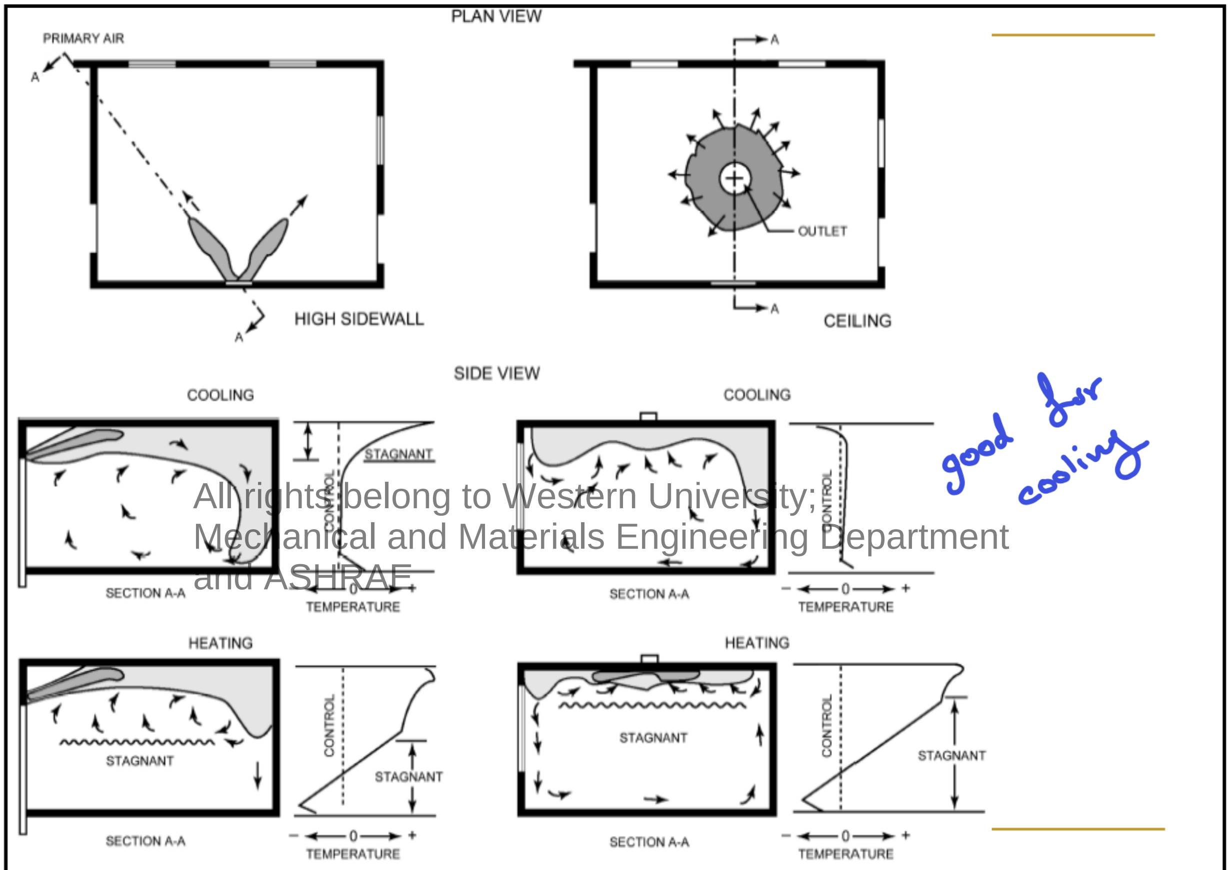
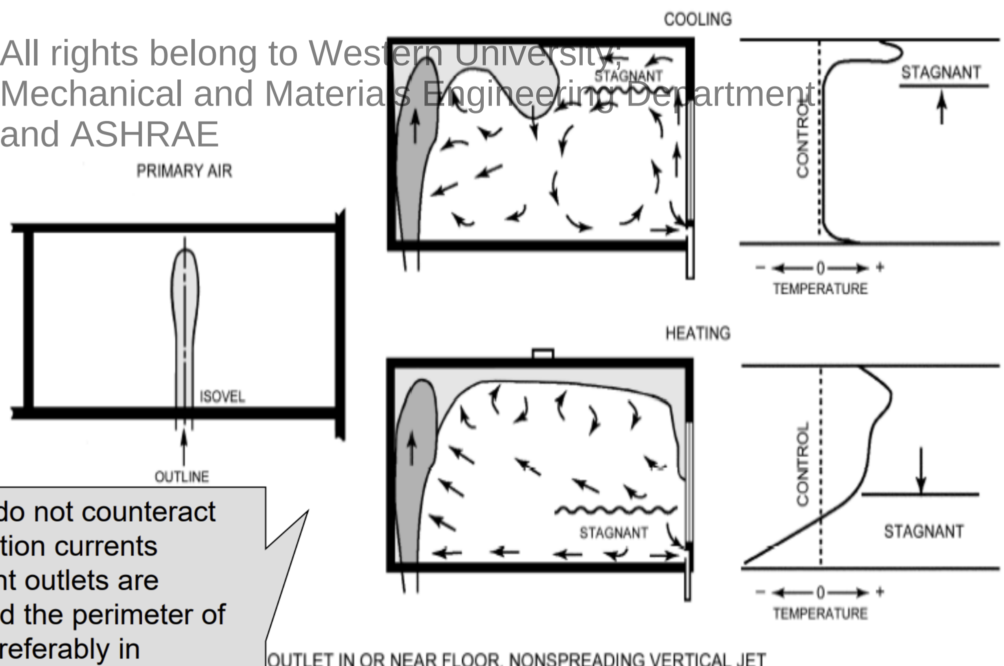
Diffuser Location



good for cooling

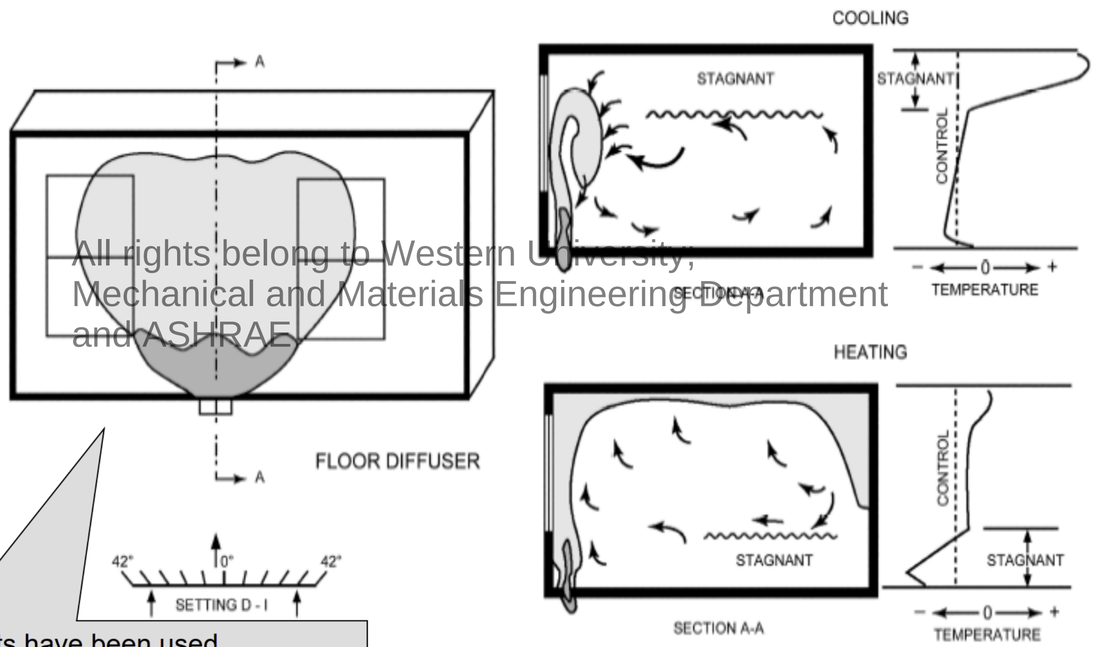
Fig. 4 Air Motion Characteristics of Group A Outlets



OUTLET IN OR NEAR FLOOR, NONSPREADING VERTICAL JET

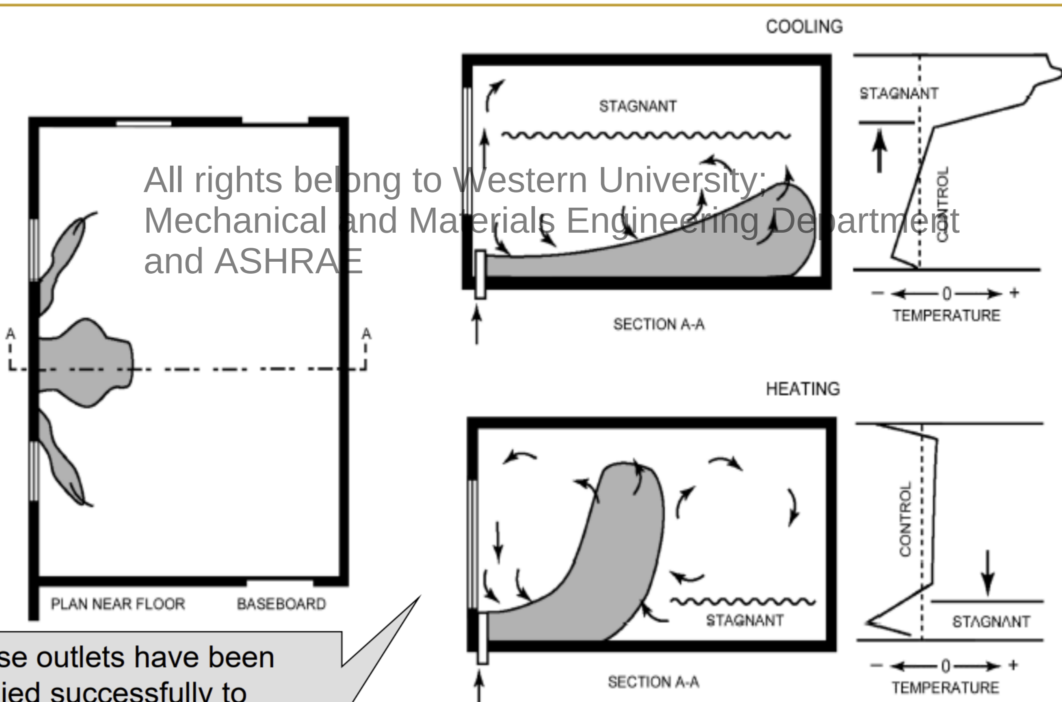
These outlets do not counteract natural convection currents unless sufficient outlets are installed around the perimeter of the space — preferably in locations of greatest heat gain or loss (under windows). The effect of drapes and blinds must be considered with outlets installed near windows.

Fig. 5 Air Motion Characteristics of Group B Outlets



The outlets have been used successfully for residential heating, but they may also offer a solution for applications where heating requirements are severe and cooling requirements are moderate.

Air Motion Characteristics of Group C Outlets



These outlets have been applied successfully to process installations where controlled air velocities are desired.

Air Motion Characteristics of Group D Outlets

outlets are used for either cooling or heating, but seldom for both.

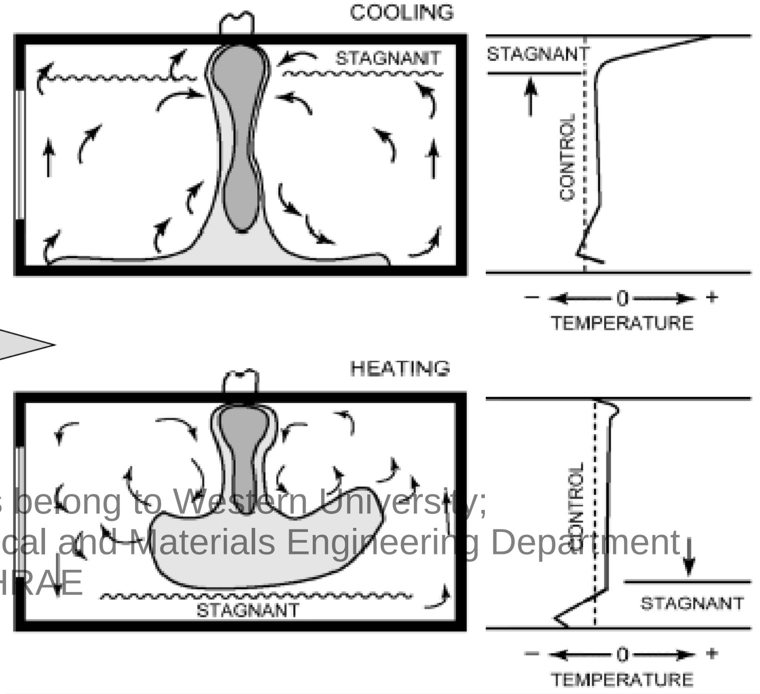


Fig. 8 Air Motion Characteristics of Group E Outlets

EDT & ADPI

Air-Distribution Performance Index (ADPI)

- The effective drat temperature (EDT) is the difference in temperature between any point in the occupied space and the control conditions:

$$\boxed{EDT = (t_x - t_r) - 0.07(V_x - 30)}$$

where:

t_x = local airstream dry temperature

t_r = room temperature

V_x = local airstream velocity

- Research indicates the high percentage of people in sedentary occupations are comfortable where EDT is between -3 F and +2 and the air velocity is less than 70 ft/min.
- The ADPI is defined as the percentage of measurements taken at many locations in the occupied zone of a space that meet the -3 F to +2 F EDT criteria.

→ ADPI = 100%

→ this is the main objective

How?

Table 11-2 Characteristic Room Length for Several Diffusers

Diffuser Type	Characteristic Length L
High sidewall grille	Distance to wall perpendicular to jet
Circular ceiling diffuser	Distance to closet wall or intersecting air jet
Sill grille	Length of room in direction of jet flow
Ceiling slot diffuser	Distance to wall or midplane between outlets
Light troffer diffusers	Distance to midplane between outlets plus distance from ceiling to top of occupied zone
Perforated, louvered ceiling diffusers	Distance to wall or midplane between outlets

Source: Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.

Table 11-3 Air Diffusion Performance Index (ADPI)

Terminal Device	Room Load, Btu/h-ft^2	x_{50}/L for Maximum ADPI	Maximum ADPI	For ADPI Greater Than	Range of x_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5–2.2
	40	1.6	78	70	1.2–2.3
	20	1.5	85	80	1.0–1.9
Circular ceiling diffusers	80	0.8	76	70	0.7–1.3
	60	0.8	83	80	0.7–1.2
	40	0.8	88	80	0.5–1.5
	20	0.8	93	90	0.7–1.3
Sill grille, Straight vanes	80	1.7	61	60	1.5–1.7
	60	1.7	72	70	1.4–1.7
	40	1.3	86	80	1.2–1.8
	20	0.9	95	90	0.8–1.3
Sill grille, Spread vanes	80	0.7	94	90	0.6–1.5
	60	0.7	94	80	0.6–1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	80	0.3	85	80	03.–0.7
	60	0.3	88	80	0.3–0.8
	40	0.3	91	80	0.3–1.1
	20	0.3	92	80	0.3–1.5
Light troffer diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated and louvered ceiling diffusers	11–51	2.0	96	90	1.4–2.7
				80	1.0–3.4

Source: Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.

Return Grills

Table 5 Recommended Return Inlet Face Velocities

Inlet Location	Velocity Across Gross Area, fpm
Above occupied zone	> 800
Within occupied zone, not near seats	600 to 800
Within occupied zone, near seats	400 to 600
Door or wall louvers	200 to 300
Through undercut area of doors	200 to 300

Noise Criteria

Noise

Table 11 Design Guidelines for HVAC-Related Background Sound in Rooms

Room Types	RC(N); QAI ≤ 5dB Criterion [a,b]
Residences, Apartments, Condominiums	25 – 35
Hotels/Motels	
Individual rooms or suites	25 – 35
Meeting/banquet rooms	25 – 35
Corridors, lobbies	35 – 45
Service/support areas	35 – 45
Office Buildings	
Executive and private offices	25 – 35
Conference rooms	25 – 35
Teleconference rooms	25 (max)
Open-plan offices	30 – 40
Corridors and lobbies	40 – 45
Hospitals and Clinics	
Private rooms	25 – 35
Wards	30 – 40
Operating rooms	25 – 35
Corridors and public areas	30 – 40
Performing Arts Spaces	
Drama theaters	25 (max)
Concert and recital halls [c]	
Music teaching studios	25 (max)
Music practice rooms	35 (max)

Room Type	NC
Laboratories (with fume hoods)	
Testing/research, minimal speech communication	45 – 55
Research, extensive telephone use, speech communication	40 – 50
Group teaching	35 – 45
Churches, Mosques, Synagogues	
General assembly	25 – 35
With critical music programs[c]	
Schools[d]	
Classrooms up to 750 ft^2	40 (max)
Classrooms over 750 ft^2	35 (max)
Large lecture rooms, without speech amplification	35 (max)
Libraries	30 – 40
Courtrooms	
Unamplified speech	25 – 35
Amplified speech	30 – 40
Indoor Stadiums, Gymnasiums	
Gymnasiums, natatoriums, and large seating-capacity spaces with speech amplification[e]	40 – 45

length of diffuser

drop

×150 ×100 ×50

Table 11-4 Performance Data for a Typical Linear Diffuser

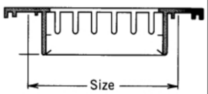
Size, in.	Area, ft²/ft	Total Pressure, in. wg	Flow, cfm/ft	NC[b]	Throw,[a] ft Min.	Mid.	Max.
2	0.055	0.009	22	—	1	1	1
		0.020	33	—	4	4	4
		0.036	44	12	7	7	7
		0.057	55	18	9	9	10
		0.080	66	23	11	11	12
		0.109	77	27	13	14	16
		0.143	88	31	14	16	18
		0.182	99	34	15	17	20
		0.225	110	37	17	19	21
3	0.096	0.009	38	—	2	2	2
		0.020	58	—	7	7	7
		0.036	77	11	10	10	11
		0.057	96	17	12	13	14
		0.080	115	22	15	16	17
		0.109	134	26	18	19	20
		0.143	154	30	20	21	23
		0.182	173	33	23	24	25
		0.225	192	36	25	25	26
4	0.139	0.009	56	—	3	3	3
		0.020	83	—	9	9	9
		0.036	111	12	13	13	13
		0.057	139	18	16	16	17
		0.080	167	23	20	20	21
		0.109	195	27	22	23	24
		0.143	222	31	24	25	26
		0.182	250	34	27	27	27
		0.225	278	37	30	30	30
5	0.179	0.009	72	—	4	4	4
		0.020	107	—	10	10	10
		0.036	143	12	14	14	14
		0.057	179	18	18	18	18
		0.080	215	23	22	22	23
		0.109	250	27	24	24	24

Air flow = 125 cfm
Throw = 10.4

Table 11-4 (*Continued*)

Active Length, ft	Multiplier Factor for Throw Value at Terminal Velocity, ft/min 150	100	50
1	0.5	0.6	0.7
10 or continuous	1.6	1.4	1.2

Active Length, ft	NC Correction	Active Length, ft	NC Correction
1	−10	10	0
2	−7	15	+2
4	−4	20	+3
6	−2	25	+4
8	−1	30	+5



[a]Minimum throw values refer to a terminal velocity of 150 ft/min, middle to 100 ft/min, and maximum to 50 ft/min, for a 4 ft active section with a cooling temperature differential of 20 F. The multiplier factors listed at the bottom are applicable for other lengths.

[b]Based on a room absorption of 80 dB referred to 10^{-12} W, and a 10 ft active section.

Source: Reprinted by permission of Environmental Elements Corporation, Dallas, Texas.

$x_{50} = 13 \times 0.7 = 9.1$

$x_{50} = 17 \times 0.7 = 11.9$

$NC = 15 - 10 = 5$

$\Delta P = (125/111)^2 (0.036) = 0.046$ in.wg

$$\left(\frac{cfm_1}{cfm_2}\right)^2 = \frac{\Delta P_1}{\Delta P_2}$$

Size, in.	cfm	Throw ft	NC	ΔP In.wg
4 x 12	125	10.5	15	0.046

Duct Design

FRICTION LOSSES

$$\Delta p_f = \frac{12fL}{D_h}\rho\left(\frac{V}{1097}\right)^2$$

where

Δp_f = friction losses in terms of total pressure, in. of water
f = friction factor, dimensionless
L = duct length, ft
D_h = hydraulic diameter [Equation (24)], in.
V = velocity, fpm
ρ = density, lb_m/ft^3

$$D_h = 4A/P$$ → for any shape

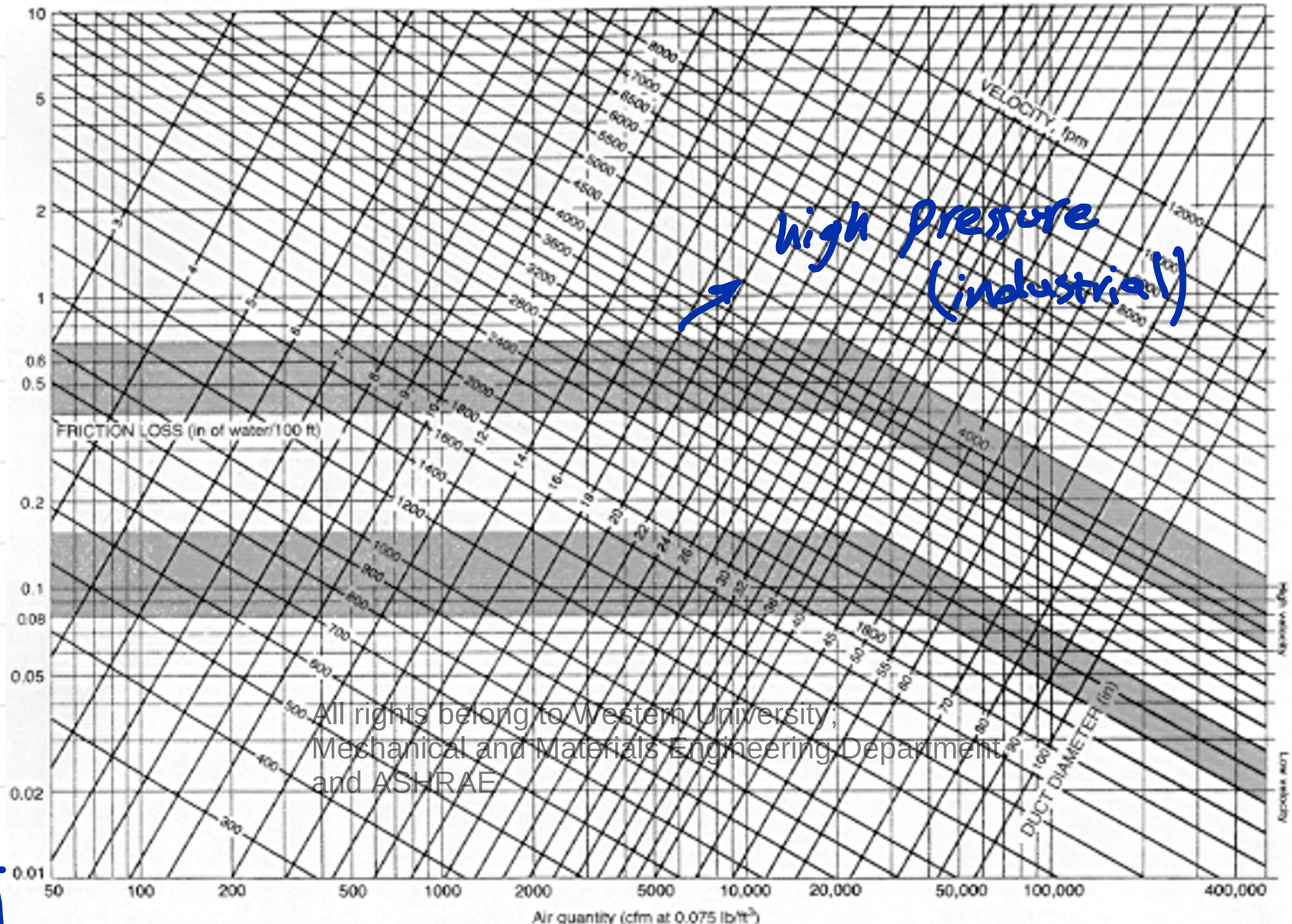
where

D_h = hydraulic diameter, in.
A = duct area, in^2
P = perimeter of cross section, in.

$$D_e = \frac{1.30(ab)^{0.625}}{(a+b)^{0.250}} \quad ($$

where

D_e = circular equivalent of rectangular duct for equal length, fluid resistance, and airflow, in.
a = length one side of duct, in.
b = length adjacent side of duct, in.



high pressure (industrial)

$\Delta P \left(\frac{\text{in of water}}{100\ \text{ft}}\right)$

air flow (cfm)

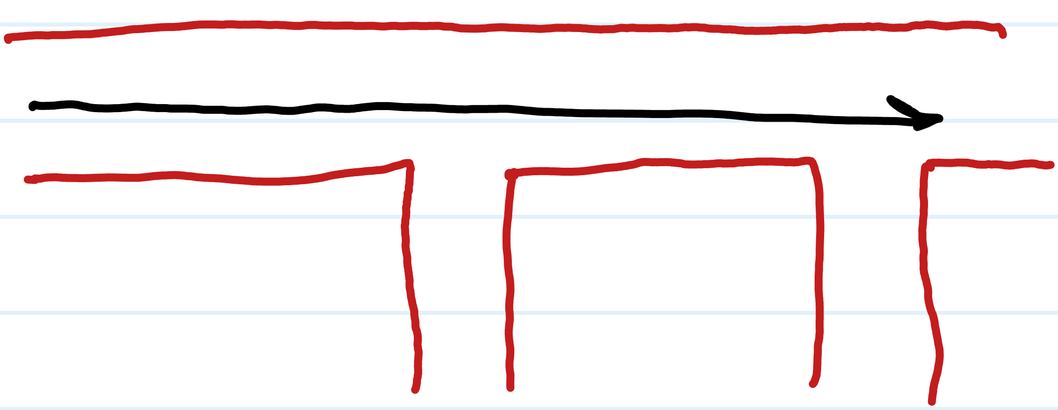
* The ranges have been selected based on the desired level of noise.

Based on the available room for installing the ducts, we can choose one of the following dimensions. But if we don't have limitation, best condition is $\frac{a}{b} > 1/3$

Table 2 Equivalent Rectangular Duct Dimensions

Circular Duct Diameter, in.	Length One Side of Rectangular Duct (a), in.																			
	4	5	6	7	8	9	10	12	14	16	18	20	22	24	26	28	30	32	34	36
	Length Adjacent Side of Rectangular Duct (b), in.																			
5	5																			
5.5	6	5																		
6	8	6																		
6.5	9	7	6																	
7	11	8	7																	
7.5	13	10	8	7																
8	15	11	9	8																
8.5	17	13	10	9																
9	20	15	12	10	8															
9.5	22	17	13	11	9															
10	25	19	15	12	10	9														
10.5	29	21	16	14	12	10														
11	32	23	18	15	13	11	10													
11.5		26	20	17	14	12	11													
12		29	22	18	15	13	12													
12.5		32	24	20	17	15	13													
13		35	27	22	18	16	14	12												
13.5		38	29	24	20	17	15	13												
14			32	26	22	19	17	14												
14.5			35	28	24	20	18	15												
15			38	30	25	22	19	16	14											
16			45	36	30	25	22	18	15											
17				41	34	29	25	20	17	16										
18				47	39	33	29	23	19	17										
19				54	44	38	33	26	22	19	18									
20					50	43	37	29	24	21	19									
21					57	48	41	33	27	23	20									
22					64	54	46	36	30	26	23	20								
23						60	51	40	33	28	25	22								
24						66	57	44	36	31	27	24	22							
25							63	49	40	34	29	26	24							
26							69	54	44	37	32	28	26	24						
27							76	59	48	40	35	31	28	25						
28								64	52	43	38	33	30	27	26					
29								70	56	47	41	36	32	29	27					
30								76	61	51	44	39	35	31	29	28				

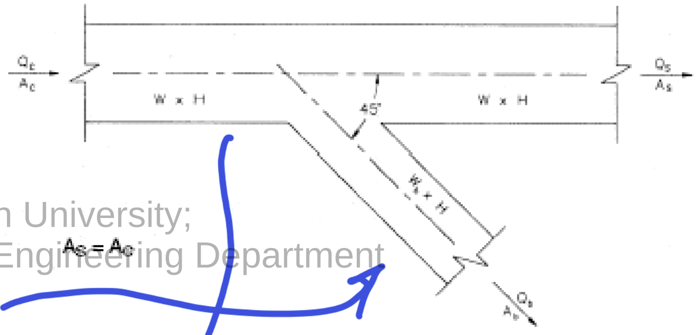
for the main line, we need to keep one dimension (usually height) constant



for the whole path.

Dynamic losses

SR5-3 Wye of the Type $A_s + A_b > A_c$, $A_s = A_c$, 45 Degree, Diverging

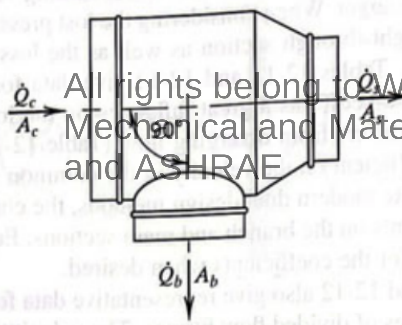


$A_s = A_c$

	C_b Values								
	Q_b/Q_c								
A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	0.60	0.52	0.57	0.58	0.64	0.67	0.70	0.71	0.73
0.2	2.24	0.56	0.44	0.45	0.51	0.54	0.58	0.60	0.62
0.3	5.94	1.08	0.52	0.41	0.44	0.46	0.49	0.52	0.54
0.4	10.56	1.88	0.71	0.43	0.35	0.31	0.31	0.32	0.34
0.5	17.75	3.25	1.14	0.59	0.40	0.31	0.30	0.30	0.31
0.6	26.64	5.04	1.76	0.83	0.50	0.36	0.32	0.30	0.30
0.7	37.73	7.23	2.56	1.16	0.67	0.44	0.35	0.31	0.30
0.8	49.92	9.92	3.48	1.60	0.87	0.55	0.42	0.35	0.32

Q_s/Q_c	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0
C_s	32.00	6.50	2.22	0.87	0.40	0.17	0.03	0.00

B. Diverging Tee, Round



$$\Delta P_j = C_{c,s}\left(\frac{V_s}{4005}\right)^2$$

$$\Delta P_j = C_{c,b}\left(\frac{V_b}{4005}\right)^2$$

	Branch, C_b								
A_b/A_c	$\dot{Q}_b/\dot{Q}_c = 0.1$	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	1.20	0.62	0.80	1.28	1.99	2.92	4.07	5.44	7.02
0.2	4.10	1.20	0.72	0.62	0.66	0.80	1.01	1.28	1.60
0.3	8.99	2.40	1.20	0.81	0.66	0.62	0.64	0.70	0.80
0.4	15.89	4.10	1.94	1.20	0.88	0.72	0.64	0.62	0.63
0.5	24.80	6.29	2.91	1.74	1.20	0.92	0.77	0.68	0.63
0.6	35.73	8.99	4.10	2.40	1.62	1.20	0.96	0.81	0.72
0.7	48.67	12.19	5.51	3.19	2.12	1.55	1.20	0.99	0.85
0.8	63.63	15.89	7.14	4.10	2.70	1.94	1.49	1.20	1.01
0.9	80.60	20.10	8.99	5.13	3.36	2.40	1.83	1.46	1.20

	Main, C_s								
A_s/A_c	$\dot{Q}_s/\dot{Q}_c = 0.1$	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	0.13	0.16							
0.2	0.20	0.13	0.15	0.16	0.28				
0.3	0.90	0.13	0.13	0.14	0.15	0.16	0.20		
0.4	2.88	0.20	0.14	0.13	0.14	0.15	0.15	0.16	0.34
0.5	6.25	0.37	0.17	0.14	0.13	0.14	0.14	0.15	0.15
0.6	11.88	0.90	0.20	0.13	0.14	0.13	0.14	0.14	0.15
0.7	18.62	1.71	0.33	0.18	0.16	0.14	0.13	0.15	0.14
0.8	26.88	2.88	0.50	0.20	0.15	0.14	0.13	0.13	0.14
0.9	36.45	4.46	0.90	0.30	0.19	0.16	0.15	0.14	0.13

Source: Reprinted by permission from *ASHRAE Duct Fitting Database*, 1992.

Thermal Gravity Effect (Stack Effect)

$$\Delta P_{se} = 0.192(\rho_a - \rho)(z_2 - z_1)$$

- *where*
- Δp_{se} = thermal gravity effect, in. of water
- z_1 and z_2 = elevation from datum in direction of airflow, ft
- ρ_a = density of ambient air, lbm/ft^3
- ρ = density of air or gas within duct, lbm/ft^3

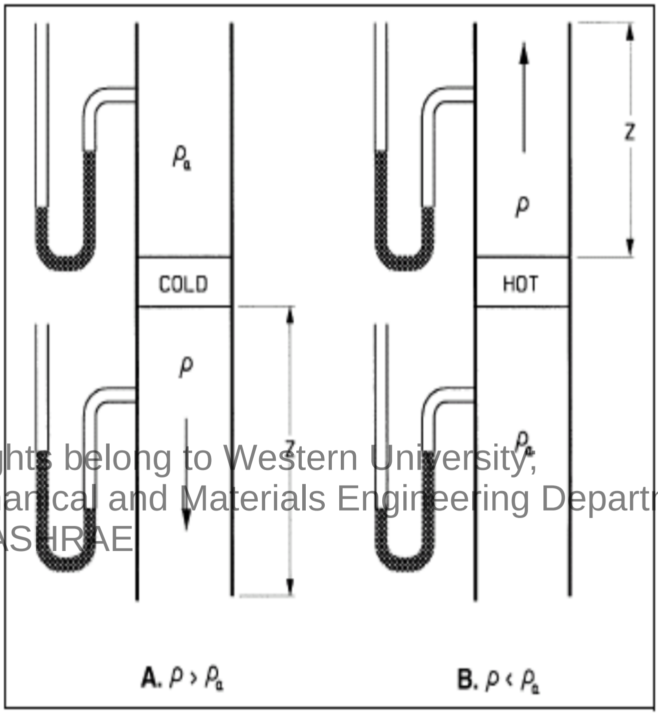


Fig. 1 Thermal Gravity Effect for Example 1

Duct leakage

$$C_L = Q/\Delta p_s^{0.65} \qquad (43)$$

where

Q = leakage rate, cfm/100 ft² (surface area)

C_L = leakage class, cfm per 100 ft duct surface at 1 in. of water static pressure

2

Table 9 Leakage as Percentage of Airflow[a,b]

Leakage Class	System cfm per ft² Duct Surface[c]	Static Pressure, in. of water					
		0.5	1	2	3	4	6
48	2	15	24	38	49	59	77
	2.5	12	19	30	39	47	62
	3	10	16	25	33	39	51
	4	7.7	12	19	25	30	38
	5	6.1	9.6	15	20	24	31
24	2	7.7	12	19	25	30	38
	2.5	6.1	9.6	15	20	24	31
	3	5.1	8.0	13	16	20	26
	4	3.8	6.0	9.4	12	15	19
	5	3.1	4.8	7.5	9.8	12	15
12	2	3.8	6	9.4	12	15	19
	2.5	3.1	4.8	7.5	9.8	12	15
	3	2.6	4.0	6.3	8.2	9.8	13
	4	1.9	3.0	4.7	6.1	7.4	9.6
	5	1.5	2.4	3.8	4.9	5.9	7.7
6	2	1.9	3	4.7	6.1	7.4	9.6
	2.5	1.5	2.4	3.8	4.9	5.9	7.7
	3	1.3	2.0	3.1	4.1	4.9	6.4
	4	1.0	1.5	2.4	3.1	3.7	4.8
	5	0.8	1.2	1.9	2.4	3.0	3.8
3	2	1.0	1.5	2.4	3.1	3.7	4.8
	2.5	0.8	1.2	1.9	2.4	3.0	3.8
	3	0.6	1.0	1.6	2.0	2.5	3.2
	4	0.5	0.8	1.3	1.6	2.0	2.6
	5	0.4	0.6	0.9	1.2	1.5	1.9

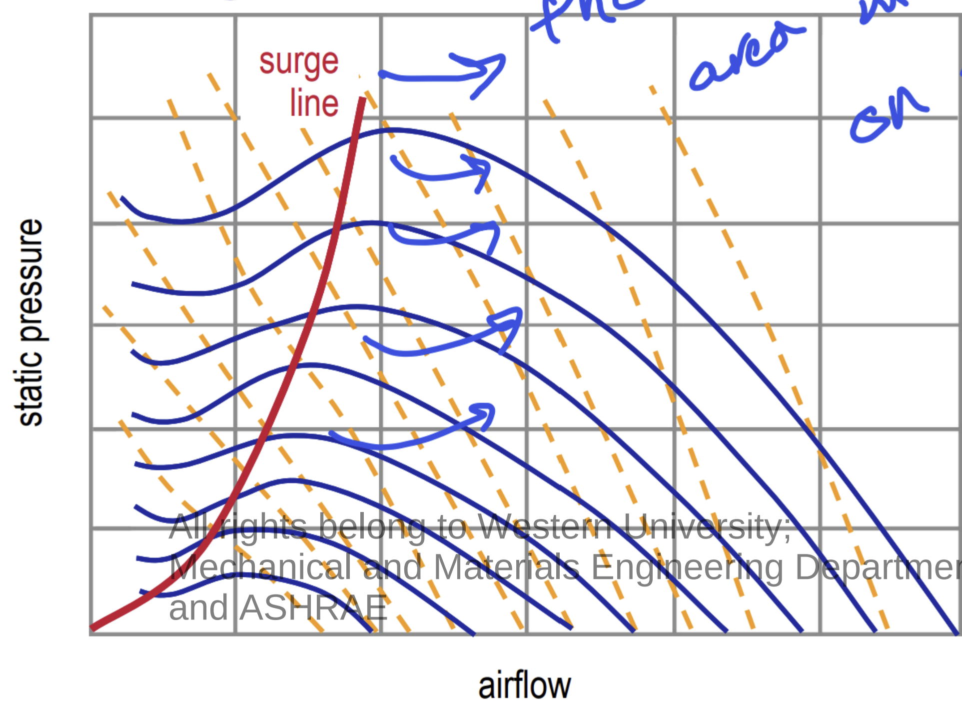
1

Table 6 Duct Leakage Classification[a]

Duct Type	Predicted Leakage Class C_L [Eq. (43)]	
	Sealed[b,c]	Unsealed[c]
Metal (flexible excluded)		
Round and flat oval	3	30 (6 to 70)
Rectangular		
≤ 2 in. of water (both positive and negative pressures)	12	48 (12 to 110)
> 2 and ≤ 10 in. of water (both positive and negative pressures)	6	48 (12 to 110)[c]
Flexible		
Metal, aluminum	8	30 (12 to 54)
Nonmetal	12	30 (4 to 54)
Fibrous glass		
Round	3	na
Rectangular	6	na

Fans

Fan Surge Line



the operating area must be on this side

$$\text{Static Efficiency (SE)} = \frac{\text{Power Out}}{\text{Power In}}$$

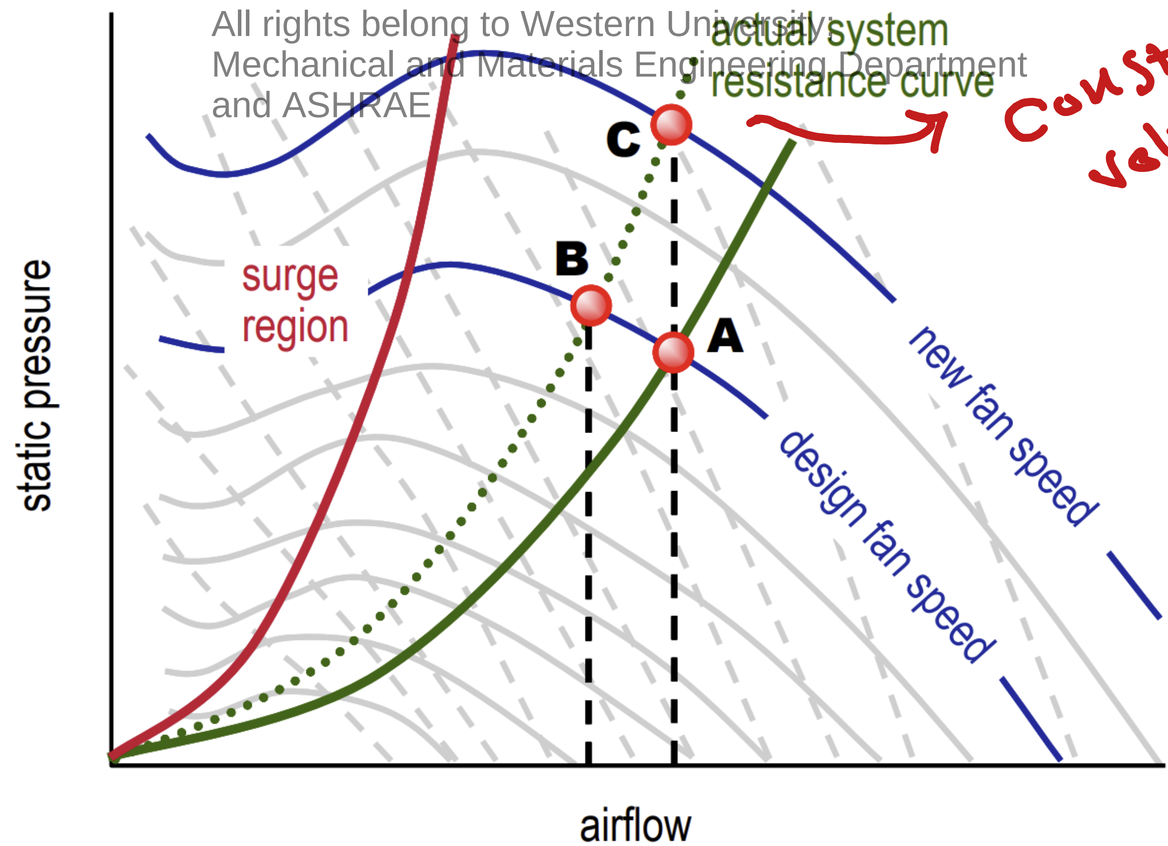
$$\text{SE} = \frac{\text{Airflow} \times \text{Static Pressure}}{\text{Constant} \times \text{Input Power}}$$

$$SE = \frac{3{,}500 \text{ cfm} \times 2.0 \text{ in. } H_2O}{6{,}362 \times 2.0 \text{ hp}} = 55\%$$

constant

$$SE = \frac{1.65 \text{ m}^3/\text{s} \times 498 \text{ Pa}}{1{,}000 \times 1.5 \text{ kW}} = 55\%$$

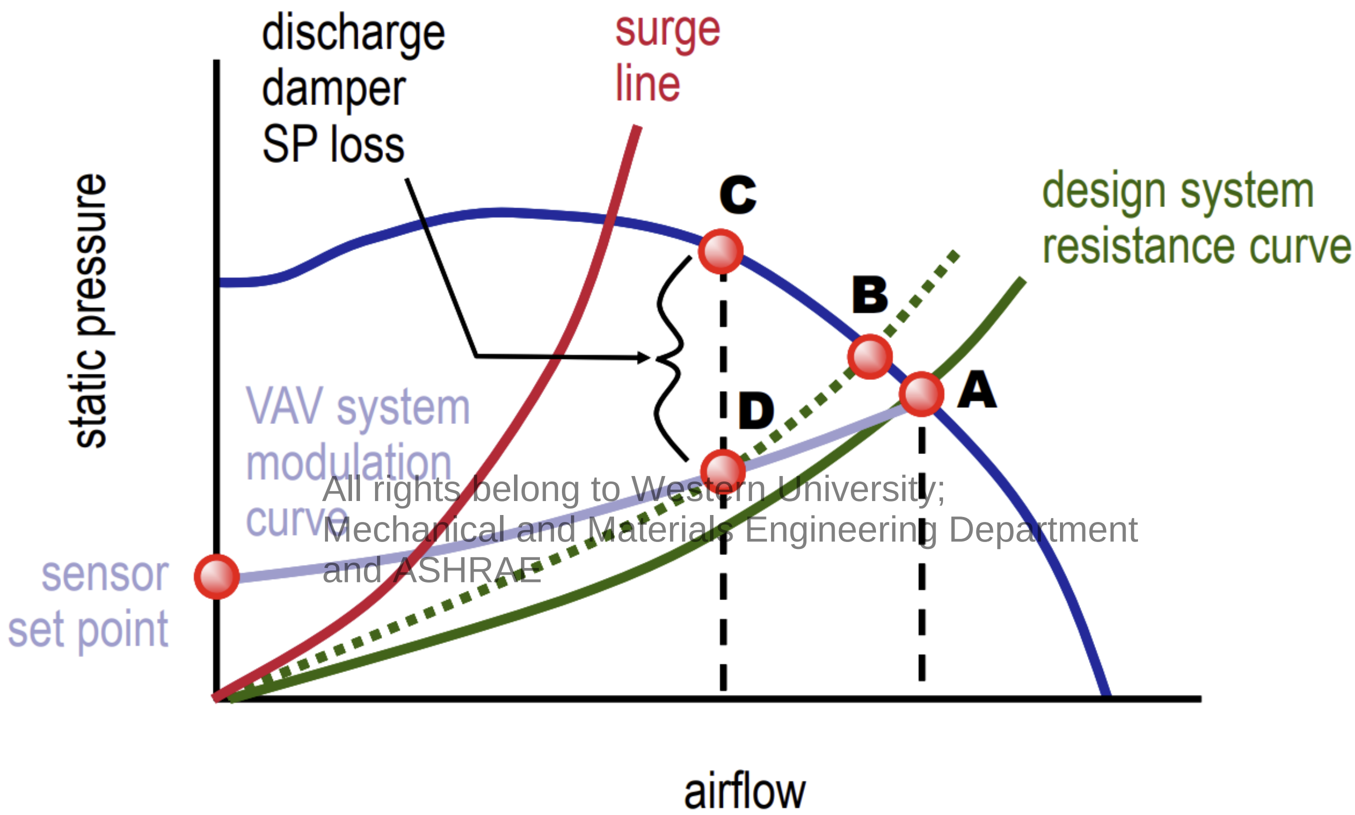
Higher System Resistance

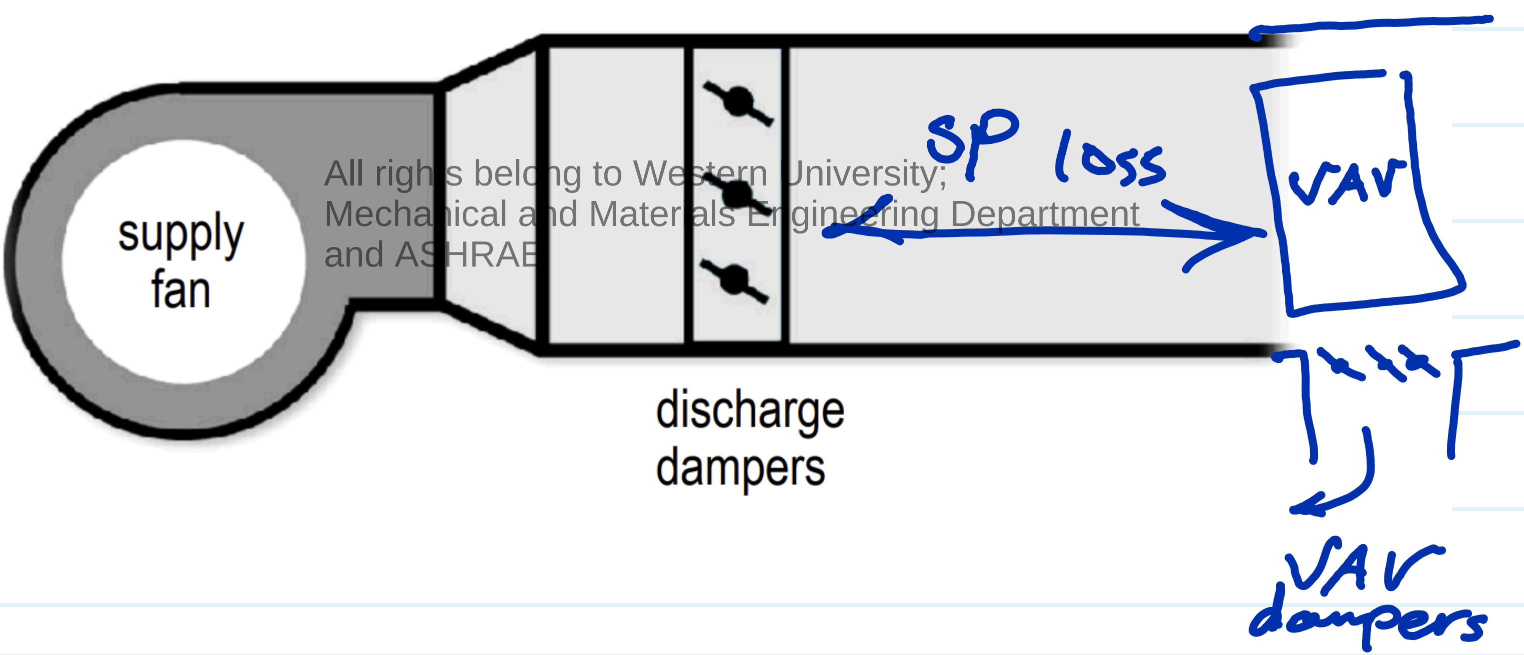


Constant volume sys

VAV & Discharge Dampers

Discharge Dampers





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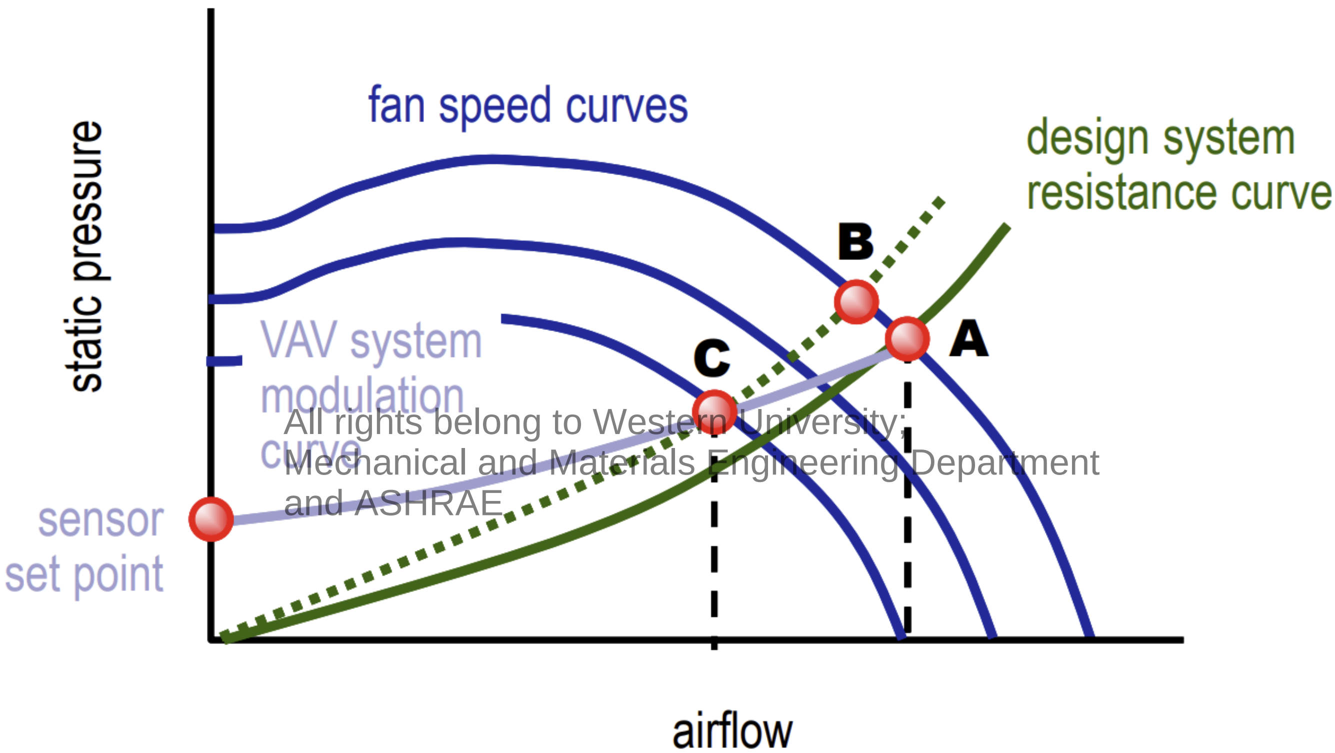
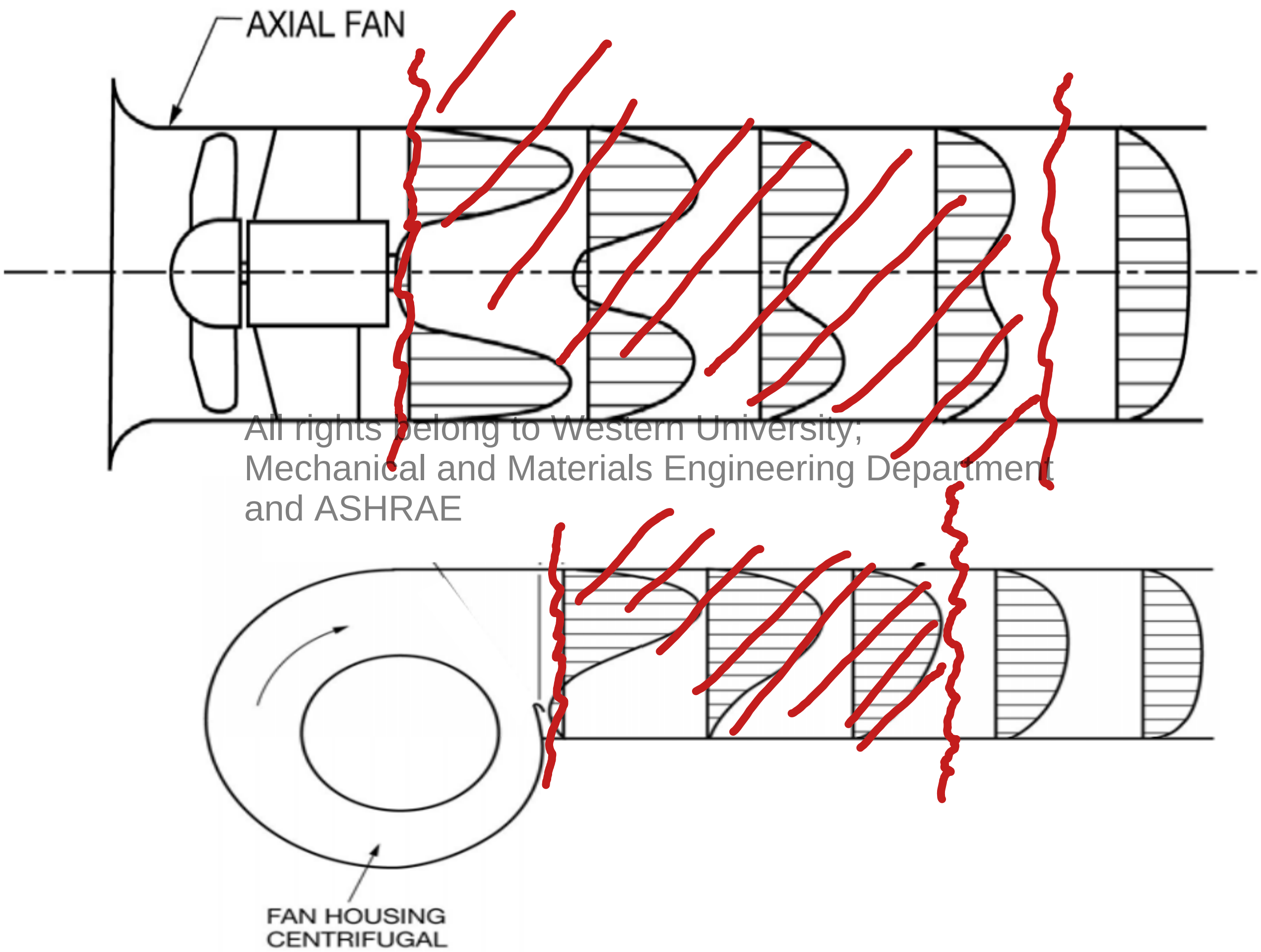
OR

Fan-Speed Control

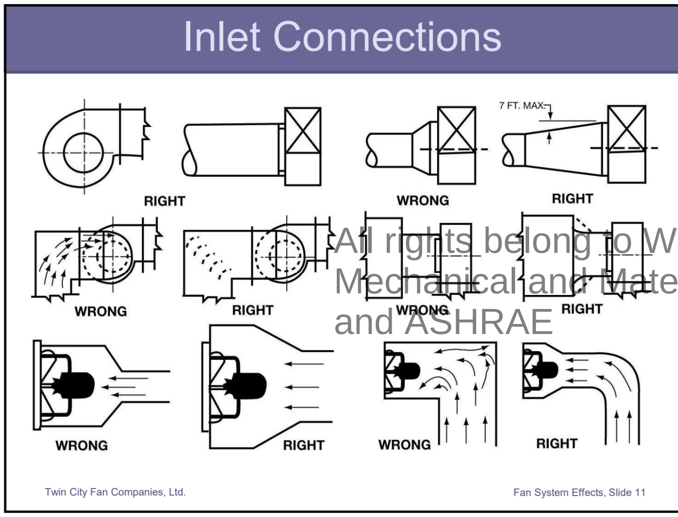


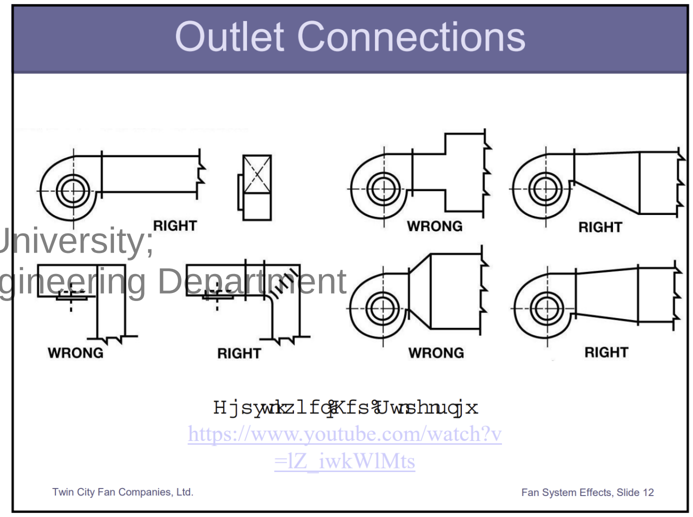
Fan System Effect

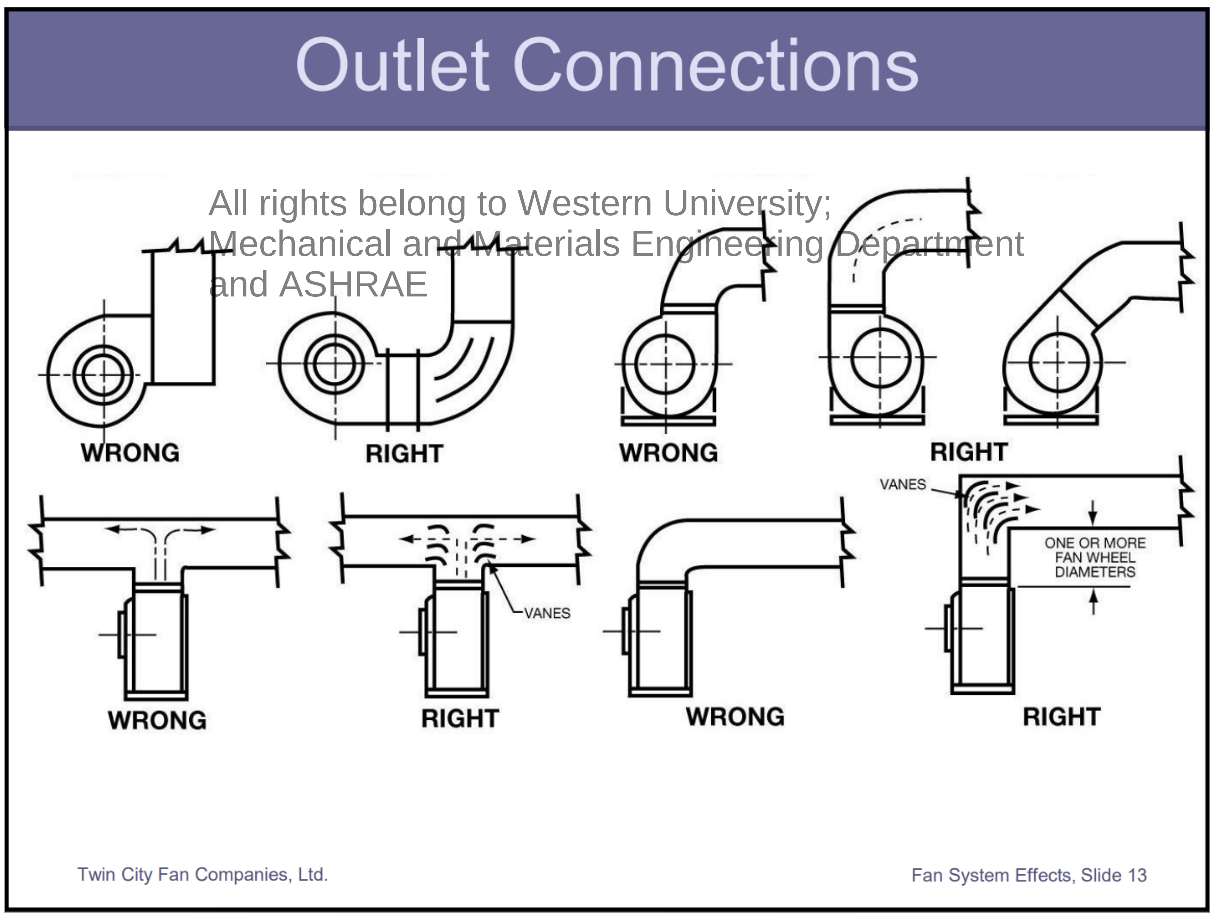
In other words, anything that is placed in close proximity before or after the fan that effects the catalogued performance.



anything placed in the red area will cause a considerable pressure drop.

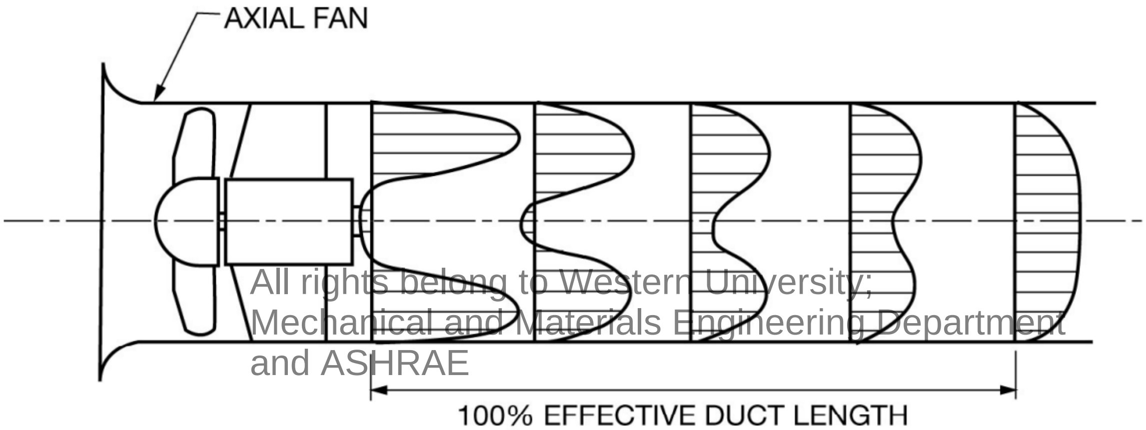






- Inlet induced losses tend to be higher than outlet losses
- Losses induced on the inlet can often exceed 20%
- Losses as high as 50% have been reported

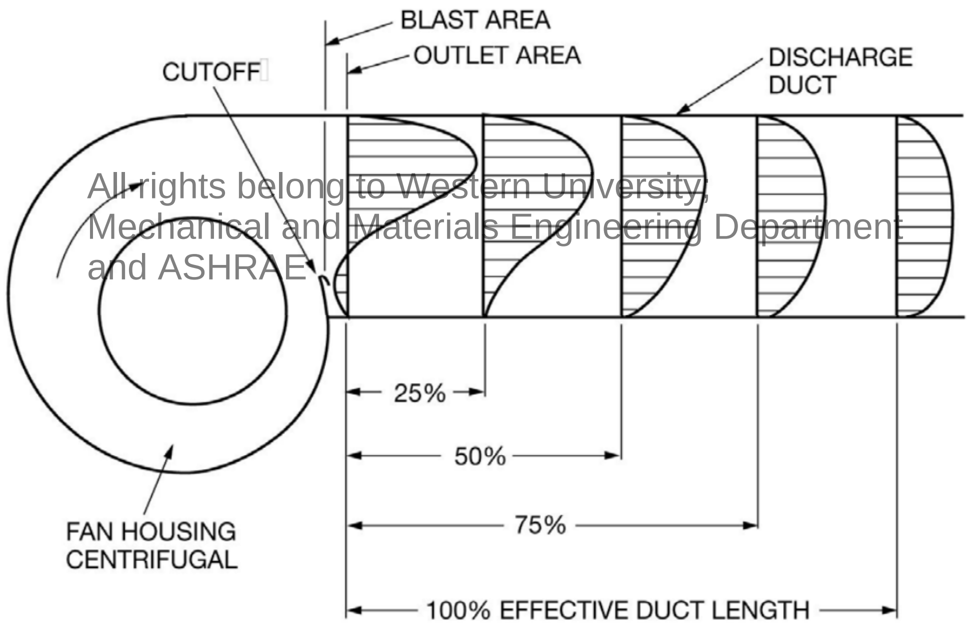
System Effect Curves for Outlet Ducts – Axial Fans



TO CALCULATE 100% EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 2 1/2 DUCT DIAMETERS FOR 2500 FPM OR LESS. ADD 1 DUCT DIAMETER FOR EACH ADDITIONAL 1000 FPM.

	NO DUCT	12% EFFECTIVE DUCT	25% EFFECTIVE DUCT	50% EFFECTIVE DUCT	100% EFFECTIVE DUCT
Tubeaxial Fan	—	—	—	—	—
Vaneaxial Fan	U	V	W	—	—

Fan Outlet Velocity Profile - Centrifugal



- If duct is rectangular with side dimensions a x b, the equivalent duct diameter is equal to (4*a*b/π)^0.5

System Effect Curves for Outlet Ducts - Centrifugal Fans

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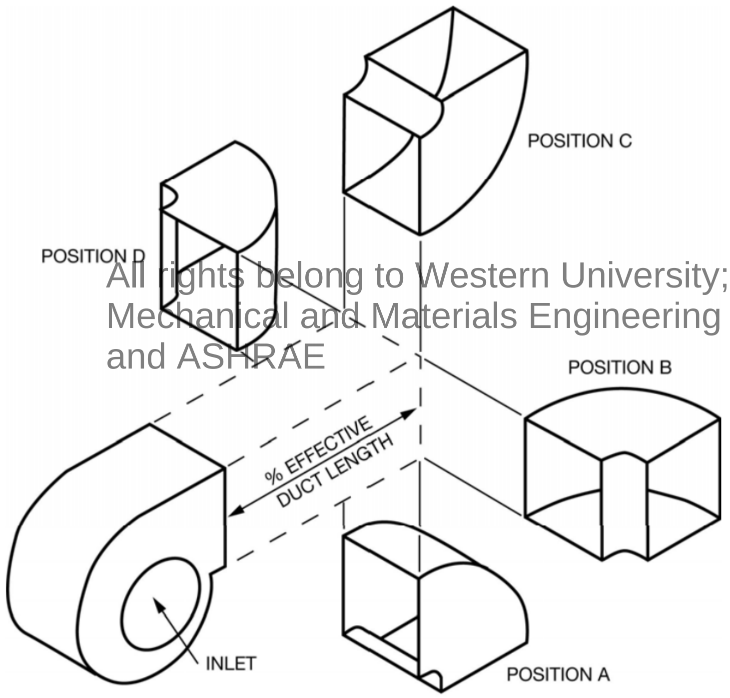
	NO DUCT	12% EFFECTIVE DUCT	25% EFFECTIVE DUCT	50% EFFECTIVE DUCT	100% EFFECTIVE DUCT
PRESSURE RECOVERY	0%	50%	80%	90%	100%
BLAST AREA / OUTLET AREA	SYSTEM EFFECT CURVE				
0.4	P	R-S	U	Q	—
0.5	P	R-S	U	W	—
0.6	R-S	S-T	U-V	W-X	—
0.7	S	U	W-X	—	—
0.8	T-U	V-W	X	—	—
0.9	V-W	W-X	—	—	—
1.0	—	—	—	—	—

Outlet Duct Elbows - Centrifugal Fans



Twin City Fan Companies, Ltd.

Fan System Effects, Slide 26

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	

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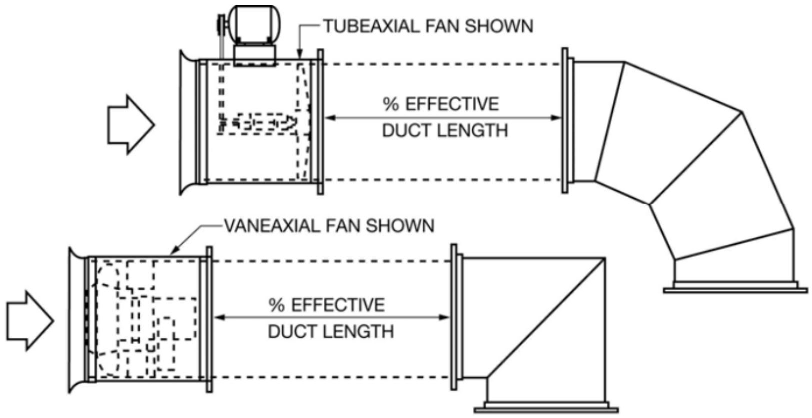
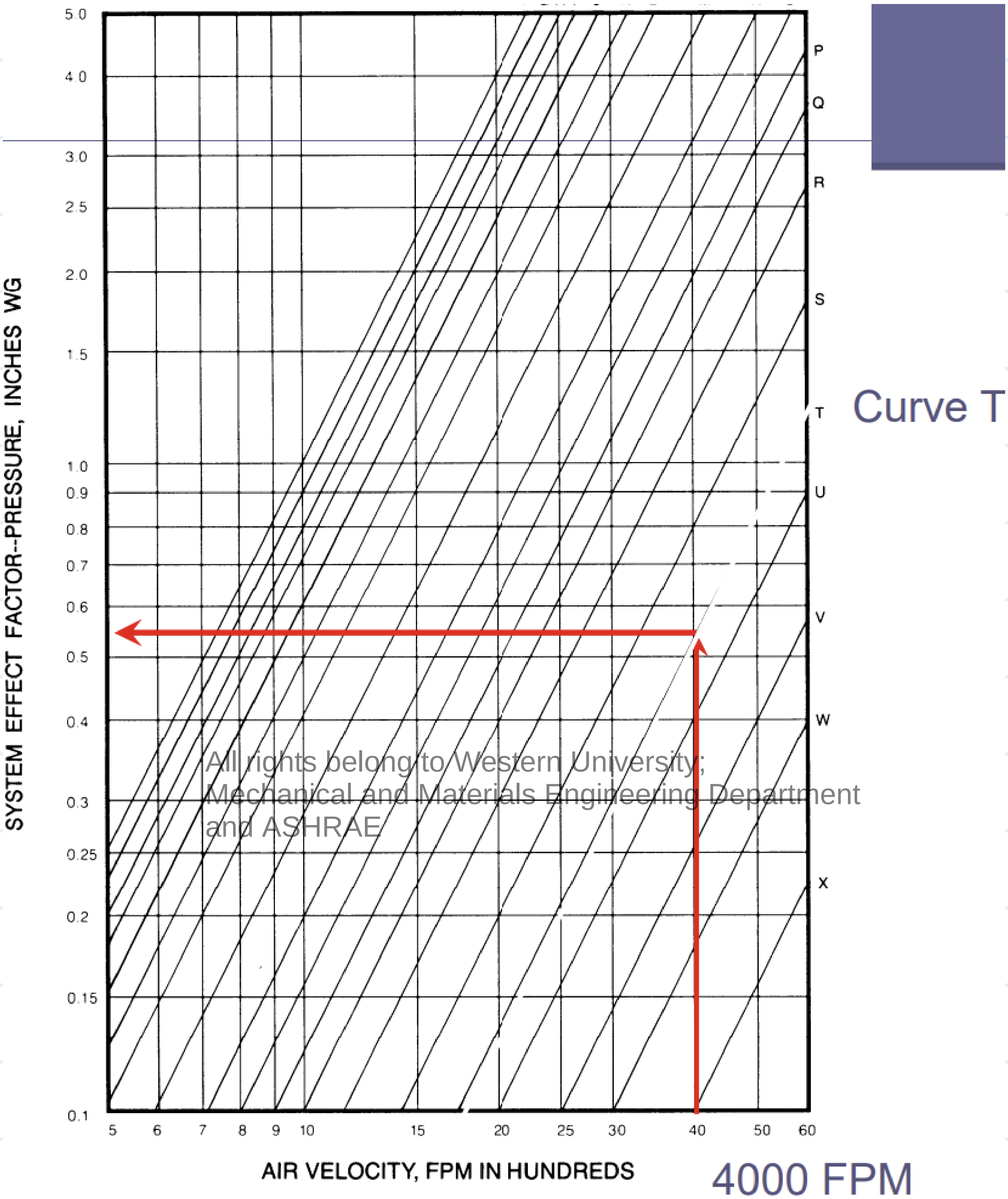
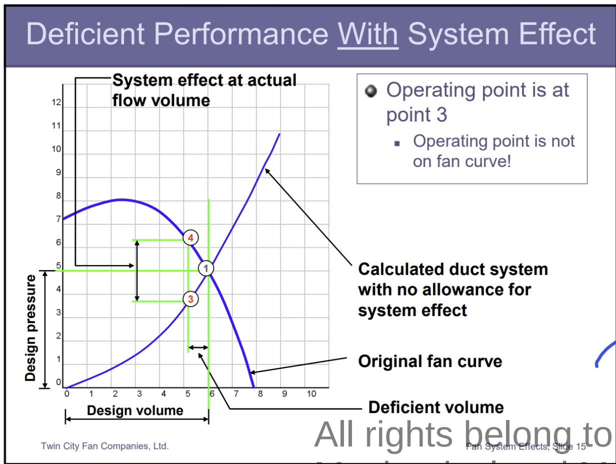
System Effect Curves for Outlet Duct Elbows - Axial Fans



	90# ELBOW	NO DUCT	12% EFFECTIVE DUCT	25% EFFECTIVE DUCT	50% EFFECTIVE DUCT	100% EFFECTIVE DUCT
Tubeaxial Fan	2 & 4 Pc	—	—	—	—	—
Vaneaxial Fan	2 Pc	U	U-V	V	W	—
Vaneaxial Fan	4 Pc	W	—	—	—	—


SYSTEM EFFECT FACTOR--PRESSURE, INCHES WG
5.0
4.0
3.0
2.5
2.0
1.5
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.25
0.2
0.15
0.1
P
Q
R
S
T
U
V
W
X
5 6 7 8 9 10 15 20 25 30 40 50 60
AIR VELOCITY, FPM IN HUNDREDS
Curve T
4000 FPM



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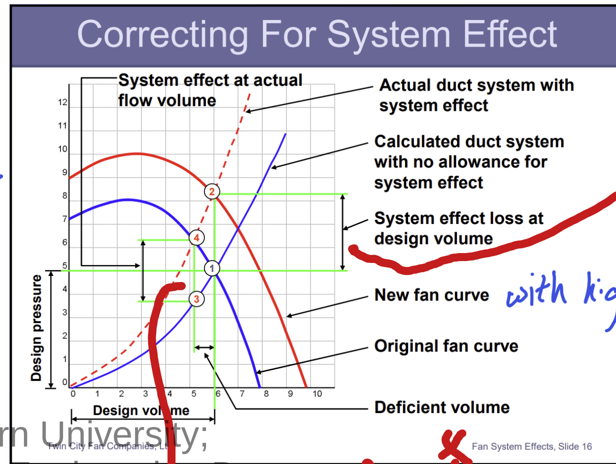
How to solve



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with higher RPM

mathematical loss of sys. effect

actual loss of sys. effect