efficiency) for Airflow= 10,000 cfm, Static Pressure (SP)~2 iwc

	Type	Dia (in)	Spd (rpm)	BHP	SE % (Static Efficiency)	LwiA (Inlet Sound Power 'A')
1	Forward Curved- SW (Centrifugal)	30	476	5.09	61.7	89
2	Backward Airfoil – SW (Centrifugal)	36.5	650	3.82	80.0	77
3	Plenum	33	800	4.25	74.0	80
4	Tubular Mixed Flow	27	1074	4.48	70.2	81
5	Tubular Vane Axial	28	1438	4.77	65.9	86
6	Propeller (Axial)	30	1998	4.92	54.4	103

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Fan Selection Considerations

- Fan selection is not a trivial process for a given application.
- Example shown applies to one design operating point. The selections will change for other operating points.
- There is no magic fan that will result in least cost, best efficiency and low noise for a wide range of operating points.
- Compromises should be well understood upfront.
- Sometimes Direct Drive (DD) is preferred due to their simplicity. (no belts, drives, etc.) DD selection speeds may further limit selections. Varying width options can optimize DD selections.
- Mechanical design requirements like balancing and vibration levels, spark and high temp resistance, corrosion resistance, arrangements, bearings, motors and drives can further challenge the selection process.

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Reference Materials

- Twin City Fan FE-1600 - Temperature and Altitude Effects on Fans
- Twin City Fan FE-1700 - Density Corrections for Moist Gases
- Twin City Fan FE-100, Fan Performance Troubleshooting Guide
- ACGIH Worldwide, Industrial Ventilation - A Manual of Recommended Practice, 23rd Edition
- AMCA stds 210, 211
- http://www.tcf.com



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Thank you

Questions?

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