CHAPTR 12 - overview

FANS

Characteristics

Performance ï fan laws

DUCTS

Losses by Equivalent Length

Losses by Loss Coefficient

OBJECTIVE

Sizing fans and ducts

Predicting performance of systems

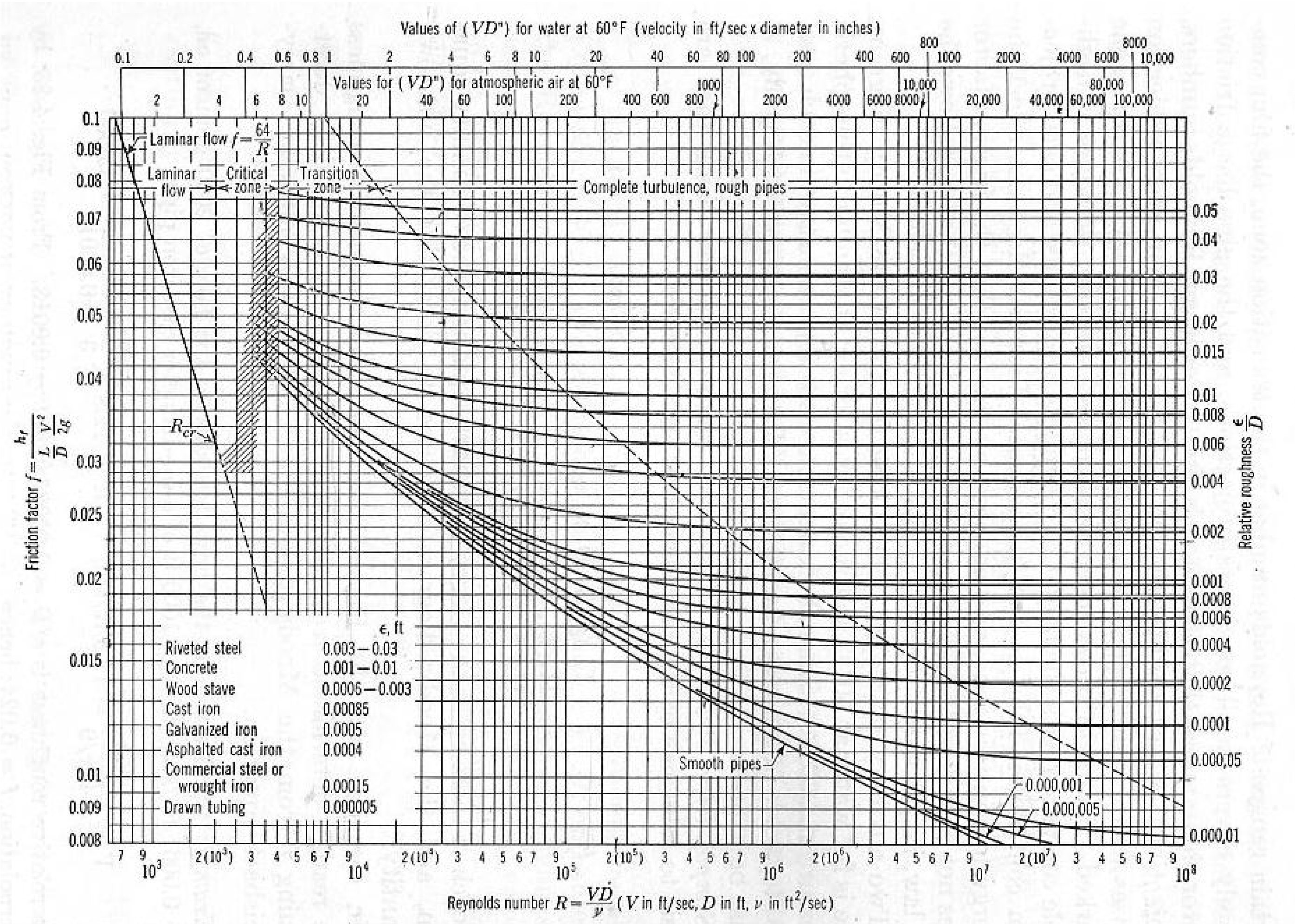


FIG. 4-33. Moody diagram. (This diagram, reproduced on a larger scale, is in an envelope attached to the back cover.)

Air

at 60F

2000ft/min,

1 ft diameter duct

$$N_{Re} = \frac{VDJ}{\varepsilon}$$

$$N_{Re} = \frac{2000\text{ft/min} \times 60 \times 1 \times .0763}{.043}$$

$$N_{Re} = 212{,}930.$$

$$h_f = f\,\frac{L}{D}\frac{V^2}{2g}$$

$$\varphi p = h_f C_c = C_c f\,\frac{L}{D2g}\frac{Q^2}{A^2}$$

$$\varphi p = \left[C_c f\,\frac{L}{D2gA^2}\right]Q^2$$

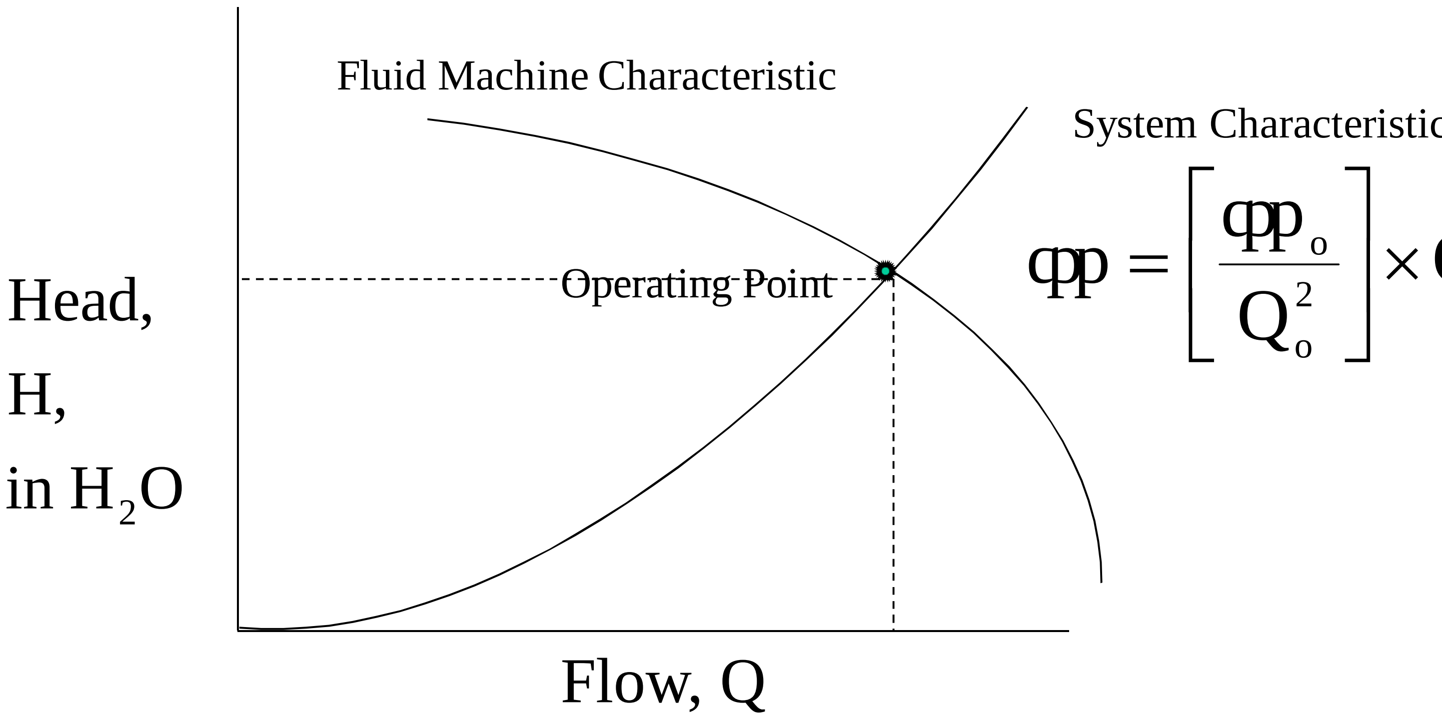
$$\varphi p = C \times Q^2$$

$$\varphi p_o = C \times Q_o^2$$

$$C = \frac{\varphi p_o}{Q_o^2}$$

$$\varphi p = \left[\frac{\varphi p_o}{Q_o^2}\right] \times Q^2$$

FLUID SYSTEM CHARACTERISTICS



$$\Delta P = \left[\frac{\Delta P_o}{Q_o^2}\right] \times Q^2$$

Energy (head) is put into the system fluid by the fluid machine in the form of velocity and pressure.

Energy (head) is removed from the system fluid by friction in the piping or duct work.

EQUATION OF MOTION

one dimensional, steady

$$\frac{dp}{dx} = \varepsilon \frac{d^2u}{dx^2}$$

dimensionless x, u, p,

$$U = \frac{u}{V_O} \quad u = V \times U$$

$$u\,du = V_O^2\,U\,dU, \quad d^2u = V_O\,d^2U$$

$$X = \frac{x}{L_O} \quad x = L_O \times X$$

$$dx = L_O\,dX, \quad dx^2 = L_O^2 dX^2$$

$$P = \frac{p}{P_O} \quad p = P_O \times P$$

$$dp = P_O\,dP$$

substituting for u, x, and p,

$$\left(\frac{P_o}{L_O}\right)\frac{dP}{dX} = \left(\frac{\varepsilon V_O}{L_O^2}\right)\frac{d^2U}{dX^2}$$

divide by $\frac{\text{Ɉ}\,V_O^2}{L_O}$,

$$\left(\frac{P_o}{\text{Ɉ}\,V_O^2}\right)\frac{dP}{dX} = \left(\frac{1}{\frac{\text{Ɉ}\,V_O L_O}{\varepsilon}}\right)\frac{d^2U}{dX^2}$$

$\frac{\text{Ɉ}\,V_O L_O}{\varepsilon}$ is the dimensionless parameter

Reynolds Number

$$\left(\frac{P_o}{\text{Ɉ}\,V_O^2}\right)\frac{dP}{dX} = \left(\frac{1}{N_{RE}}\right)\frac{d^2U}{dX^2}$$

FLUID MACHINE DIMENSIONLESS PARAMETERS

If the full set of equations for fluid machine,

Energy Balance - First Law
Mass balance - continuity equation
Equations of motion F=mass x acceleration

are non-dimensionalized 6 dimensionless parameters result.

If the machine variables are changes so that 5 of these dimensionless parameters remain constant, the 6th parameter will also remain constant.

In the operation of a pump or fan Mach Number, and Specific Heat Ratio remain constant and Reynolds Number changes very little.

If the Specific Speed and Specific Diameter of a fluid machine remain the same even though rotational speed, head and flow many change, the same efficiency will be achieved.

$$\text{Specific Speed,} \quad N_s = \frac{N \times Q^{.5}}{H^{.75}}$$

$$\text{Specific Diameter,} \quad D_S = \frac{D \times H^{.25}}{Q^{.5}}$$

$N = \text{rotational speed}$

$Q = \text{volume flow}$

$H = \text{head}$

$D = \text{diameter}$

$$\text{Mach Number,} \quad M = \frac{\text{Velocity}}{\text{Sonic Velocity}}$$

$$\text{Reynolds Number,} \quad N_{RE} = \frac{\varepsilon \times V \times L}{\text{Ɉ}}$$

$V = \text{velocity}$

$\varepsilon = \text{viscosity}$

$\text{Ɉ} = \text{density}$

$$\text{Specific Heat Ratio,} \quad k = \frac{c_p}{c_v}$$

$$\text{Efficiency} \quad \text{ɖ}_{compressor} = \frac{\text{Ideal Work}}{\text{Actual Work}}$$

$$\text{ɖ}_{turbine} = \frac{\text{Actual Work}}{\text{Ideal Work}}$$

PUMP SPECIFIC SPEED SPECIFIC DIAMETER DIAGRAM

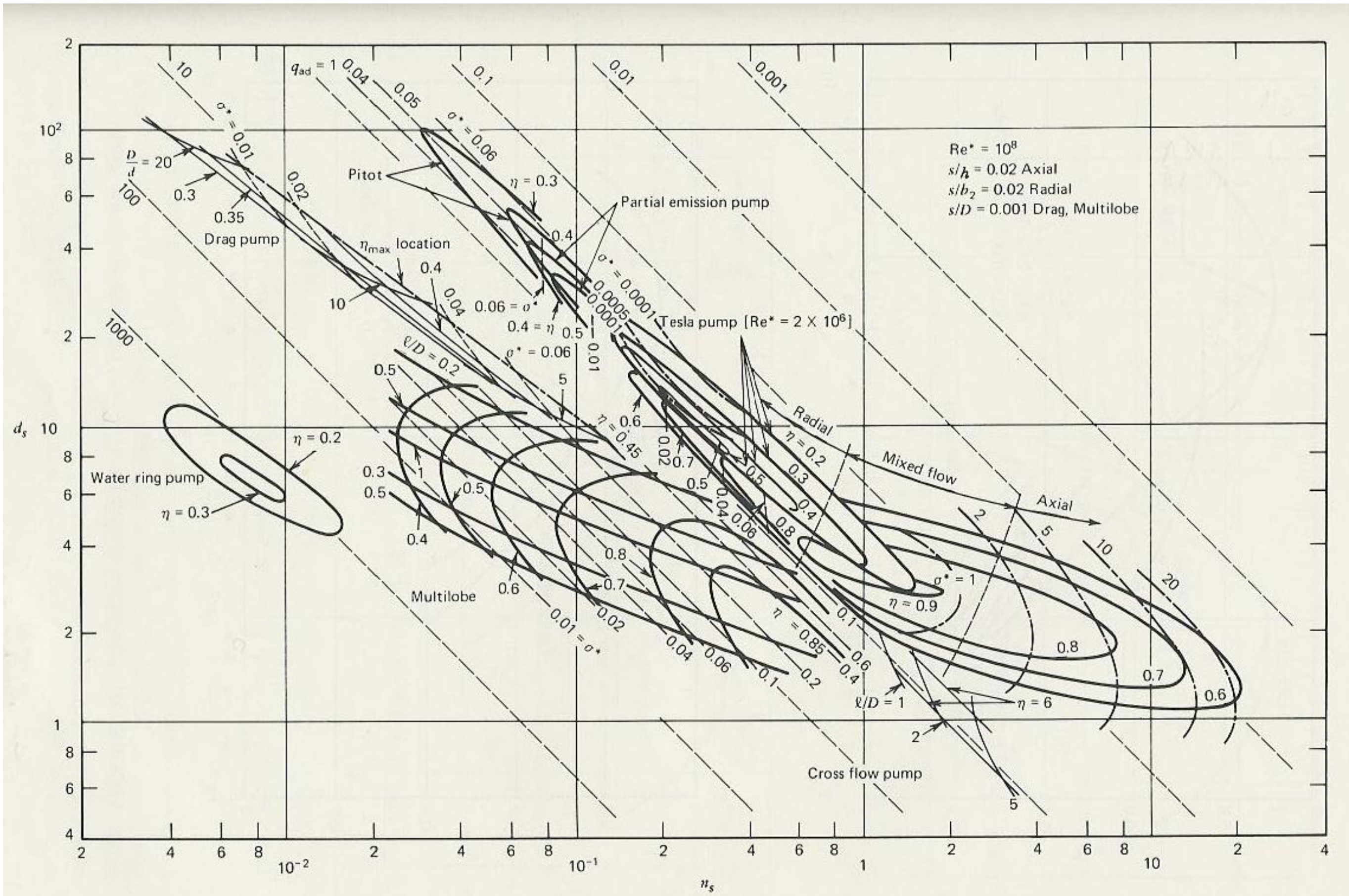


FIGURE 3.8 $n_s d_s$ diagram for single stage pumps.

1) Similar geometry+Constant Specific Speed and Specific Diameter = Same Efficiency

2) Each machine type has an optimum Specific Speed for maximum efficiency.

MACHINERY AFFINITY LAWS , (pump laws, fan laws)

$$\text{SPECIFIC DIAMETER, } D_s = \frac{DH^{.25}}{Q^{.5}} = \text{CONSTANT}$$

$$\frac{D_0 H_0^{.25}}{Q_0^{.5}} = \frac{D_1 H_1^{.25}}{Q_1^{.5}}$$

$$\left(\frac{Q_1}{Q_0}\right)^{.5} = \frac{D_`}{D_0}\left(\frac{H_1}{H_0}\right)^{.25}$$

$$\left(\frac{Q_1}{Q_0}\right)^{.5} = \frac{D_1}{D_0}\left(\left(\frac{N_1}{N_0}\right)^2\left(\frac{D_1}{D_0}\right)^2\right)^{.25}$$

$$\frac{Q_1}{Q_0} = \left(\frac{D_1}{D_0}\right)^3\left(\frac{N_1}{N_0}\right)$$

for the same impeller, $D_0 = D$

$$\frac{Q_1}{Q_0} = \left(\frac{N_1}{N_0}\right)$$

$$\text{POWER} = Q \times H$$

$$\frac{\text{Power}_1}{\text{Power}_2} = \left(\frac{Q_1}{Q_0}\right)\left(\frac{H_1}{H_0}\right) = \left(\frac{N_1}{N_0}\right)\left(\frac{N_1}{N_0}\right)^2$$

$$\frac{\text{Power}_1}{\text{Power}_2} = \left(\frac{N_1}{N_0}\right)^3$$

$$\text{SPECIFIC SPEED, } N_s = \frac{NQ^{.5}}{H^{.75}} = \text{CONSTANT}$$

$$\frac{N_0 Q_0^{.5}}{H_0^{.75}} = \frac{N_1 Q_1^{.5}}{H_1^{.75}}$$

$$\left(\frac{Q_1}{Q_0}\right)^{.5} = \frac{N_{0`}}{N_1}\left(\frac{H_1}{H_0}\right)^{.75}$$

$$\frac{D_1}{D_0}\left(\frac{H_1}{H_0}\right)^{.25} = \frac{N_0}{N_1}\left(\frac{H_1}{H_0}\right)^{.75}$$

$$\left(\frac{H_1}{H_0}\right)^{.5} = \left(\frac{N_1}{N_0}\right)\left(\frac{D_1}{D_0}\right)$$

$$\left(\frac{H_1}{H_0}\right) = \left(\frac{N_1}{N_0}\right)^2\left(\frac{D_1}{D_0}\right)^2$$

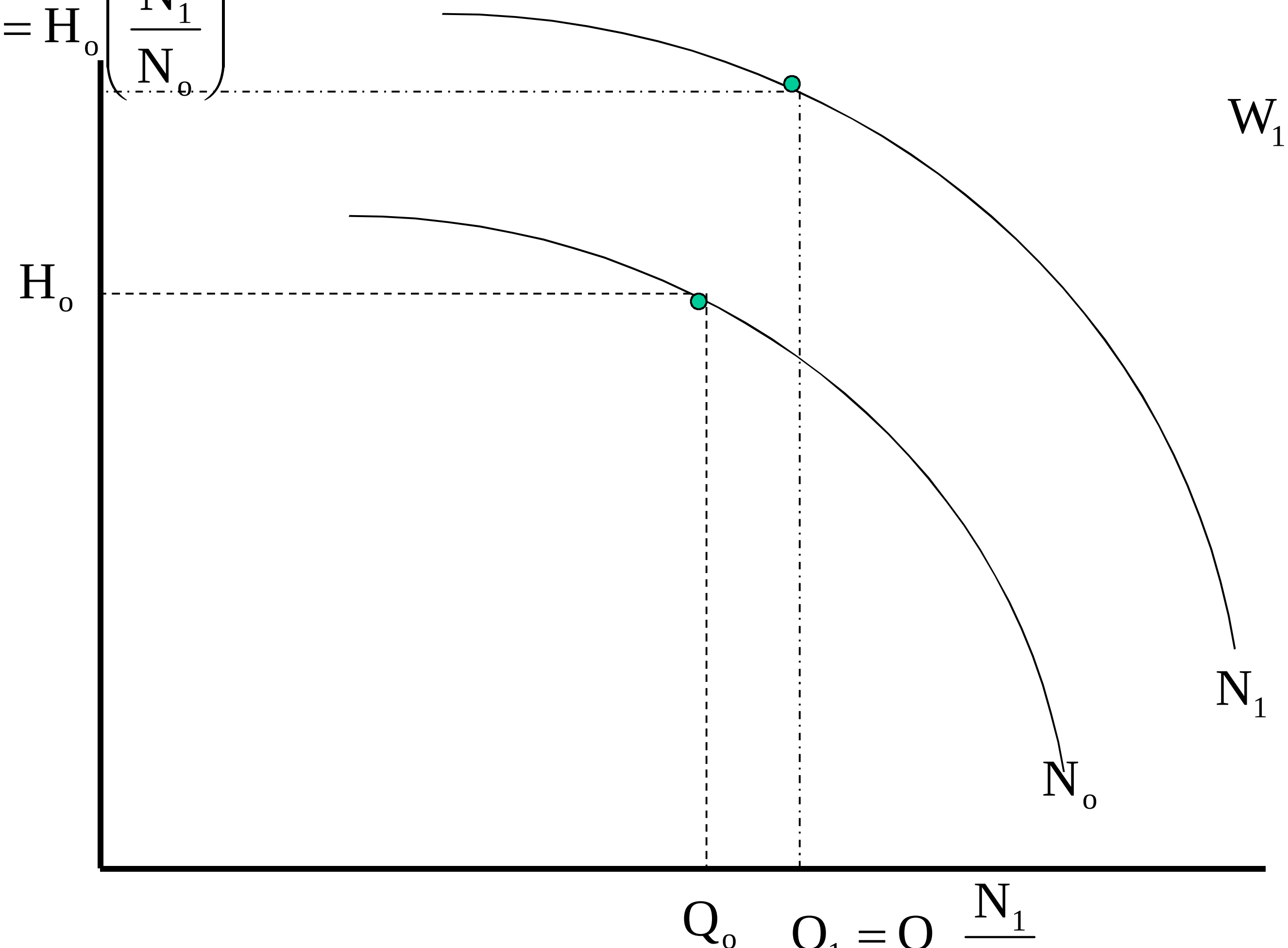
for the same impeller, $D_0 = D_1$

$$\frac{H_1}{H_0} = \left(\frac{N_1}{N_0}\right)^2$$

AFFINITY LAWS (pump laws, fan laws)

$$H_1 = H_o\left(\frac{N_1}{N_o}\right)^2$$

$$W_1 = W_o\left(\frac{N_1}{N_o}\right)^3$$



$$Q_1 = Q_o\,\frac{N_1}{N_o}$$

Fan Work

$$W = m\int v\,dp = m\frac{(p_{2\ total} - p_{1\ total})}{\rho} = \frac{lb}{sec}\,\frac{lb/ft^2}{lb/ft^3} = ft\ lb/sec$$

$$\rho_{\ standard\ air} = .0765\ lb/ft^3,\quad 59^{O}F,\ 29/.92\ \ inHg$$

$$W_{ideal} = \frac{Q\ \ cfm}{60\ \ sec/min}\ \frac{.0765\ \ lb/ft^3 \times \varphi P\ \ in\ \ H_2O \times \frac{62.4}{12}}{550}$$

$$W_{ideal} = \frac{Q\,cfm \times (p_{2\,total} - p_{1\,total})\ \ in\ H_2O}{6350} = HP, \qquad \eta = \frac{W_{ideal}}{W_{actual}} \qquad (12\text{-}4b)$$

Velocity Pressure

$$p_{total} = p_{static} + p_{velocity} = p_{static} + \frac{\rho V^2}{2g}, \qquad p_{velocity} = \frac{\rho V^2}{2g} = \frac{lb}{ft_3}\,\frac{ft^2}{sec^2} / \frac{ft}{sec^2}$$

$$p_{velocity} = \frac{.0765\ lb/ft^3\left(\frac{V\ ft/min}{60\ sec/min}\right)^2}{2 \times 32.2\ ft/sec^2}\ \frac{12\ in\ H_2O}{62.4 lb/ft^2}\ \ \text{for standard air}$$

$$p_{velocity} = \left(\frac{V\ ft/min}{4005}\right)^2 = in\ H_2O\ \ \text{page 405}$$

$Q = \varphi U + W$ First Law

$dQ = dU + dW$

$dQ_{\substack{\text{intenally}\\ \text{reversibe}}} = TdS$ Second Law

internally reversible - no irreversibilities within the boundaries of the system.

$dW = pdV$ Boundary Work

$du = c_v dT$ u property definition

subsitiuting,

$Tds = du + pdv$

$h = u + pv$ h property definition, h is an exact differential

$dh = du + dv + vdp$

subsitiuting,

$Tds = dh - pdv - vdp + pdv$

$Tds = dh - vdp$

for $Q = Tds = 0$

$dh = vdp$

Example: water pumped from 10 psia to 30 psia

$$w = h_2 - h_1 = v(p_2 - p_1)$$

$$w = \frac{(30\text{psia} - 15\text{psia}) \times 144\text{psf/psi}}{62.4\text{lb/ft}^3}$$

$$w = \frac{1509\,\frac{\text{lb}_f}{\text{ft}^2}}{62.4\,\frac{\text{lb}_m}{\text{ft}^2}\,\frac{1}{\text{ft}}} = 69.1\,\frac{\text{ft lb}_f}{\text{lb}_m},\ (\text{ft of fluid})$$

$$w = 69.1\,\frac{\text{ft lb}_f}{\text{lb}_m} \times \frac{1\,\text{BTU}}{778\,\text{ft lb}_f} = 69.1\text{BTU/lb}_m$$

$$w = .0888\,\text{BTU/lb}$$

Example: water pumped from 100 kPa to 300 kPa

$$w = v(p_2 - p_1)$$

$$w = .0010432 \times (300 - 200)$$

$$w = .02086\ \frac{m_3}{kg}\,kPa, \quad kJ/kg$$

12-1. A centrifugal fan is delivering 1700 cfm (0.8 m^3/s) of air at a total pressure differential (across the fan) of 1.4 in. wg (350 Pa). The fan has an outlet area of 0.71 ft^2 (0.07 m^2) and requires 0.7 hp (0.52 kW) shaft input. Compute (a) the total power, (b) the static efficiency, (c) the total efficiency, and (d) the fan static pressure.

12-1

1700 cfm @ 1.4 in H_2O total pressure

.71 ft^2 exit area, .7 shaft HP

a) $W_{total} = \frac{Q(p_2 - p_1)}{6350} = \frac{1700 \times 1.4}{6350} = .375 \text{ HP}$

b)

$V = \frac{Q}{A} = \frac{1700}{.71} = 2394 \text{ ft/min}$

$p_v = \left(\frac{V}{4005}\right)^2 = \left(\frac{2394}{4005}\right)^2 = .357 \text{ in } H_2O$

$p_{static} = 1.4 - .357 = 1.043 \text{ in } H_2O$

$W_{static} = \frac{Q(p_{2static} - p_{1statuic})}{6350} = \frac{1700 \text{ ft}^3/\text{min} \times 1.043}{6350}$

$W_{static} = .279 \text{ HP}$

$\eta_{static} = \frac{W_{static}}{W_{shaft}} = \frac{.279}{.7} = 39.9\%$

c) $\eta_{total} = \frac{W_{total}}{W_{shaft}} = \frac{.375}{.7} = 53.7\%$

d) $p_{static} = 1.043 \text{ in } H_2O$

12-9. A small system requires 0.88 in. wg total pressure at a flow rate of 1420 cfm. Select a suitable fan using the data of Table 12-2*a*. (a) Sketch the system and fan characteristics, showing the operating point. (b) What are the fan speed and power?

Table 12-2*a* Pressure–Capacity Table for a Forward-Curved Blade Fan

Volume Flow Rate, cfm	Outlet Velocity, ft/min	$\frac{1}{2}$ in. wg[a]		$\frac{5}{8}$ in. wg		$\frac{3}{4}$ in. wg		1 in. wg		$1\frac{1}{4}$ in. wg		$1\frac{1}{2}$ in. wg	
		rpm	bhp[b]	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
851	1200	848	0.13	933	0.16	1018	0.19	—	—	—	—	—	—
922	1300	866	0.15	945	0.18	1019	0.21	—	—	—	—	—	—
993	1400	884	0.17	957	0.20	1030	0.23	1175	0.30	—	—	—	—
1064	1500	901	0.19	973	0.22	1039	0.26	1182	0.32	—	—	—	—
1134	1600	926	0.22	997	0.24	1057	0.29	1190	0.35	1320	0.43	—	—
1205	1700	954	0.25	1020	0.27	1078	0.31	1200	0.38	1325	0.46	1436	0.55
1276	1800	983	0.28	1044	0.31	1100	0.34	1210	0.42	1330	0.50	1440	0.59
1347	1900	1011	0.31	1068	0.35	1126	0.38	1230	0.46	1341	0.54	1447	0.63
1418	2000	1039	0.35	1092	0.39	1152	0.42	1250	0.50	1352	0.59	1458	0.66
1489	2100	1068	0.39	1115	0.43	1178	0.47	1275	0.54	1370	0.62	1470	0.72
1560	2200	1096	0.44	1147	0.47	1204	0.51	1300	0.59	1390	0.67	1482	0.77
1631	2300	1124	0.48	1179	0.52	1230	0.56	1325	0.64	1420	0.73	1500	0.83
1702	2400	1152	0.53	1210	0.58	1256	0.62	1350	0.70	1448	0.78	1525	0.88

[a] Static pressure.

[b] Shaft power in horsepower.

Note. Data are for a 9 in. wheel diameter and an outlet of 0.71 ft^2.

12-9

a) 1420 cfm required at .88 in Toal Pressure

b) In Table 12 – 2a at 1418 cfm, V = 2000 ft/min

$$p_{velocity} = \left(\frac{V}{4005}\right)^2 = \left(\frac{2000}{4005}\right)^2 = .25 \text{ in } H_2O$$

$$p_o = p_s + p_v = .88 \text{ in } H_2O$$

$$p_s = .88 - .25 = .63 \text{ in } H_2O$$

In Table 12 – 2a in .625 in H_2O column,

N = 1092 rpm, HP = .39

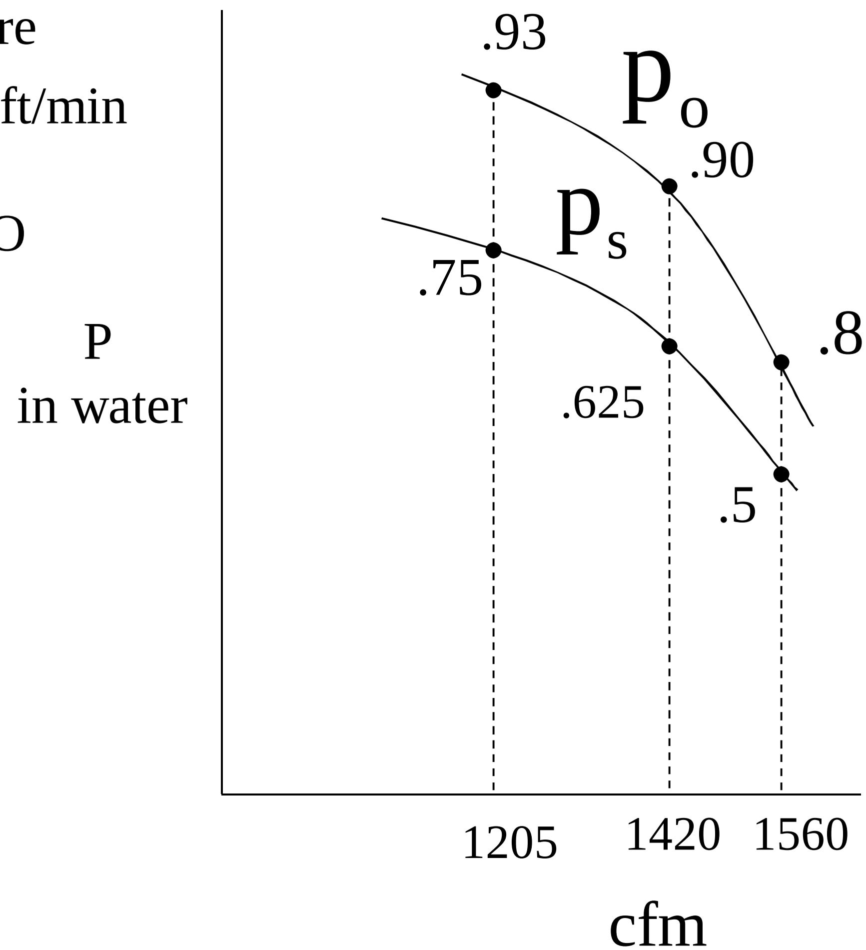
Total Pressure Characteristic

$$p_{v1560} = \left(\frac{2200}{4005}\right)^2 = .30 \text{ in } H_2O$$

$$p_{01560} = .5 + .3 = .8 \text{ in } H_2O$$

$$p_{v1205} = \left(\frac{1700}{4005}\right)^2 = .18 \text{ in } H_2O$$

$$p_{01205} = .75 + .18 = .93 \text{ in } H_2O$$



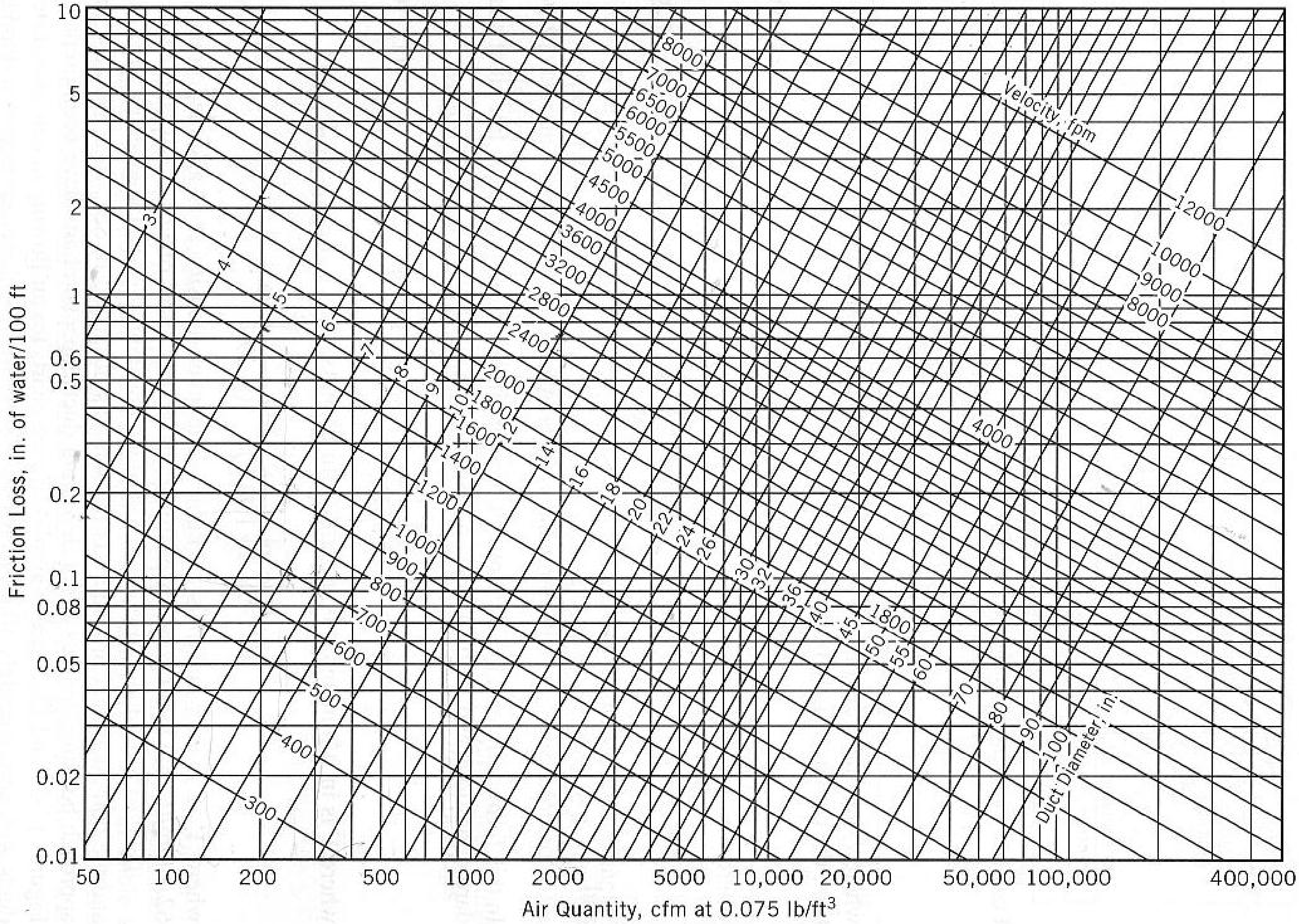


Figure 12-21 Pressure loss due to friction for galvanized steel ducts, IP units. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume IP*, 1997.)

EQUIVALENT LENGTH METHOD

$$\varphi h = (\text{length of duct} + \text{fitting equivalent length of duct}) \times \frac{\varphi p}{L_e}$$

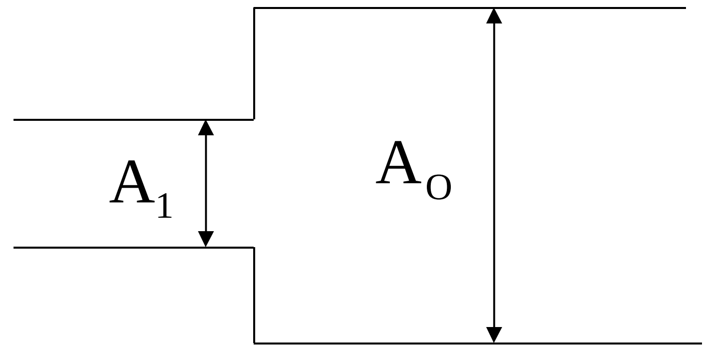
$$\varphi h = (\text{problem given} + \text{Table } 12-14) \times \text{Figure } 12-21 \text{ or } 22$$

LOSS COEFFICIENT METHOD

$$\varphi h = \left(\text{duct length} \times \frac{\varphi p}{L_e} \right) + C_O \text{ of fittings} \times \left(\frac{V}{4005} \right)^2$$

$$(\text{problem given} \times \text{Figure } 12-21 \text{ or } 22) + \text{Table } 12-8 \text{ to } 12-12 \times \left(\frac{V}{4005} \right)^2$$

12-34. What is the lost pressure for an 18 × 18 in. (46 × 46 cm) duct discharging into a large plenum? The flow rate is 4500 cfm (2.1 m^3/s), and the duct expansion ratio A_0/A_1 is 4.0. Figure 12-10B applies to this situation. (a) Assume an abrupt entrance; (b) assume a 20 degree transition exists at the entrance to the plenum.



<u>12-34</u>

a) $\varphi p = C_o \times p_{vo}$

C_o is based on A_o and V_O

$$C_o = 10.56 \ @ \ \text{đ} = 180^o, \frac{A_o}{A_1} = 4, \quad \text{Table } 12-9\text{B (rectangular duct p 427)}$$

$$V_o = \frac{\text{cfm}}{A_o} = \frac{4500\,\text{cfm}}{4 \times A_1} = \frac{4500\,\text{cfm}}{4 \times 18^2/144} = 500\,\text{ft/min}$$

$$\varphi p = C_o \times \left(\frac{V_o}{4005}\right)^2 = 10.56 \times \left(\frac{500}{4005}\right)^2 = .165\,\text{in } H_2O$$

b) $C_o = 3.52 \ @ \ \text{đ} = 20^o, \ \frac{A_o}{A_1} = 4, \quad \text{Table } 12-10\text{B (rectangular duct)}$

$$\varphi p = C_o \times \left(\frac{V_o}{4005}\right)^2 = 3.52 \times \left(\frac{500}{4005}\right)^2 = .055\,\text{in}H_2O$$

Table 12-7 Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity—Dimensions in Inches, Feet, or Meters

Side *a* of Rectangular Duct	Diameter D_e of Circular Duct																
	b = 6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	22	24
6	6.6																
7	7.1	7.7															
8	7.5	8.2	8.8														
9	8.0	8.6	9.3	9.9													
10	8.4	9.1	9.8	10.4	10.9												
11	8.8	9.5	10.2	10.8	11.4	12.0											
12	9.1	9.9	10.7	11.3	11.9	12.5	13.1										
13	9.5	10.3	11.1	11.8	12.4	13.0	13.6	14.2									
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3								
15	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4							
16	10.4	11.4	12.2	13.0	13.7	14.4	15.1	15.7	16.3	16.9	17.5						
17	10.7	11.7	12.5	13.4	14.1	14.9	15.5	16.1	16.8	17.4	18.0	18.6					
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.6	17.3	17.9	18.5	19.1	19.7				
19	11.2	12.2	13.2	14.1	14.9	15.6	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8			
20	11.5	12.5	13.5	14.4	15.2	15.9	16.8	17.5	18.2	18.8	19.5	20.1	20.7	21.3	21.9		
22	12.0	13.1	14.1	15.0	15.9	16.7	17.6	18.3	19.1	19.7	20.4	21.0	21.7	22.3	22.9	24.1	
24	12.4	13.6	14.6	15.6	16.6	17.5	18.3	19.1	19.8	20.6	21.3	21.9	22.6	23.2	23.9	25.1	26.2
26	12.8	14.1	15.2	16.2	17.2	18.1	19.0	19.8	20.6	21.4	22.1	22.8	23.5	24.1	24.8	26.1	27.2
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.6	24.4	25.0	25.7	27.1	28.2
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9	26.7	28.0	29.3
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.6	24.4	25.2	26.0	26.7	27.5	28.9	30.1
34	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5	28.3	20.7	31.0
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	23.0	23.9	24.8	25.8	26.6	27.4	28.3	29.0	30.5	32.0
38	15.0	16.4	17.8	19.0	20.3	21.4	22.5	23.5	24.5	25.4	26.4	27.3	28.1	29.0	29.8	31.4	32.8
40	15.3	16.8	18.2	19.4	20.7	21.9	23.0	24.0	25.1	26.0	27.0	27.9	28.8	29.7	30.5	32.1	33.6

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Table 12-14 Approximate Equivalent Lengths for Selected Fittings in Circular Ducts[a]

Fitting	L_e/D	Equivalent Length, ft (m) at Diameter, in. (cm) 6 (15)	8 (20)	10 (25)	12 (30)
Elbows					
Pleated, 90 deg	15	8 (2.4)	10 (3.1)	13 (4.0)	15 (4.6)
Pleated, 45 deg	9	5 (1.5)	6 (1.8)	8 (2.4)	9 (2.7)
Mitered, 90 deg	60	30 (9.1)	40 (12.2)	50 (15.2)	60 (18.3)
Mitered with vanes	10	5 (1.5)	7 (2.1)	8 (2.4)	10 (3.1)
Transitions					
Converging, 20 deg	4	2 (0.6)	3 (0.9)	3 (0.9)	4 (1.2)
Diverging, 120 deg	40	20 (6.1)	27 (8.2)	33 (10.1)	40 (12.2)
Abrupt expansion	60	30 (9.1)	40 (12.2)	50 (15.2)	60 (18.3)
Round to rectangular boot, 90 deg	50	25 (7.6)	33 (10.1)	40 (12.2)	50 (15.2)
Round to rectangular boot, straight	10	5 (1.5)	7 (2.1)	8 (2.4)	10 (3.1)
Entrances					
Abrupt, 90 deg	30	15 (4.6)	20 (6.1)	25 (7.6)	30 (9.1)
Bellmouth	12	6 (1.8)	8 (2.4)	10 (3.1)	12 (3.7)
Branch Fittings, Diverging					
Wye, 45 deg, branch	20	10 (3.1)	13 (4.0)	17 (5.2)	20 (6.1)
Wye, 45 deg, through	8	4 (1.2)	5 (1.5)	7 (2.1)	8 (2.4)
Tee, branch	40	20 (6.1)	27 (8.2)	33 (10.1)	40 (12.2)
Tee, through	8	4 (1.2)	5 (1.5)	7 (2.1)	8 (2.4)
Branch Fittings, Converging[b]					
Wye, 45 deg, branch	20	10 (3.1)	13 (4.0)	17 (5.2)	20 (6.1)
Wye, 45 deg, through	10	5 (1.5)	7 (2.1)	8 (2.4)	10 (3.1)
Tee, branch	40	20 (6.1)	27 (8.2)	33 (10.1)	40 (12.2)
Tee, through	12	6 (1.8)	8 (2.4)	10 (3.1)	12 (3.7)

[a]Equivalent lengths are approximate and based on Tables 12-7 through 12-12 using typical operating conditions with velocity less than about 1200 ft/min or 6 m/s.

[b]It is difficult to assign one value of L/D to this type fitting. Consult Table 12-12.

12-37. Compute the loss in total pressure for each run of the duct system shown in Fig. 12-35. The ducts are of round cross section. Turns and fittings are as shown. Use the loss coefficient and the equivalent length approaches (Table 12-15), and compare the answers. Use English units.

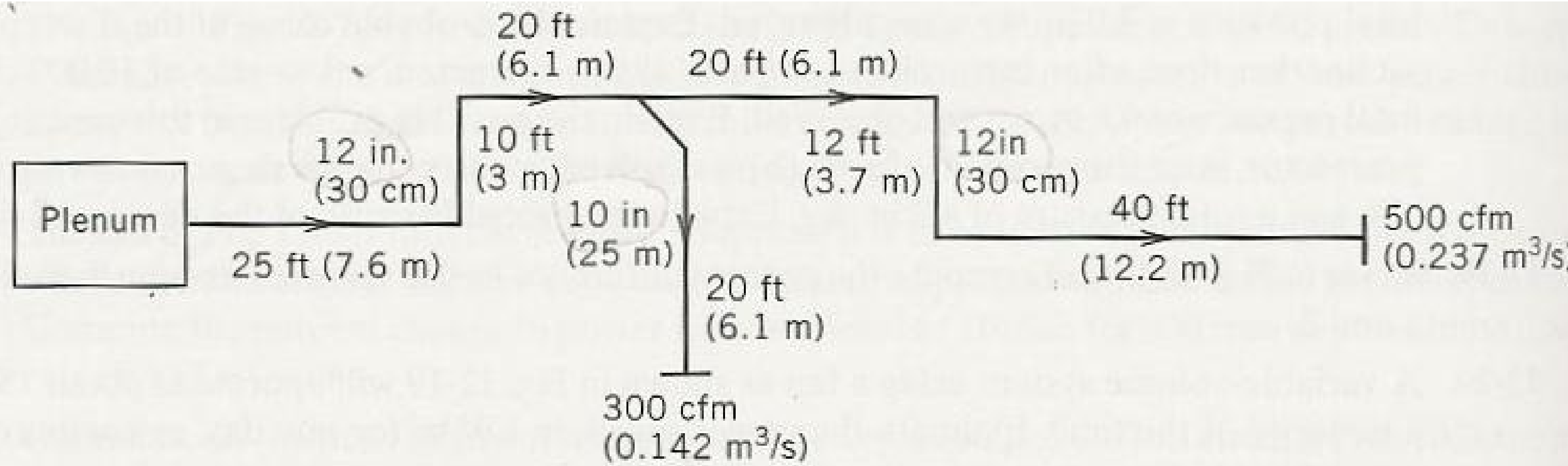
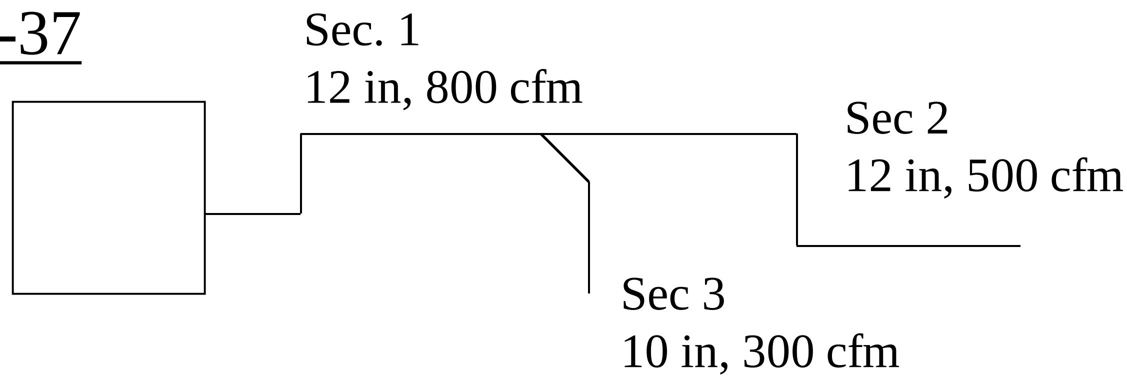


Figure 12-35 Schematic for Problem 12-37.

12-37



Equivalent Length Method

				Fittings						
Sec.	D	L	Q	Ent.	El.	Bch.	L_e	φp/L	V	φp
1	12	55	800	30	2×15	-	115	.135	1030	.155
2	12	72	500	-	2×15	-	102	.055	630	.056
3	10	20	300	-	8	17	45	.053	550	.024

Table 12-14, p 435, @ D and fitting type

Figure 12-21, p 420, @ Q and D

12-37

Loss Coeficient Method

Sec.	D	L	ΔP/L	Δh	Q	Q/Q	A/A	Ent.	El.	Bch.	V	φh
1	12	55	.135	.074	800	1.0	1.	.5	2×.26	-	1030	.141
2	12	72	.055	.040	500	.63	1.	-	2×.26	.16	630	.067
3	10	20	.053	.011	300	.38	.7	-	.17	2.0	550	.052

Figure 12 - 21, @ D and Q

Table 12 - 10

Table 12 - 8

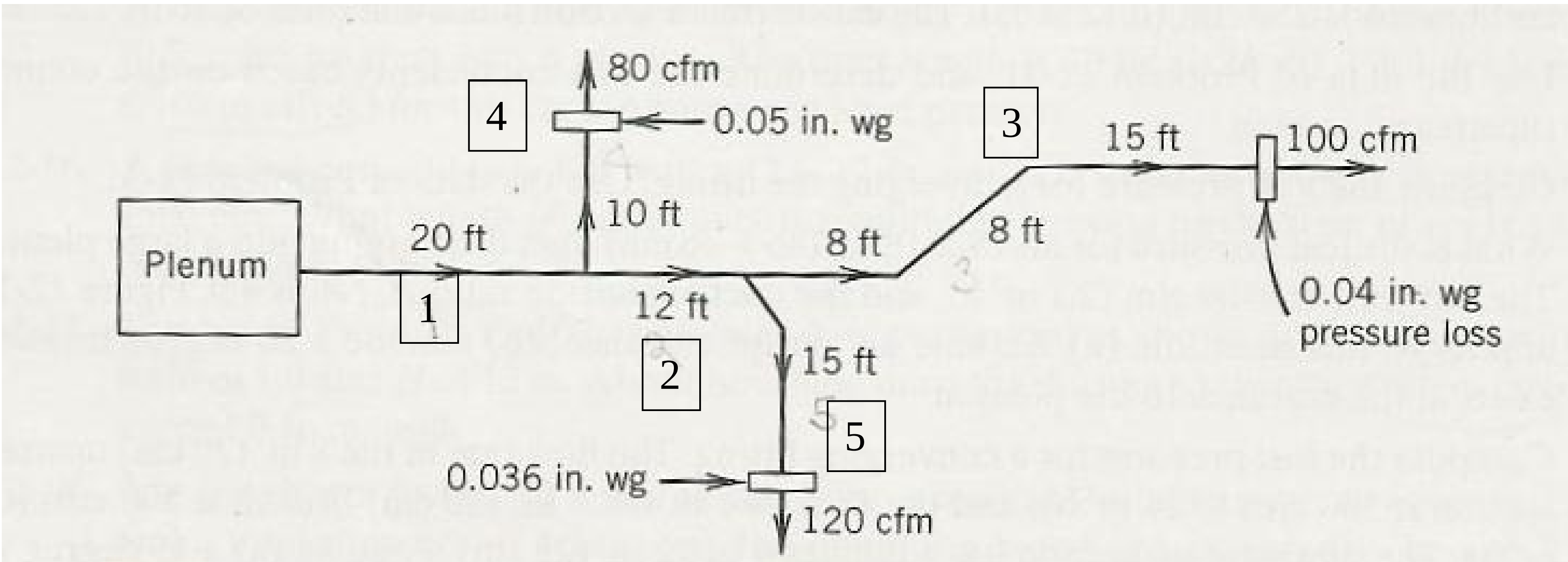
$$\varphi h = L \times \left(\frac{\varphi p}{L} \right) + C_o \left(\frac{V}{4005} \right)^2$$

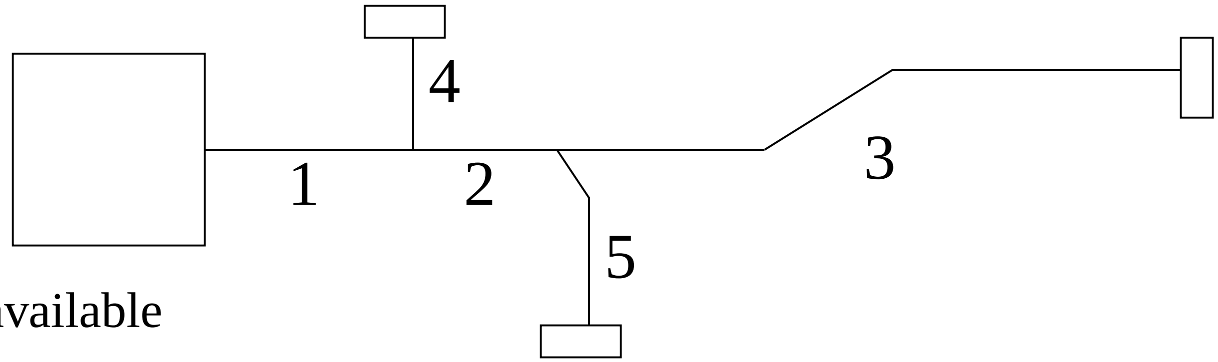
$$\varphi h_1 = 55 \times .135 + (.5 + 2 \times .26) \left(\frac{1030}{4005} \right)^2 = .141 \text{ in } H_2O$$

Comparison

Sec.	L eq	Co
1	.155	.141
2	.056	.067
3	.024	.052

12-40. The duct system shown in Fig. 12-36 is one branch of a complete air-distribution system. The system is a perimeter type located below the floor. The diffuser boots are shown in Table 12-15. Size the various sections of the system, using the equal-friction method and round pipe. A total pressure of 0.13 in. wg is available at the plenum. Compute the actual loss in total pressure for each run, assuming that the proper amount of air is flowing.





$.13 \text{ in } H_2O$ available

longest run $1-2-3=61 \text{ ft}$

assume $\Delta p/L_e = .10 \text{ in } H_2O/100\text{ft}$ and iterate until $\varphi P \le .13$

Equal Friction Method, Equivalent Length Losses

				Actual	Fitting Losses					
Sec	Q	D	$\varphi P/L_e$	L	Ent	T	El	Boot	L_e	φP
1	300	9	.080	20	25	-	-	-	70.	.036
2	220	8	.090	10	-	8	-	-	18.	.0162
3	100	6	.088	31	-	6	10	45	92.	.081
4	80	5	.014	10	-	10	-	36	60.7	.085
5	120	6	.125	15	-	20	5	29	69.	.075

D: Figure 12-21; T, El, Boot: Table 12-14

boot $L_e = \varphi P/(\varphi P/L)$

$\varphi p_{1\to4} = .036 + .085$

$\varphi p_{1\to4} = .121 \text{ in} H_2O$

$\varphi p_{1\to2\to5} = .036 + .0162 + .075$

$\varphi p_{1\to2\to5} = .127 \text{ in} H_2O$

$\varphi p_{1\to2\to3} = .036 + .0162 + .081$

$\varphi p_{1\to2\to3} = .133 \text{ in} H_2O$

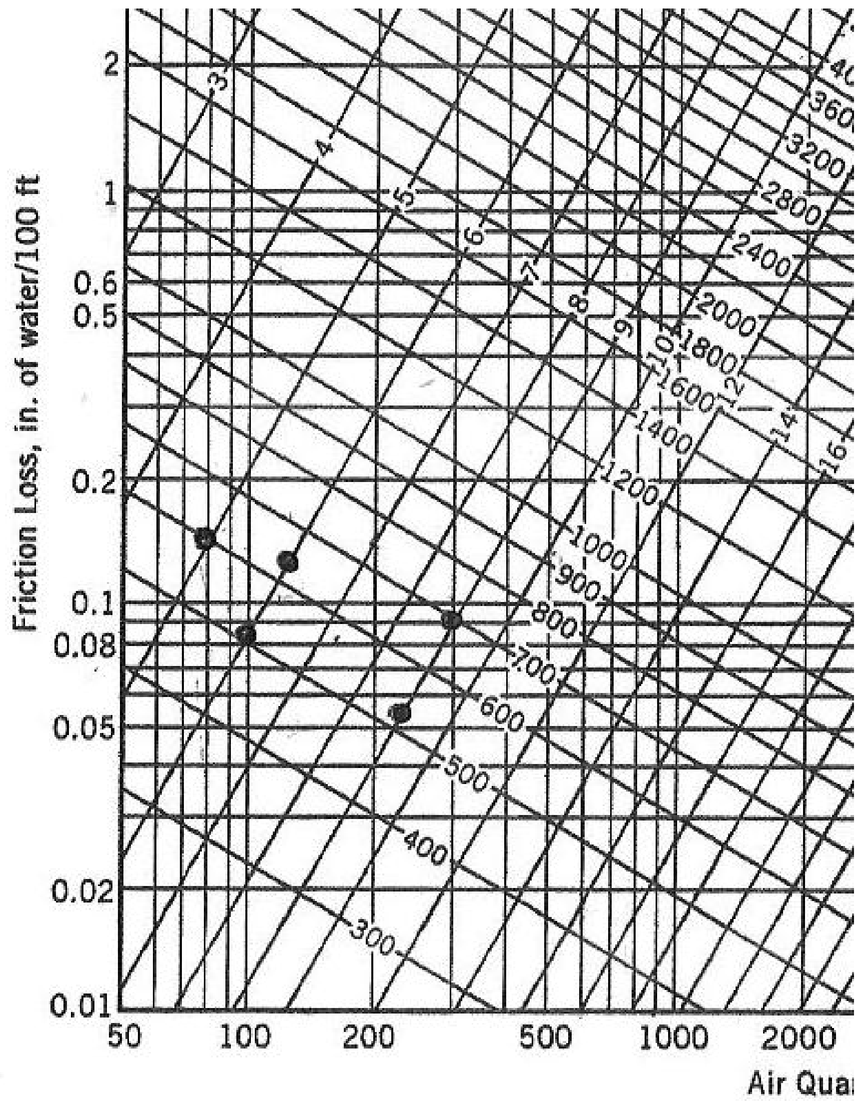
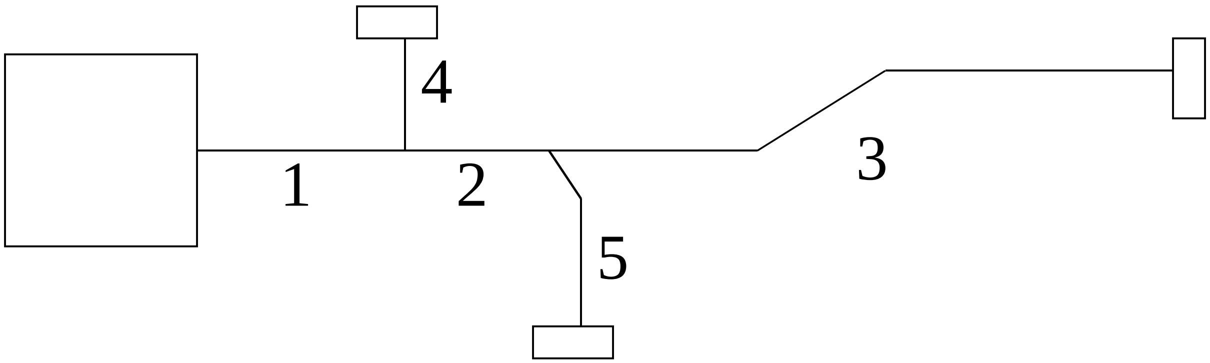


Figure 12-21 Pressure loss due to friction for galvanized steel
Fundamentals Volume IP, 1997.)

Design
φP/100ft



boot $L_{eq} = 100 \times .04/.088$

boot $L_{eq} = 45.45 \text{ in } H_2O$

boot $L_e = \varphi P/(\varphi P/L)$

$\varphi p_{1\to4} = .0405 + .085$

$\varphi p_{1\to4} = .1255 \text{ in} H_2O$

$\varphi p_{1\to2\to5} = .0405 + .0162 + .075$

$\varphi p_{1\to2\to5} = .1317 \text{ in} H_2O$

$\varphi p_{1\to2\to3} = .0405 + .0162 + .081$

$\varphi p_{1\to2\to3} = .1377 \text{ in} H_2O$

$.13 \text{ in } H_2O$ available

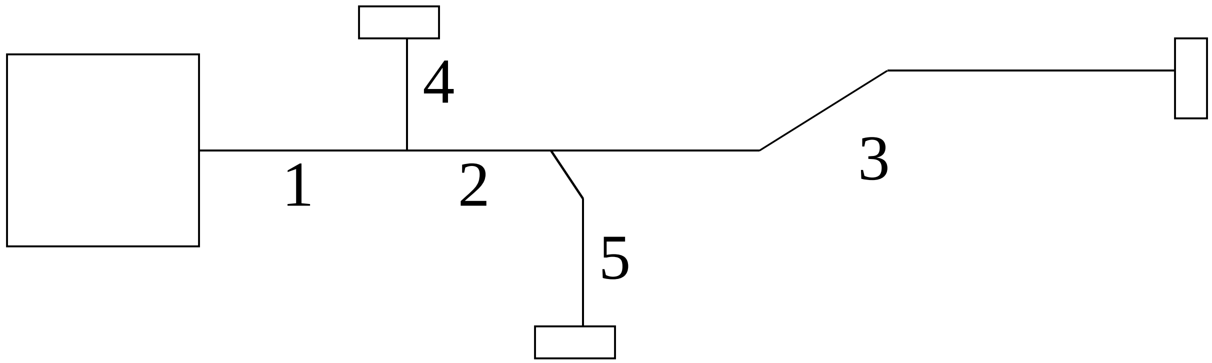
longest run $1-2-3 = 61$ ft

assume $\Delta p/L_e = .10 \text{ in } H_2O/100\text{ft}$ and iterate until $\varphi P \le .13$

Equal Friction Method, Equivalent Length Losses

			Actual		Fitting Losses									
Sec	Q	D	$\varphi P/L_e$	L	Ent	T_{thru}	T_{branch}	Y_{thru}	Y_{branch}	El_{90}	EL_{45}	Boot	L_e	φP
1	300	9	.090	20	25	-	-	-					45.	.0405
2	220	8	.090	10	-	8	-	-					18.	.0162
3	100	6	.088	31	-			5	10			45	92.	.081
4	80	5	.014	10	-	10	-					36	60.7	.085
5	120	6	.125	15	-				20		5	29	69.	.075

Figure 12-21 Table 12-14



$L_{eq} = 100 \times (\varphi\varphi P/_{eq})/\varphi_{boot}$

$\text{boot } L_{eq} = 100 \times .04/.088$

$\text{boot } L_{eq} = 45.45 \text{ in } H_2O$

$\varphi p_{1\to 4} = .0405 + .085$

$\varphi p_{1\to 4} = .1255 \text{ in} H_2O$

$\varphi p_{1\to 2\to 5} = .0405 + .0099 + .075$

$\varphi p_{1\to 2\to 5} = .1254 \text{ in} H_2O$

$\varphi p_{1\to 2\to 3} = .0405 + .0099 + .081$

$\varphi p_{1\to 2\to 3} = .1314 \text{ in} H_2O$

$.13 \text{ in } H_2O$ available

longest run $1-2-3 = 61 \text{ ft}$

assume $\Delta p/L_e = .10 \text{ in } H_2O/100\text{ft}$ and iterate until $\varphi P \leq .13$

Equal Friction Method, Equivalent Length Losses

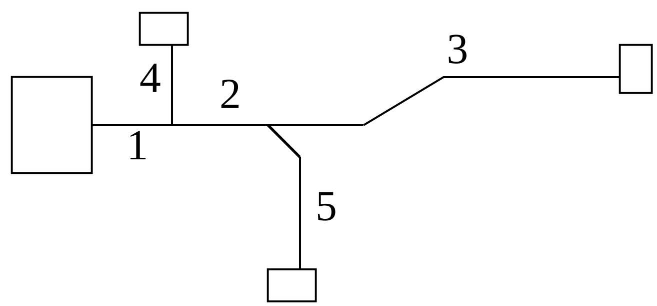
			Actual		Fitting Losses									
Sec	Q	D	$\varphi P/L_e$	L	Ent	T_{thru}	T_{branch}	Y_{thru}	Y_{branch}	El_{90}	EL_{45}	Boot	L_e	φP
1	300	9	.090	20	25	-	-	-	-	-	-	-	45.	.0405
2	220	9	.055	10	-	8	-	-	-	-	-	-	18.	.0099
3	100	6	.088	31	-	-	-	-	-	-	10	45.	92.	.081
4	80	5	.014	10	-	10	-	-	-	-	-	36.	60.7	.085
5	120	6	.125	15	-	-	-	-	-	20.	5.	29.	69.	.075

Figure 12-21   Table 12-14

Balanced Capacity Method, Equivalent Length Losses

Sec	Q	D	Actual φP/L_e	L	Fitting Losses Ent	T_{thru}	T_{branch}	Y_{thru}	Y_{branch}	El_{90}	EL_{45}	Boot	L_e	φP	$\sum$XP
1	300	9	.090	20	25	-	-	-	-	-	-	-	45.	.0405	
2	220	9	.055	10	-	8	-	-	-	-	-	-	18.	.0099	
3	100	6	.088	31	-	-	-	-	-	-	10	45.	92.	.081	.1314

Sec	Q	allowable φP/100	D	V
4	80	.1314		
		-.0405 Sec 1		
		$.02736 \times 100/60.7$		
		.04507 in H_2O/100 ft	6	500
5	120	.1314		
		-.045 Sec 1		
		-.0099 Sec 2		
		$.0765 \times 100/69$		
		.1109in H_2O/100ft	5	630



operating points.

12-47. Refer to Fig. 12-42, and construct the energy grade line (total pressure versus length) for the system shown. The change in total pressure in in. wg is shown for each part of the system. What total pressure must each fan produce?

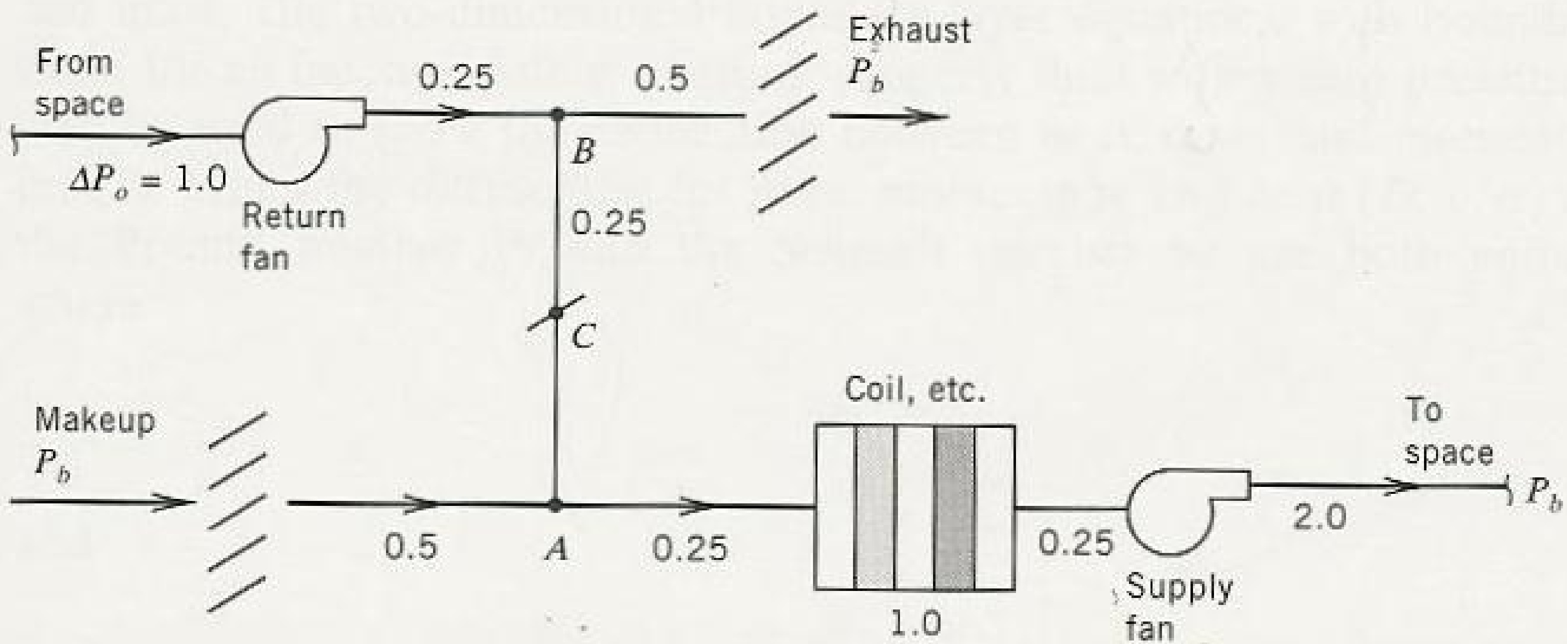
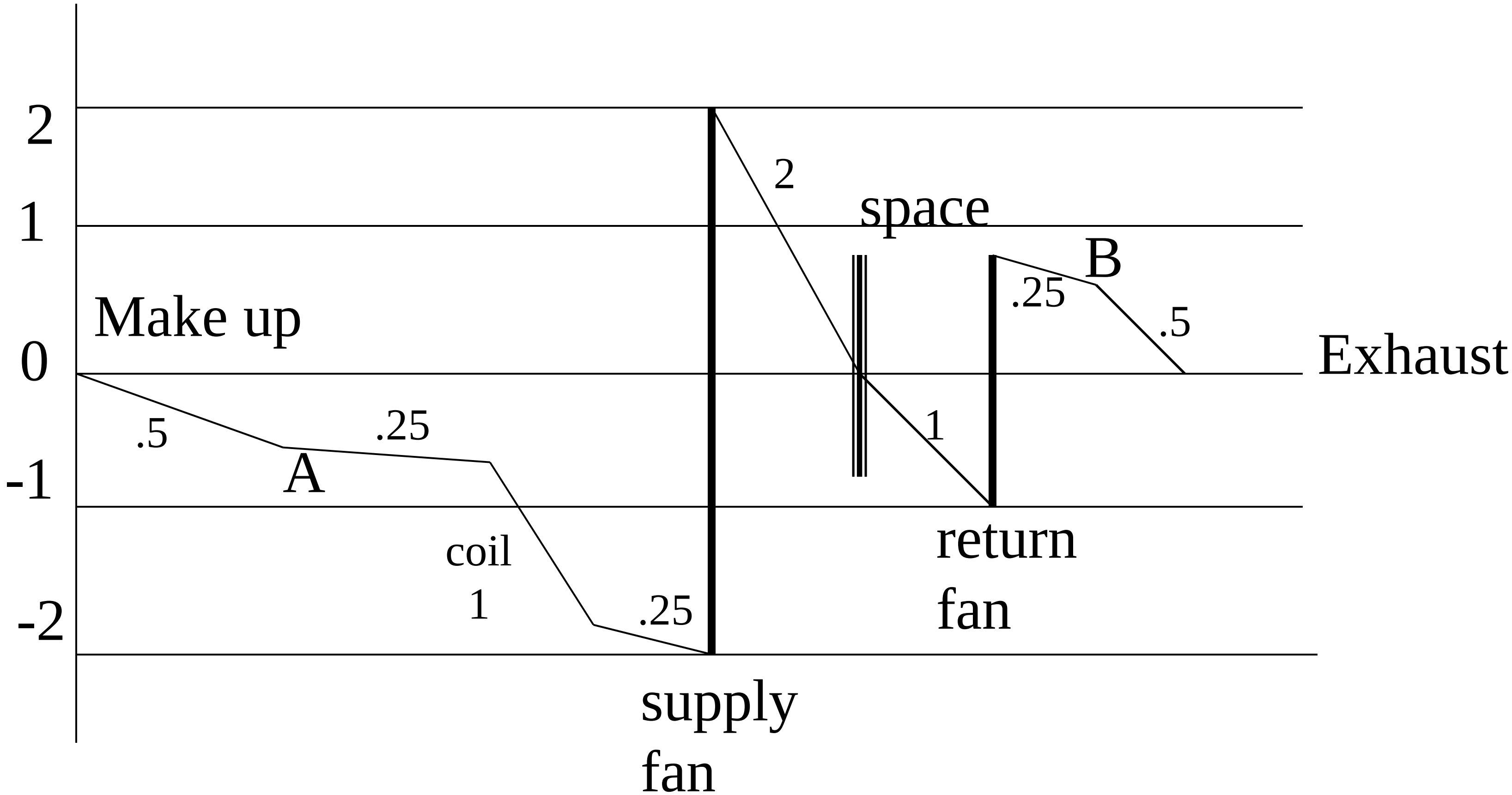


Figure 12-42 Schematic of a makeup and exhaust air system for Problem 12-47.

<u>12-47</u>

The exhaust, makeup and space are at the same pressure



Supply Fan 4 in water
Return Fan 1.75 in water

AIR SYSTEMS

Duct Loss Figure 12 – 21 page 420

Figure 12 - 22 page 421

Equivalent Length Method

Table 12 – 14 page 435

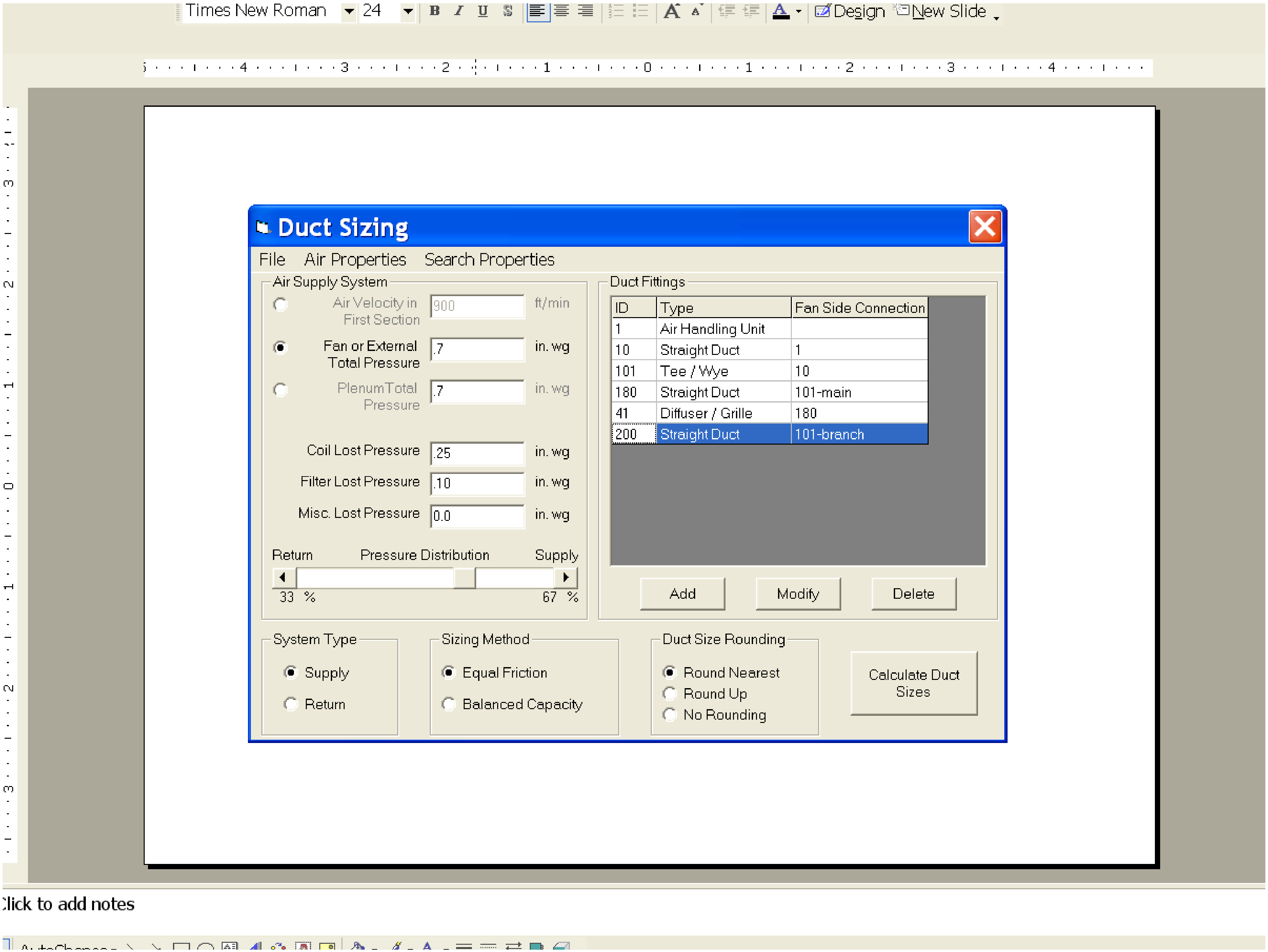
Loss Coefficient Method

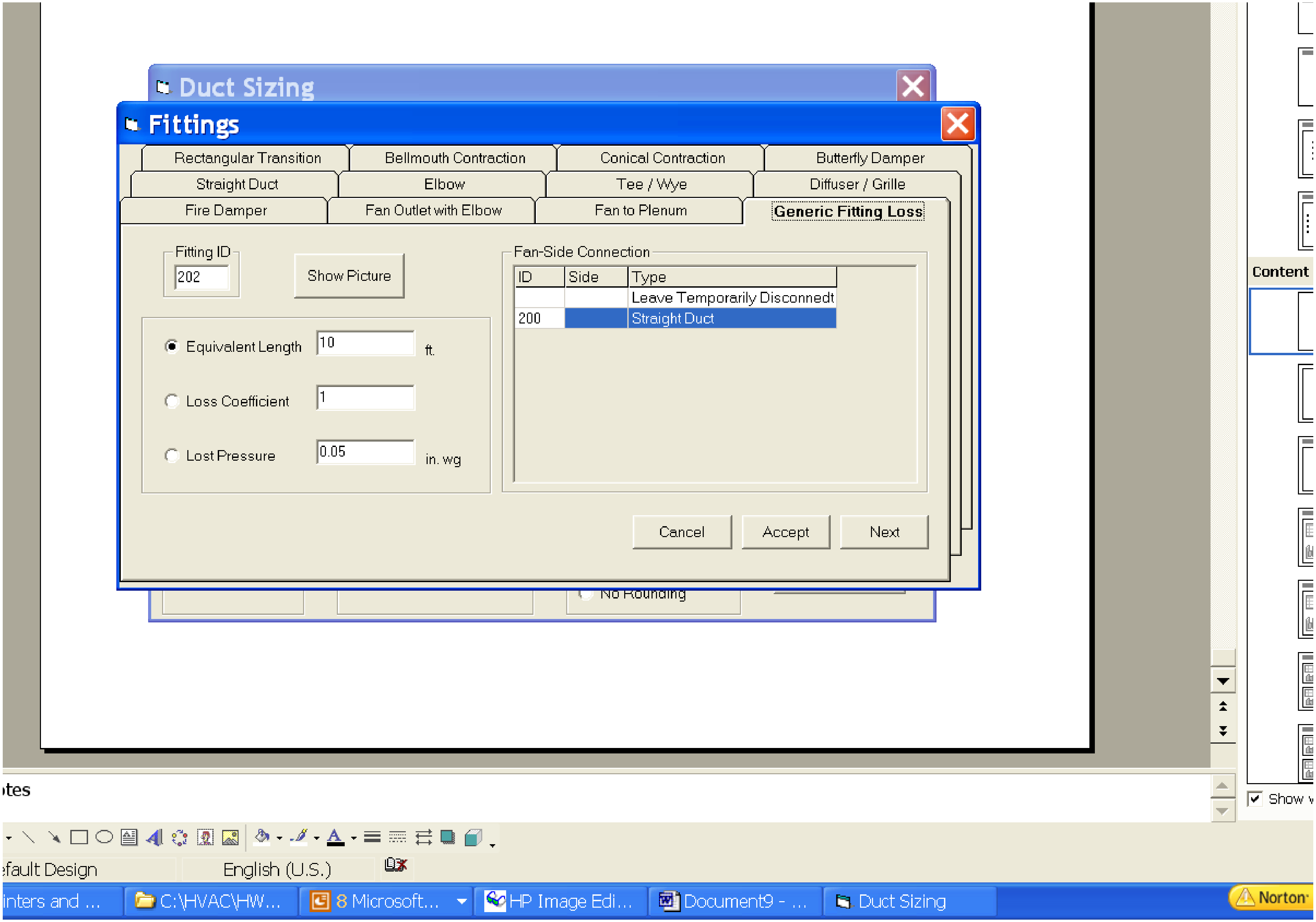
LIQUID SYSTEMS

Pipe Loss Figure 10 – 20,21 page 320

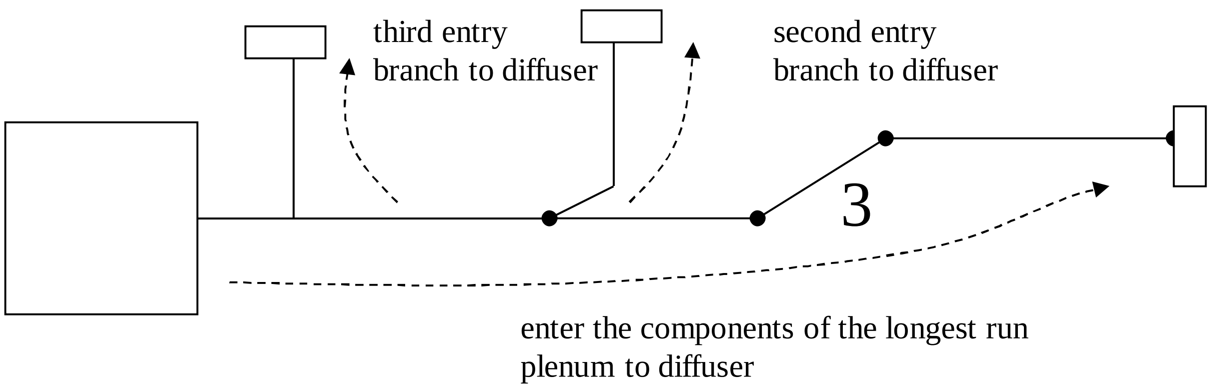
Figure 10 – 22b page 323

Equivalent Length


Times New Roman
24
Design
New Slide
Duct Sizing
File
Air Properties
Search Properties
Air Supply System
Air Velocity in First Section
900
ft/min
Fan or External Total Pressure
.7
in. wg
PlenumTotal Pressure
.7
in. wg
Coil Lost Pressure
.25
in. wg
Filter Lost Pressure
.10
in. wg
Misc. Lost Pressure
0.0
in. wg
Return
Pressure Distribution
Supply
33 %
67 %
Duct Fittings
ID | Type | Fan Side Connection
1 | Air Handling Unit |
10 | Straight Duct | 1
101 | Tee / Wye | 10
180 | Straight Duct | 101-main
41 | Diffuser / Grille | 180
200 | Straight Duct | 101-branch
Add
Modify
Delete
System Type
Supply
Return
Sizing Method
Equal Friction
Balanced Capacity
Duct Size Rounding
Round Nearest
Round Up
No Rounding
Calculate Duct Sizes
lick to add notes


Duct Sizing
Fittings
Rectangular Transition
Bellmouth Contraction
Conical Contraction
Butterfly Damper
Straight Duct
Elbow
Tee / Wye
Diffuser / Grille
Fire Damper
Fan Outlet with Elbow
Fan to Plenum
Generic Fitting Loss
Fitting ID
202
Show Picture
Fan-Side Connection
ID
Side
Type
Leave Temporarily Disconnedt
200
Straight Duct
Equivalent Length
10
ft.
Loss Coefficient
1
Lost Pressure
0.05
in. wg
Cancel
Accept
Next
No Rounding
Content
Show
tes
fault Design
English (U.S.)
inters and ...
C:\HVAC\HW...
8 Microsoft...
HP Image Edi...
Document9 - ...
Duct Sizing
Norton

DUCT SIZING Program



1. Enter the components of the longest run of ducting beginning with the plenum and ending with a diffuser.

2. Move to the first branch encountered going from the longest run diffuser to the plenum. Enter the components of this run ending with a diffuser.

3. Move to the next branch encountered going towarh the plenum. Enter the components of this run ending with a diffuser.

4. Repeat 3 for as many branches as the duct system has.

Basic Space Flow Patterns

Diffusers have been classified into five groups (1):

Group A. Diffusers mounted in or near the ceiling that discharge air horizontally.

Group B. Diffusers mounted in or near the floor that discharge air vertically in a nonspreading jet.

Group C. Diffusers mounted in or near the floor that discharge air vertically in a spreading jet.

Group D. Diffusers mounted in or near the floor that discharge air horizontally.

Group E. Diffusers mounted in or near the ceiling that project air vertically down.

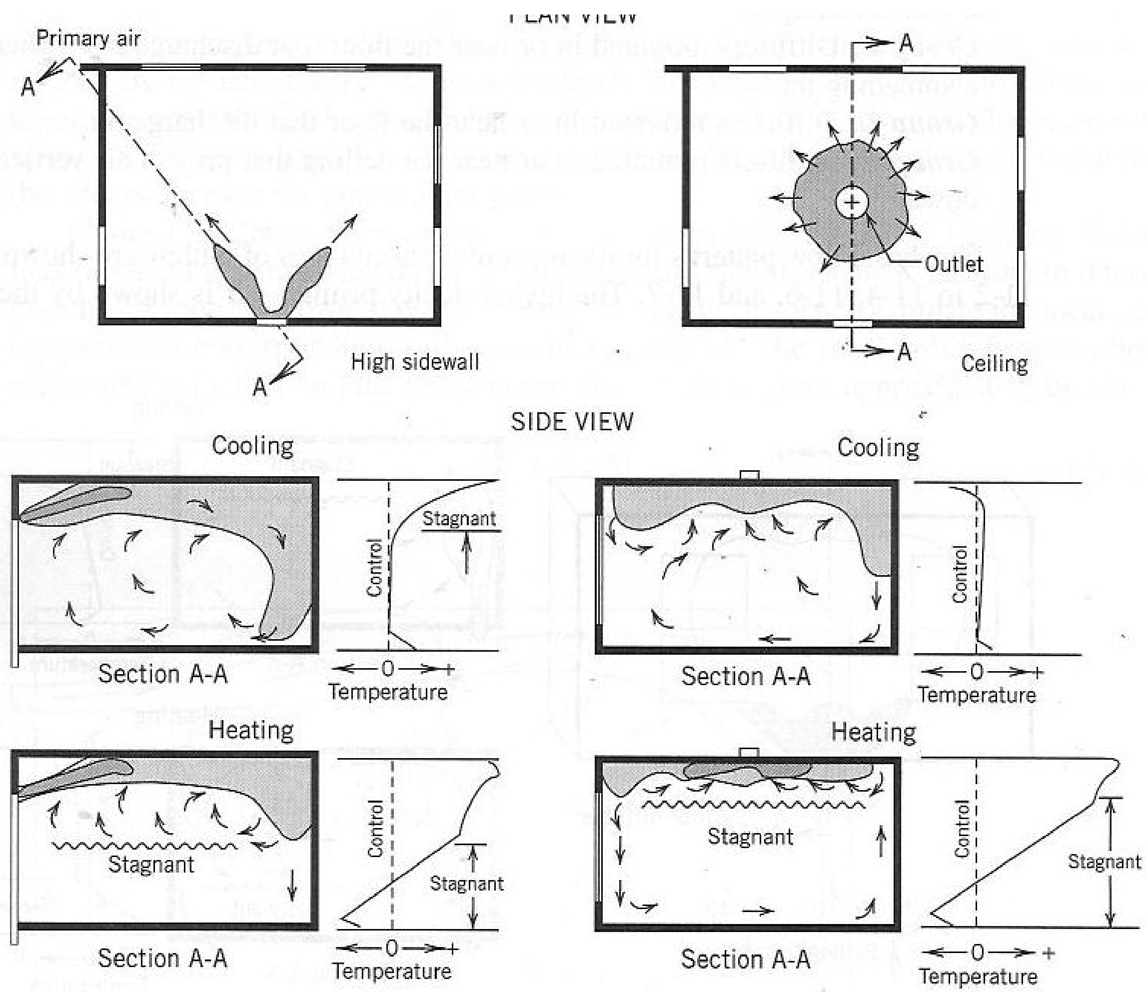


Figure 11-4 Air motion characteristics of Group A outlets. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume,* 1997.)

Group A, ceiling horizontal discharge

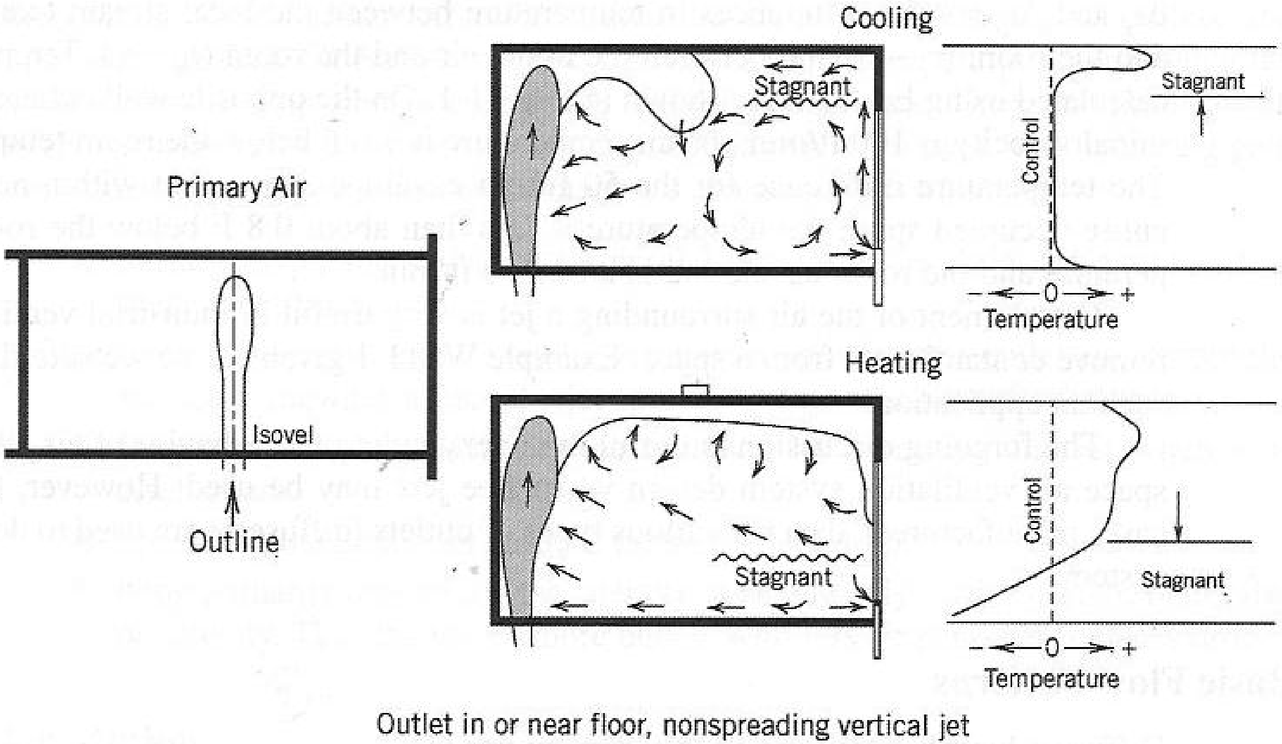


Figure 11-3 Air motion characteristics of Group B outlets. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.)

Group B, vertical jet discharge at floor

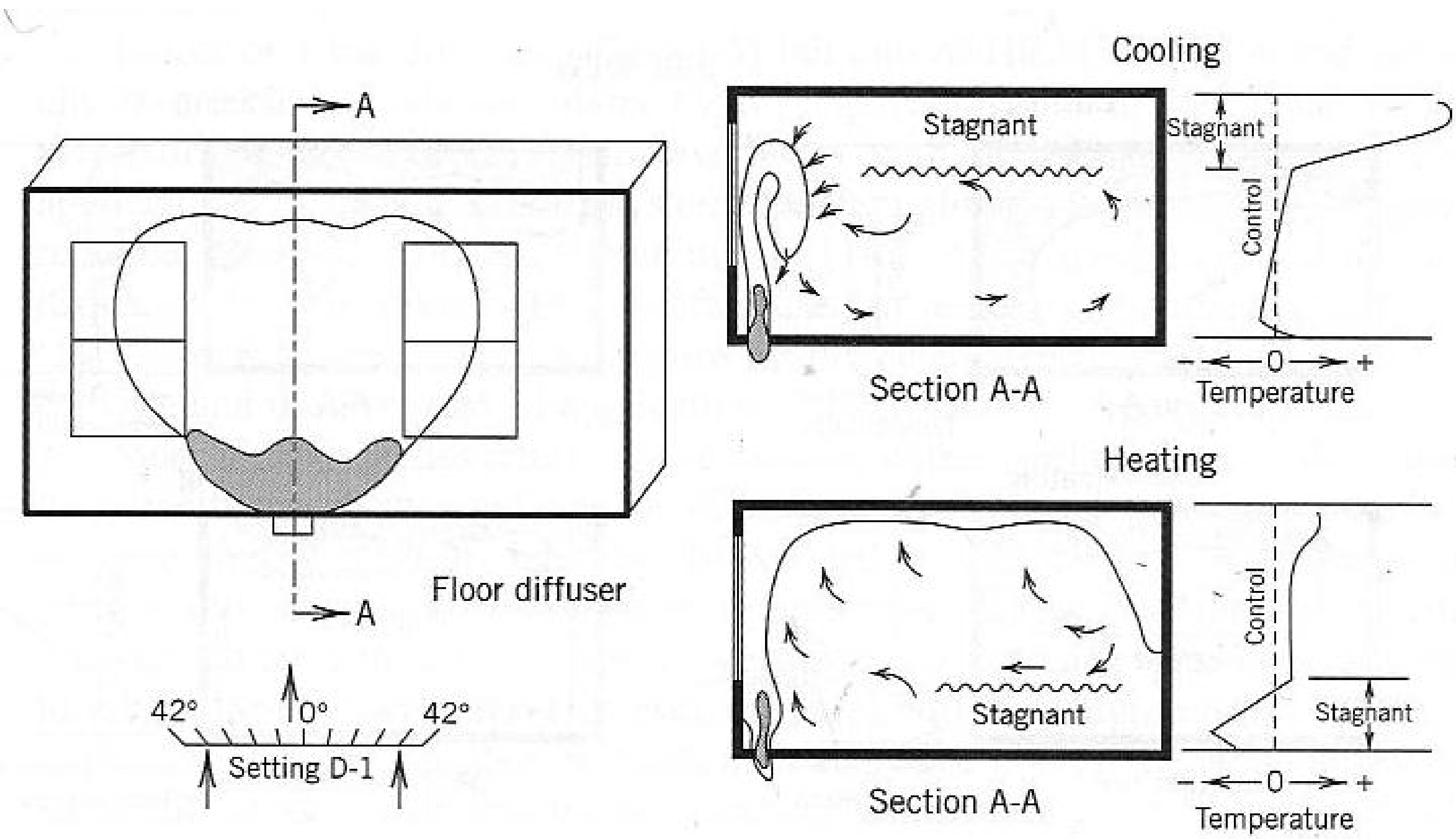


Figure 11-2 Air motion characteristics of Group C outlets. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.)

Group C, vertical spreading discharge at the floor

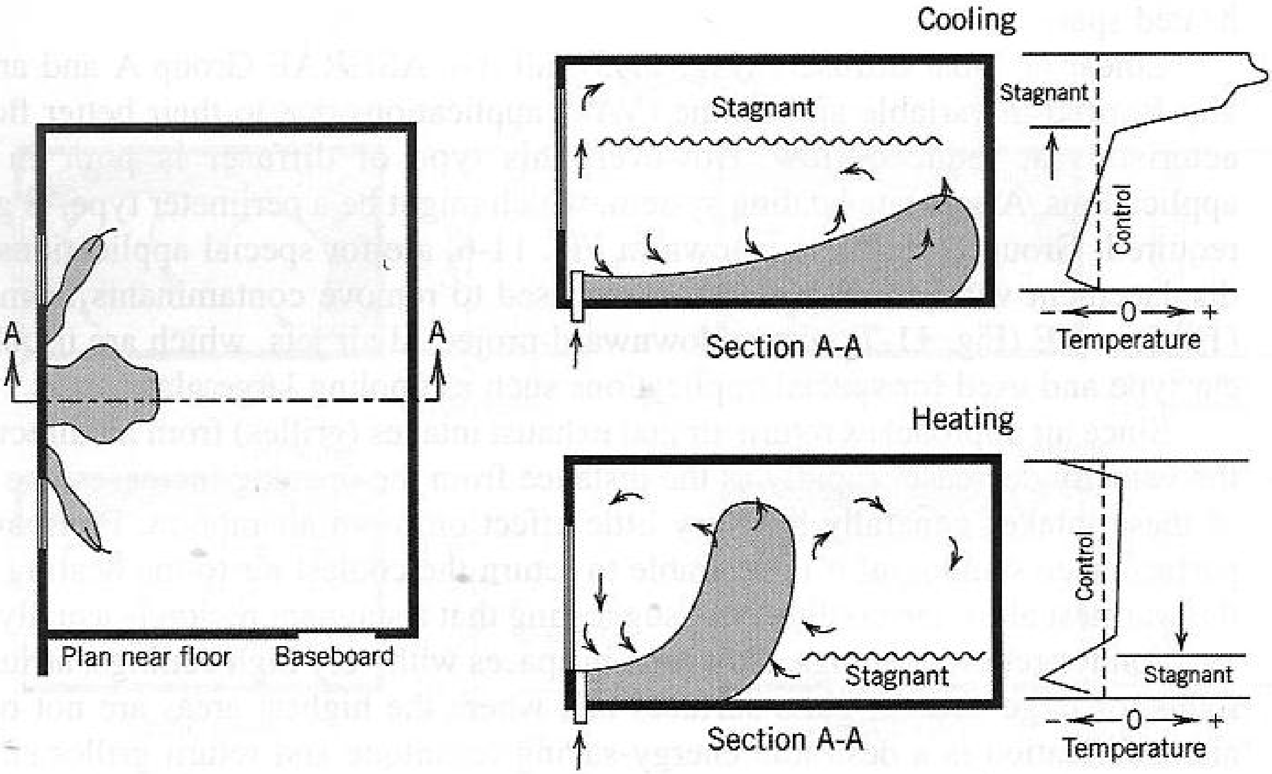


Figure 11-6 Air motion characteristics of Group D outlets. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.)

Group D, floor horizontal discharge

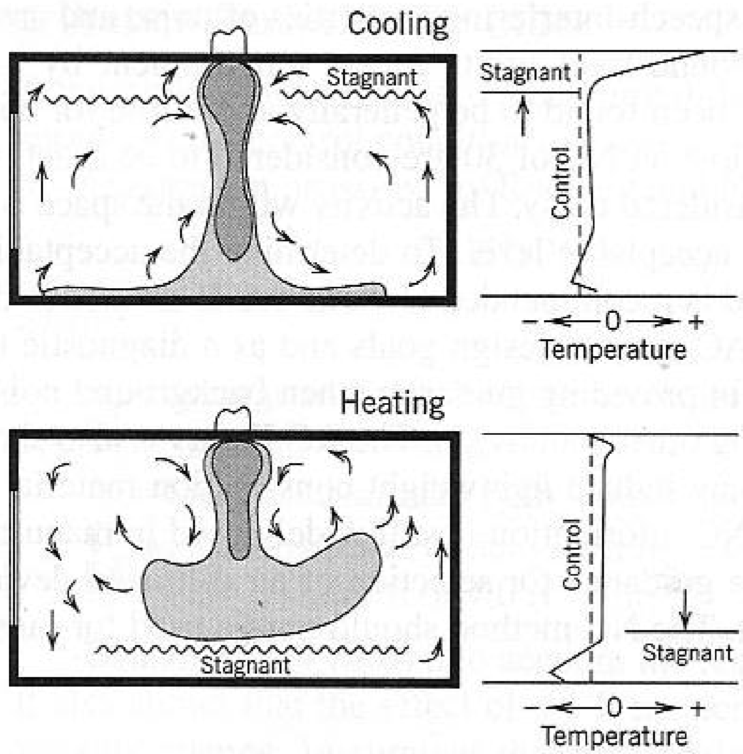


Figure 11-7 Air motion characteristics of Group E outlets. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.)

Group E, ceiling discharge down

SCALE MODEL TEST DATA

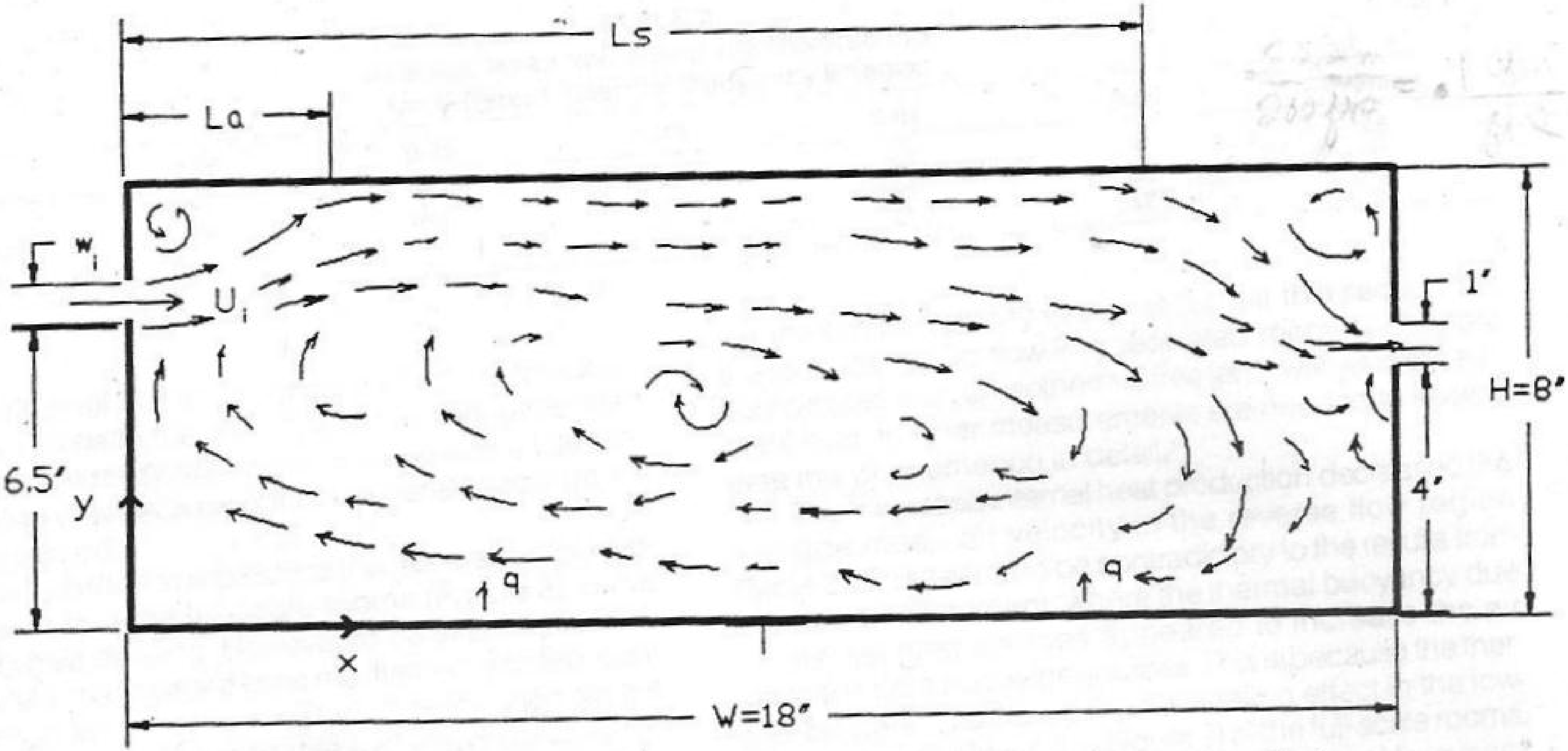


Figure 4 *Flow pattern studied in the 1/12th-scale model building (43 × 18 × 8 in.)*

$$\text{Inlet Velocity, } U_1 = 230 \text{ ft/min}$$

CFD CALCULATION

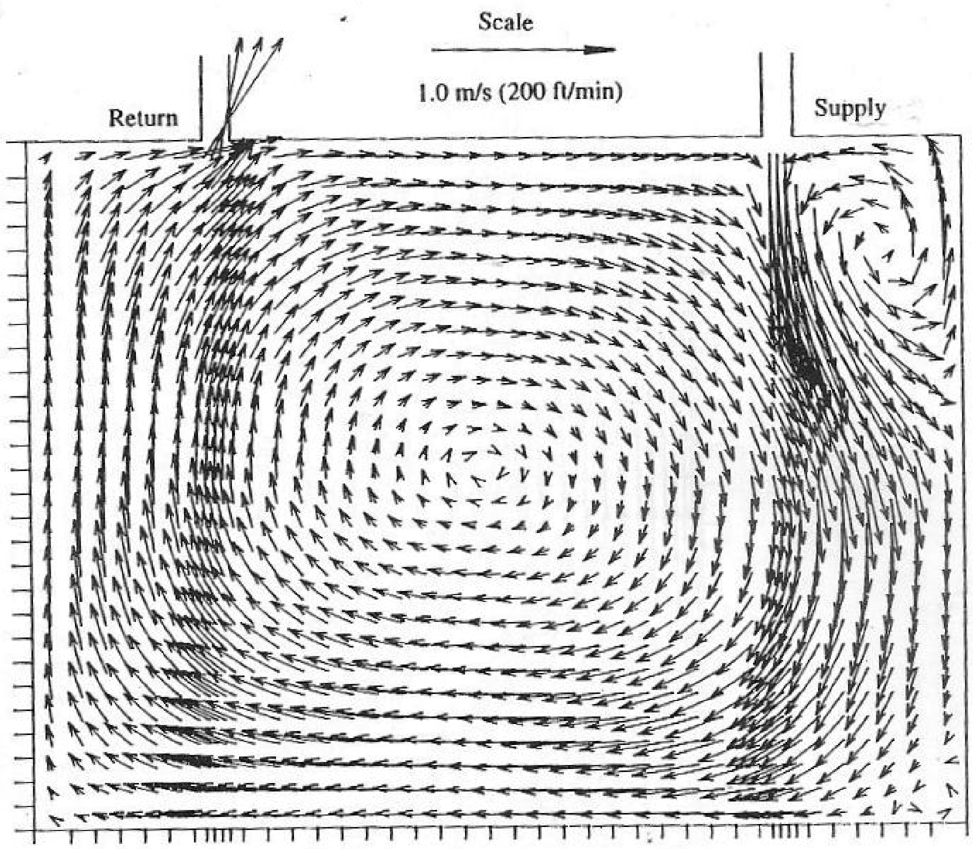


Figure 18.13 Velocity vector field simulated using high-Reynolds-number turbulence model from Launder and Spalding [5].

Supply downward at 200 ft/min
inlet temperature= room temperature

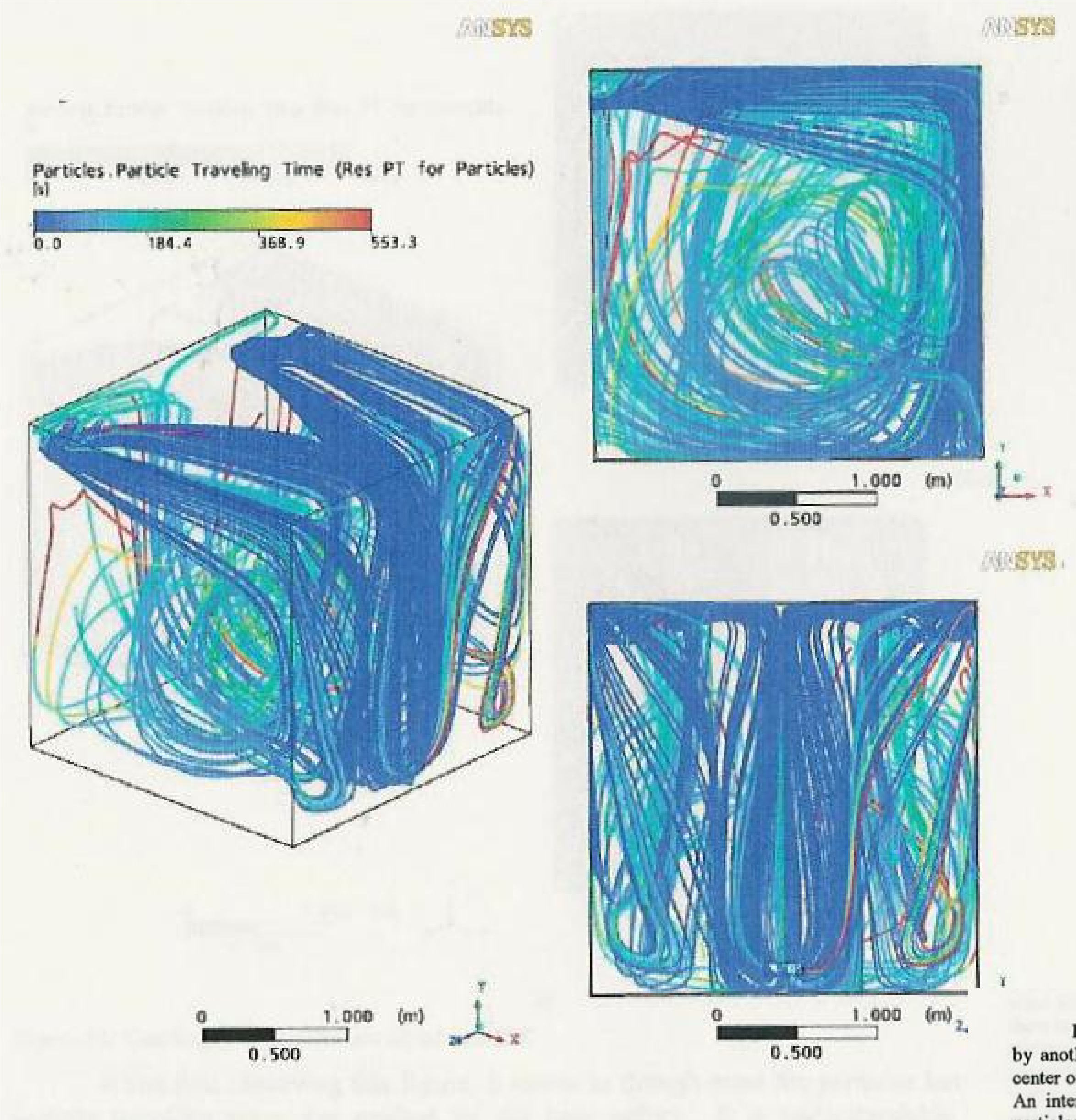


Figure 30: Coefficient of Restitution equal to .01

Ansys CFX CFD Code
Particle Tracking
NYSTAR Grant

In this figure one can observe the effect of lowering the coefficient of restitution by another order of magnitude. Again one can observe the vortices that form in the center of the room (left figure), and the two vortices that form in each corner of the room. An interesting change in this figure compared to the previous illustration is the few particles at seem to start "sticking" to the walls once they enter the boundary layers near the walls. This can be seen in the particles that are red which are located mainly on the inlet and two side walls. This means that the time the particle stays on the wall is much larger relative to the other particles traveling around the room. Once these particles are on the wall, they seem to stay there over the course of the simulation instead of being entrained by the main circulation within the room.

Chicago Weather Tape

```
LOCATION,Chicago Ohare Intl Ap,IL,USA,TMY3,725300,41.98,-87.92,-6.0,201.0
DESIGN CONDITIONS,1,Climate Design Data 2009 ASHRAE Handbook,,Heating,1,-20,-16.6,-25.7,0.4,-19.2,-22.1,0.5,-15.7,12.4,-3.5,11.4,-3.2,4
TYPICAL/EXTREME PERIODS,6,Summer - Week Nearest Max Temperature For Period,Extreme,7/13,7/19,Summer - Week Nearest Average Temperature
GROUND TEMPERATURES,3,.5,,,,-1.89,-3.06,-0.99,2.23,10.68,17.20,21.60,22.94,20.66,15.60,8.83,2.56,2,,,,2.39,0.31,0.74,2.45,8.10,13.21,17
HOLIDAYS/DAYLIGHT SAVINGS,No,0,0,0
COMMENTS 1,Custom/User Format -- WMO#725300; NREL TMY Data Set (2008); Period of Record 1973-2005 (Generally)
COMMENTS 2, -- Ground temps produced with a standard soil diffusivity of 2.3225760E-03 {m**2/day}
DATA PERIODS,1,1,Data,Sunday, 1/ 1,12/31
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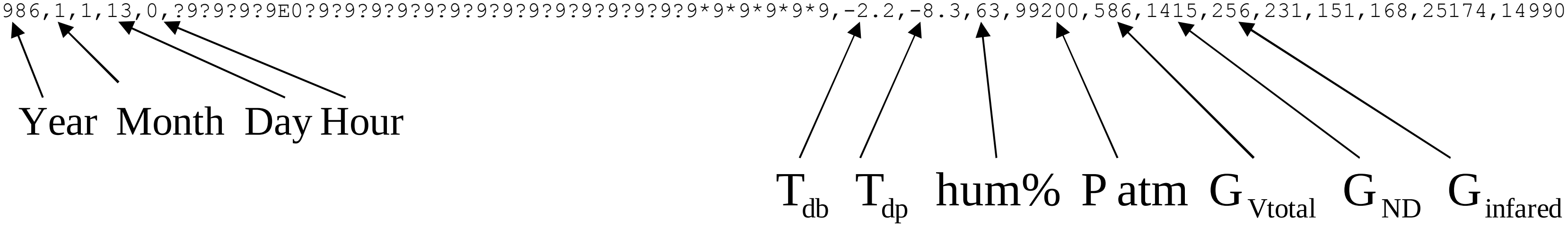


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COMMENTS 2, -- Ground temps produced with a standard soil diffusivity of 2.3225760E-03 {m**2/day}
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BIN WEATHER DATA

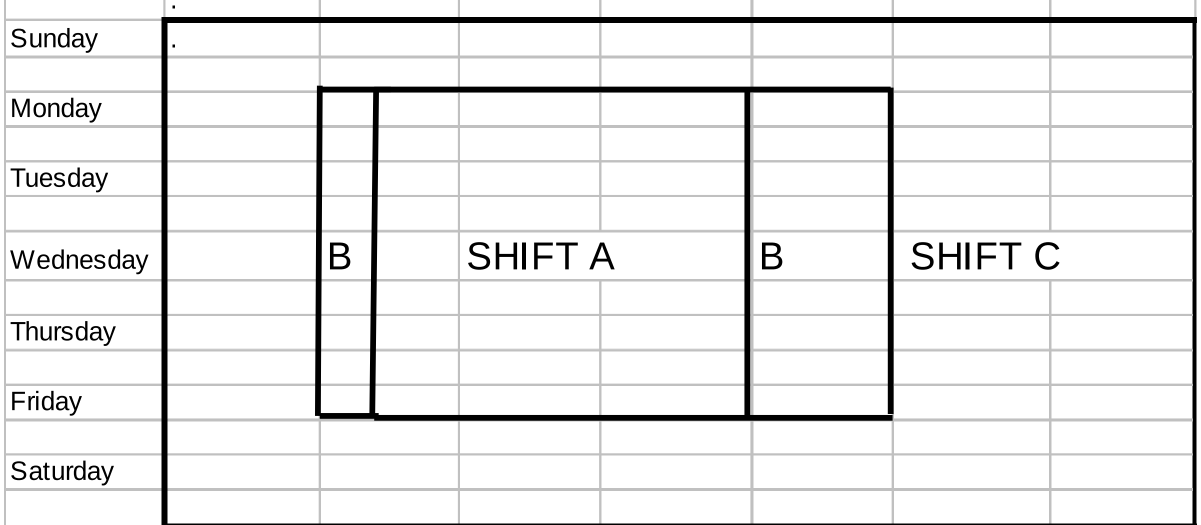
Table B-3 Annual Bin Weather Data for Chicago, Illinois, 41°47′N, 87°45′W, 607 ft Elevation

Temperature Bin, F	Hours of Dry-bulb Occurrence: MCWB						
	Time Group 1–4	5–8	9–12	13–16	17–20	21–24	Total
90/94	0:0	0:0	15:74	64:73	18:71	0:0	97:73
85/89	0:0	0:0	64:71	113:68	45:70	0:0	222:69
80/84	0:0	7:71	131:67	113:65	90:67	21:70	362:67
75/79	25:70	51:68	114:64	116:64	125:65	81:68	512:65
70/74	120:66	125:65	131:62	111:62	151:63	167:65	805:64
65/69	156:62	155:62	88:60	65:58	80:59	143:60	687:61
60/64	159:57	130:57	78:54	73:53	72:55	103:57	615:56
55/59	111:52	112:52	93:49	109:49	96:51	101:52	622:51
50/54	112:48	92:48	88:46	89:47	100:47	104:47	585:47
45/49	92:44	102:43	97:42	96:40	83:42	107:43	577:42
40/44	120:38	104:39	72:37	94:37	131:37	115:38	636:38
35/39	106:34	105:34	133:33	134:33	131:33	111:34	720:33
30/34	182:30	182:30	156:29	129:29	135:30	173:30	957:30
25/29	93:25	98:25	66:25	68:24	94:25	92:25	511:25
20/24	75:21	75:21	60:20	40:20	44:21	60:21	354:21
15/19	63:16	56:16	30:16	21:15	33:15	40:16	243:16
10/14	25:11	32:11	25:10	13:10	11:11	19:11	125:11
5/9	5:7	14:6	11:5	8:6	16:6	12:5	66:6
0/4	15:1	16:0	7:0	4:1	5:0	11:1	58:0
–5/–1	1:–4	4:–7	1:–6	0:0	0:0	0:0	6:–6

Source: Reprinted by permission from *Bin and Degree Hour Weather Data for Simplified Energy Calculations*, ASHRAE, Inc., Atlanta, GA, 1986.

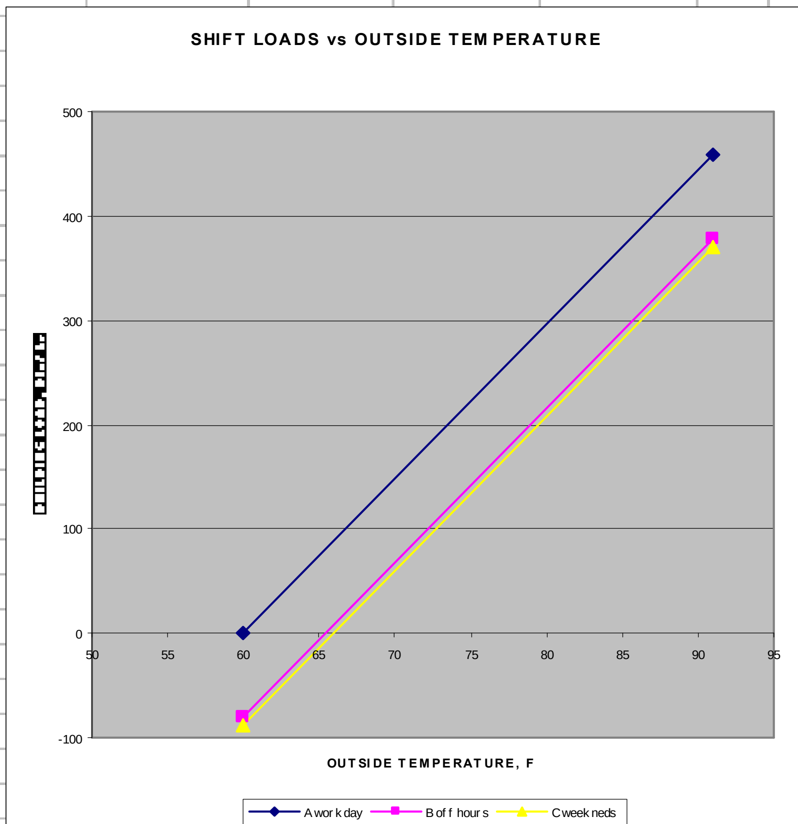
Chicago BIN Data page 603

	I	II	III	IV	V	VI	Total
	1 am- 4 am	5 am - 8 am	9 am - 12 a	1 pm - 4 pm	5 pm -8 pm	9 pm - 12 pm	
90/94	0	0	15	64	18	0	97
85/89	0	0	64	113	45	0	222
80/84	0	7	131	113	90	21	362
75/79	25	51	114	116	125	81	512
70/74	120	125	131	111	151	167	805
65/69	156	155	88	65	80	143	687
60/64	159	130	78	73	72	103	615
	..						3300



Hrs Shift A	0	5	20	20	0	0	45
Hrs Shift B	0	10	0	0	20	0	30
Hrs Shift C	28	13	8	8	8	28	93
Total	28	28	28	28	28	28	168
Fraction A	0.000	0.179	0.714	0.714	0.000	0.000	
Fraction B	0.000	0.357	0.000	0.000	0.714	0.000	
Fraction C	1.000	0.464	0.286	0.286	0.286	1.000	
Total	1.000	1.000	1.000	1.000	1.000	1.000	

	SHIFT A		SHIFT B		SHI	
	Load	T	Load	T	Load	T
	458	91	378	91	370	91
	0	60	-80	60	-88	60



$$\text{Load A} = \left(\frac{458}{31}\right) \times (T_{\text{outside}} - 60)$$

$$\text{Load B} = \left(\frac{480}{31}\right) \times (T_{\text{outside}} - 60) - 80$$

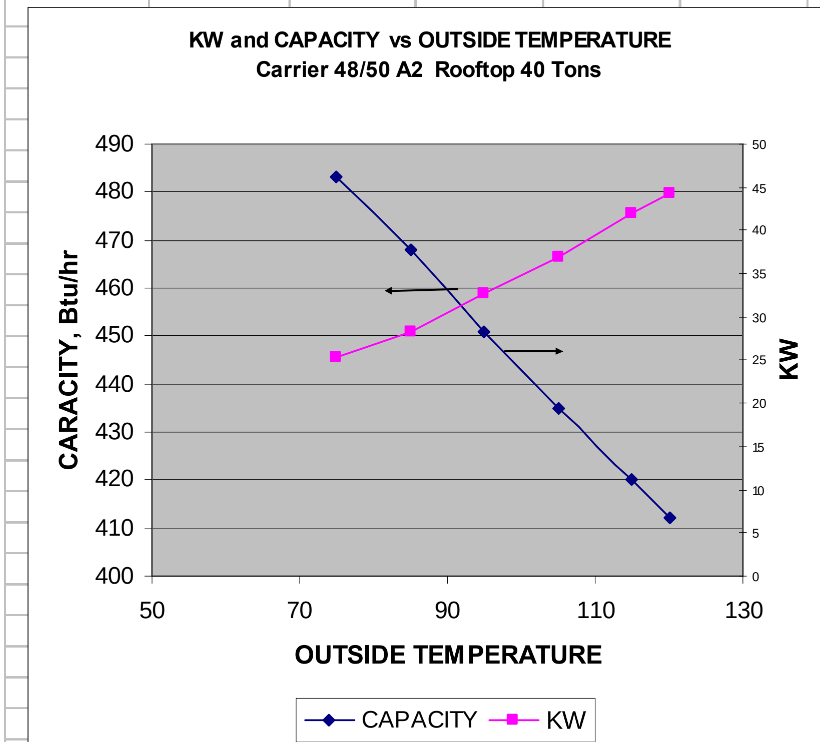
$$\text{Load C} = \left(\frac{480}{31}\right) \times (T_{\text{outside}} - 60) - 88$$

CARRIER

48/50A2,A3,A4,A5040 (40 TONS) (cont) cfm

Temp (F) Air Entering Condenser (Edb)		16,000				
		75	72	67	62	57
	TC	579	555	519		477
	SHC	249	304	394		477
	kW	26.9	26.5	25.9		25.3
	BF	0.10	0.09	0.07		0.35
	TC	560	539	503		464
	SHC	243	298	388		464
	kW	30.2	29.9	29.3		28.7
	BF	0.10	0.09	0.07		0.37
	TC	543	522	485		450
	SHC	238	292	380		450
	kW	34.0	33.7	33.2		32.6
	BF	0.09	0.08	0.08		0.39
	TC	523	501	464		434
	SHC	232	285	373		434
	kW	38.2	38.0	37.5		37.0
	BF	0.09	0.08	0.07		0.41
	TC	500	479	443		417
	SHC	224	278	365		417
	kW	43.1	42.9	42.7		42.2
	BF	0.09	0.08	0.07		0.43
	TC	488	467	432		408
	SHC	220	274	361		408
	kW	45.9	45.7	45.5		45.2
	BF	0.09	0.08	0.07		0.45

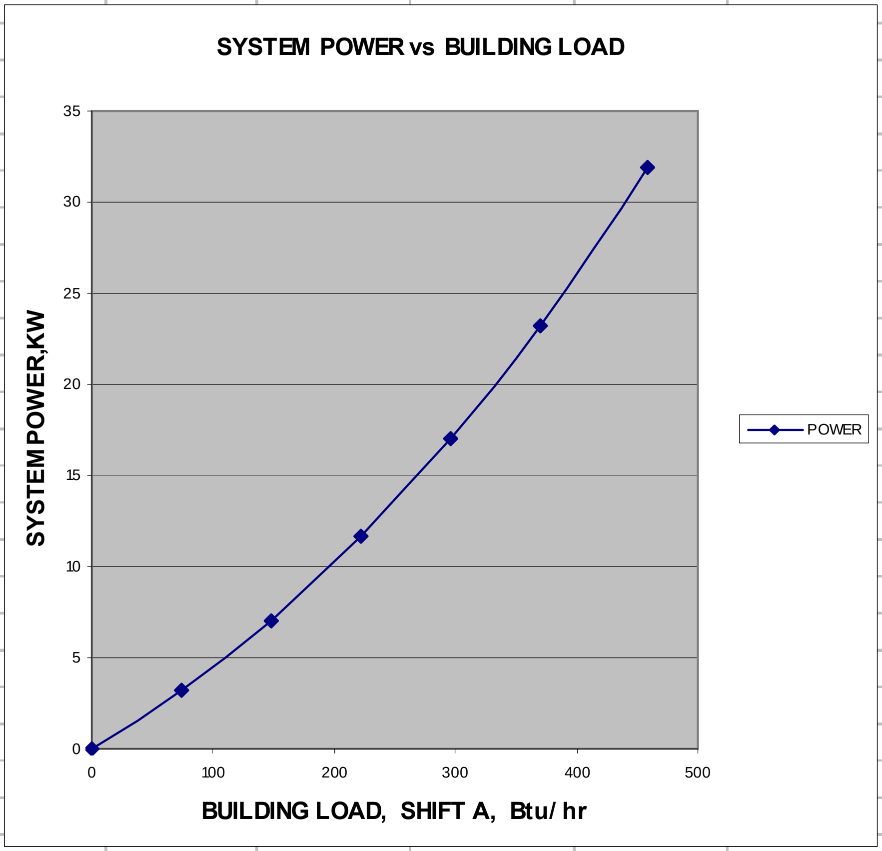
	Outside Temperature	Q	KW
CARRIER	75	483	25.4
48/50	85	468	28.2
Rooftop	95	451	32.6
40 Tons	105	435	36.9
	115	420	42
	120	412	44.4



$$CAPACITY = 412. + \left(\frac{71}{45}\right) \times (120 - T_{outside})$$

$$KW = 44.4 \times \left(\frac{T_{outside}}{120}\right)^{1.2}$$

Outside Temperature	SYSTEM KW	SYSTEM CAPACITY	BUILDING LOAD A	LOAD/ CAPACITY	POWER
91	31.915155	457.75556	458	1.000534	31.9321976
85	29.406951	467.22222	369.354839	0.7905335	23.247181
80	27.343573	475.11111	295.483871	0.6219258	17.0056744
75	25.305842	483	221.612903	0.4588259	11.6109756
70	23.295119	490.88889	147.741935	0.3009682	7.01108954
65	21.312937	498.77778	73.8709677	0.148104	3.1565306
60	19.361045	506.66667	0	0	0



DESIGN LOAD, 91 F 458,000 Btu/hr

100 % people,lights, eqpmt, ventilation 88,000 Btu/hr

10 % people,lights, eqpmt, ventilation 8,000 Btu/hr

$$\text{CAPACITY} = 412. + \left(\frac{71}{45}\right) \times (120 - T_{outside})$$

$$\text{KW} = 44.4 \times \left(\frac{T_{outside}}{120}\right)^{1.2}$$

$$\text{Load A} = \left(\frac{458}{31}\right) \times (T_{outside} - 60)$$

$$\text{Load B} = \left(\frac{458}{31}\right) \times (T_{outside} - 60) - 80$$

$$\text{Load C} = \left(\frac{458}{31}\right) \times (T_{outside} - 60) - 88$$

$$BIN\ Hrs \times Shift\ Fraction$$

$$Power = Run\ Fraction \times System\ KW \times BIN\ HRS$$

$$Run\ Fraction = BuildingLoad/SystemCapacity$$

$$SystemKW = 44.4(T_{outside}/120)^{1.2}$$

$$SystemCapacity = 412. + (71/45) \times (120 - T_{outside})$$

$$Building\ Load\ A = (458/31) \times (T_{outside} - 60)$$

SHIFT A

Outside Temperature	I 1 am- 4 am	II 5 am - 8 am	III 9 am - 12 a	IV 1 pm - 4 pm	V 5 pm -8 pm	VI 9 pm -12 pm	Total Hrs	Building Load A	System Capacity	System KW	Run Fraction	POWER KW/hr
92.5	0.00	0.00	10.71	45.71	0.00	0	56.43	480.16	455.39	32.488941	1.0543983	1933.033
87.5	0.00	0.00	45.71	80.71	0.00	0	126.43	406.29	463.28	30.39311	0.8769907	3369.887
82.5	0.00	1.25	93.57	80.71	0.00	0	175.54	332.42	471.17	28.321106	0.7055239	3507.417
77.5	0.00	9.11	81.43	82.86	0.00	0.00	173.39	258.55	479.06	26.27408	0.5397044	2458.752
72.4	0.00	22.32	93.57	79.29	0.00	0	195.18	183.20	487.10	24.213176	0.3761018	1777.417
67.5	0.00	27.68	62.86	46.43	0.00	0	136.96	110.81	494.83	22.260248	0.2239268	682.7213
62.5	0.00	23.21	55.71	52.14	0.00	0	131.07	36.94	502.72	20.296516	0.7959889	2117.564
							995.00					

SHIFT B

$$Building\ Load\ B = (458/31) \times (T_{outisde} - 60) - 80$$

Temperature	I	II	III	IV	V	VI	Total Hrs	Building Load	System Capacity	System KW	Run Fraction	POWER KW/hr
92.5	0.00	0.00	0.00	0.00	12.86	0	12.86	400.16	455.39	32.488941	0.8787243	367.0563
87.5	0.00	0.00	0.00	0.00	32.14	0	32.14	326.29	463.28	30.39311	0.7043082	688.0537
82.5	0.00	2.50	0.00	0.00	64.29	0	66.79	252.42	471.17	28.321106	0.5357326	1013.309
77.5	0.00	18.21	0.00	0.00	89.29	0.00	107.50	178.55	479.06	26.27408	0.3727091	1052.703
72.5	0.00	44.64	0.00	0.00	107.86	0	152.50	104.68	486.94	24.253314	0.2149679	795.0868
67.5	0.00	55.36	0.00	0.00	57.14	0	112.50	30.81	494.83	22.260248	0.0622562	155.9069
62.5	0.00	46.43	0.00	0.00	51.43	0	97.86	-43.06				
							582.14					

SHIFT C

$$Building\ Load\ C = (458/31) \times (T_{outisde} - 60) - 88$$

Temperature	I	II	III	IV	V	VI	Total Hrs	Building Load	System Capacity	System KW	Run Fraction	POWER KW/hr
92.5	0.00	0.00	4.29	18.29	5.14	0	27.71	392.16	455.39	32.488941	0.8611569	775.3924
87.5	0.00	0.00	18.29	32.29	12.86	0	63.43	318.29	463.28	30.39311	0.6870399	1324.47
82.5	0.00	3.25	37.43	32.29	25.71	21	119.68	244.42	471.17	28.321106	0.5187535	1758.278
77.5	25.00	23.68	32.57	33.14	35.71	81.00	231.11	170.55	479.06	26.27408	0.3560096	2161.736
72.5	120.00	58.04	37.43	31.71	43.14	167	457.32	96.68	486.94	24.253314	0.1985389	2202.106
67.5	156.00	71.96	25.14	18.57	22.86	143	437.54	22.81	494.83	22.260248	0.0460892	448.8924
62.5	159.00	60.36	22.29	20.86	20.57	103	386.07	-51.06				
							1722.86					

SEASON COOLING POWER, KW Hr **28589.78**