HVAC II

Course Notes

$$1\ ft^3 \equiv 7.48\ gal$$

$$1\ \frac{lb}{ft \cdot s} \equiv 1490\ \text{centipoises}\quad (\text{viscosity})$$

$$\text{specific gravity} = \frac{\rho}{\rho_{water}}$$

$$\rho_{water} = 62.4\ \frac{lb}{ft^3}$$

$$g = 32.17\ \frac{lb_m \cdot ft}{lb_f \cdot s^2}$$

$$\Delta P \left[\frac{lb_f}{ft^2}\right] = y\,[\text{in water}] \times \frac{1}{12} \times \rho_{water}$$

$$\Delta P \left[\frac{lb_f}{ft^2}\right] = y\,[\text{in Hg}] \times \underbrace{70.704}$$

$$\Delta P\,[ft\ water] = \Delta P\,[ft_{Hg}] \times \frac{\rho_{Hg}}{\rho_{water}} \qquad \frac{\rho_{Hg}}{12}$$

Atmospheric pressure: 29.92 in Hg

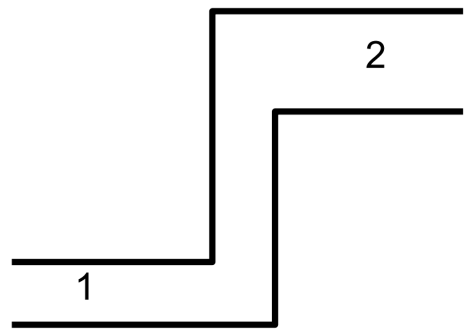
$2116.02 \ lbf/ft^2$

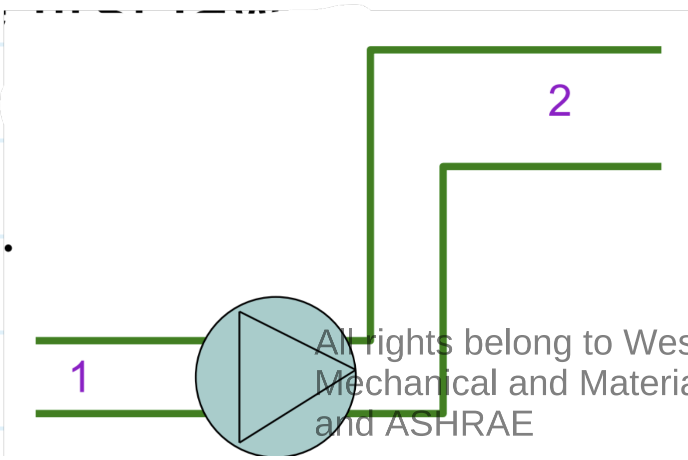
Local barometric pressure: 14.7 Psia = 0 Psig

$1 \text{ psi} = 144 \ lbf/ft^2$

$Psia = Psig + 14.7$

$$\dot{Q} = V_1 A_1 = V_2 A_2$$





Bernoulli's principle

$$\frac{P_1}{\rho_1} + \frac{V_1^2}{2g_c} + \frac{Z_1 g}{g_c} = \frac{P_2}{\rho_2} + \frac{V_2^2}{2g_c} + \frac{Z_2 g}{g_c} - w + \frac{g}{g_c} l_f$$

required work

head loss

P= static pressure, lbf/ft^2 or N/m^2
ρ= mass density, lbm/ft^3 or kg/m^3
V= average velocity, ft/sec or m/sec
g= gravity, ft/sec^2 or m/s^2
g_c= constant =32.17 (lbm-ft)/(lbf-sec^2)
=1.0 (kg-m)/(N-s^2)
Z= elevation, ft or m
w= work, (ft-lbf)/lbm or J/kg
l_f = lost head, ft or m

Regular (friction) head loss

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

moody friction factor

Head loss in fittings

$$\Delta h_{1-2} = K \frac{v^2}{2g}$$

$$\Delta P_{1-2} = \left(\frac{\dot{Q}}{C_d}\right)^2$$

$$C_d : \left[\frac{gpm}{psi}\right]$$

Table 1 *K* Factors—Threaded Pipe Fittings

Nominal Pipe Dia., in.	90° Standard Elbow	90° Long-Radius Elbow	45° Elbow	Return Bend	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve	Bell Mouth Inlet	Square Inlet	Projected Inlet
3/8	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
1/2	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
3/4	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
1	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
1-1/4	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
1-1/2	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
2	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
2-1/2	0.85	0.35	0.30	0.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
3	0.80	0.31	0.29	0.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
4	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: *Engineering Data Book* (Hydraulic Institute 1979).

Table 2 *K* Factors—Flanged Welded Pipe Fittings

Nominal Pipe Dia., in.	90° Standard Elbow	90° Long-Radius Elbow	45° Long-Radius Elbow	Return Bend Standard	Return Bend Long-Radius	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve
1	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
1-1/4	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
1-1/2	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
2	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
2-1/2	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
3	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
4	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
6	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
8	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
10	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
12	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

Source: *Engineering Data Book* (Hydraulic Institute 1979).

All rights belong to Western University; Mechanical and Materials Engineering Department and ASHRAE

Table 10-1 Formulas, Definition of Terms, and Values of f_t for Fig. 10-24

Formula 1: $K_2 = \dfrac{K_1 + \left(\sin\frac{\theta}{2}\right)0.8(1-\beta^2) + 2.6(1-\beta^2)^2}{\beta^4}$

Formula 2: $K_2 = \dfrac{K_1 + 0.5\left(\sin\frac{\theta}{2}\right)(1-\beta^2) + (1-\beta^2)^2}{\beta^4}$

$\beta = \dfrac{D_1}{D_2}$; $\beta^2 = \left(\dfrac{D_1}{D_2}\right)^2 = \dfrac{A_1}{A_2}$; D_1 = smaller diameter, A_1 = smaller area

Nominal Size, in.	Friction Factor f_t	Nominal Size, in.	Friction Factor f_t
$\frac{1}{2}$	0.027	4	0.017
$\frac{3}{4}$	0.025	5	0.016
1	0.023	6	0.015
$1\frac{1}{4}$	0.022	8–10	0.014
$1\frac{1}{2}$	0.021	12–16	0.013
2	0.019	18–24	0.012
$2\frac{1}{2}$,3	0.018		

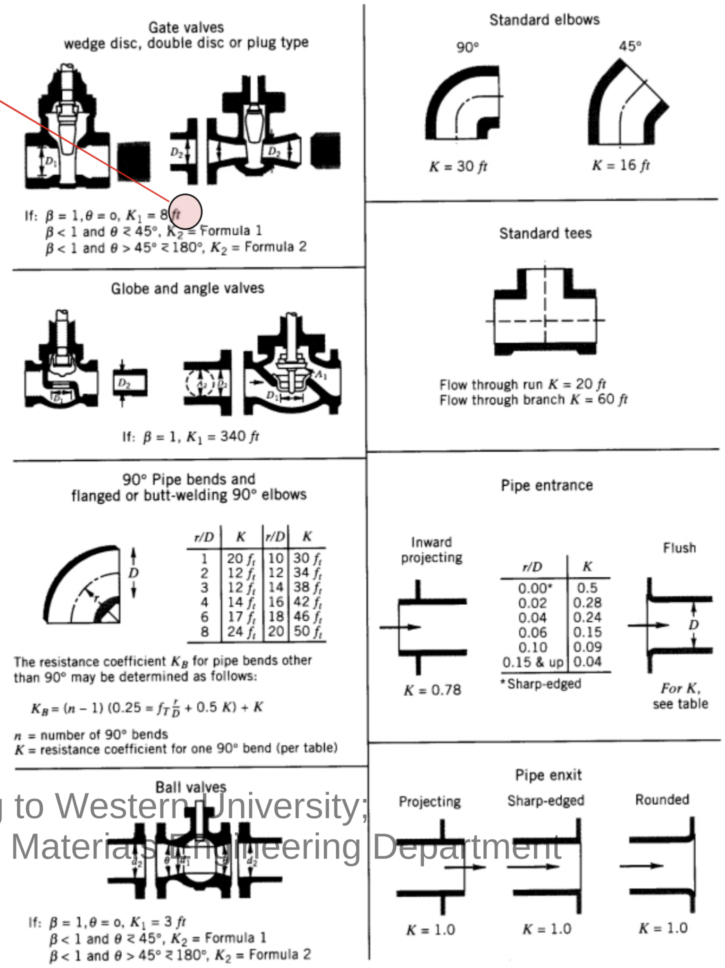
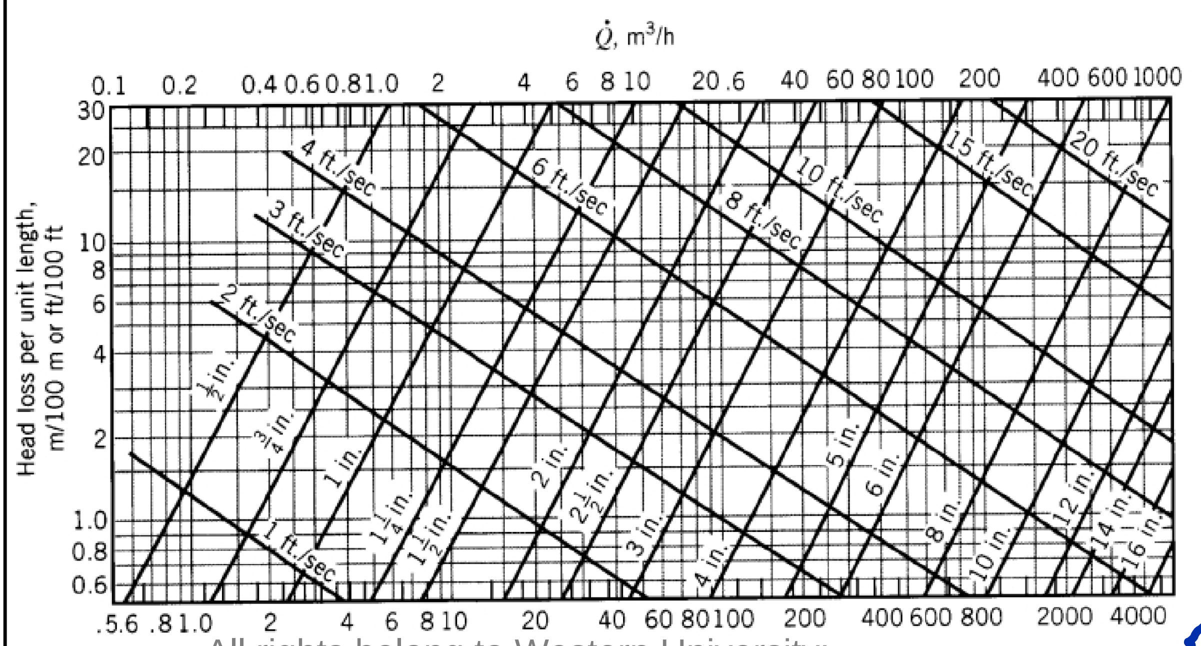


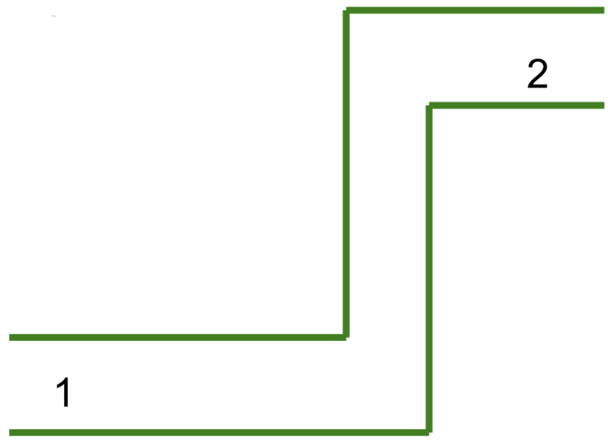
Figure 10-24a Resistance coefficients K for various valves and fittings. (Courtesy of the Crane Company, Technical Paper No. 410)

The standard design friction loss should be in the range of 2.5 - 4 ft/100ft.



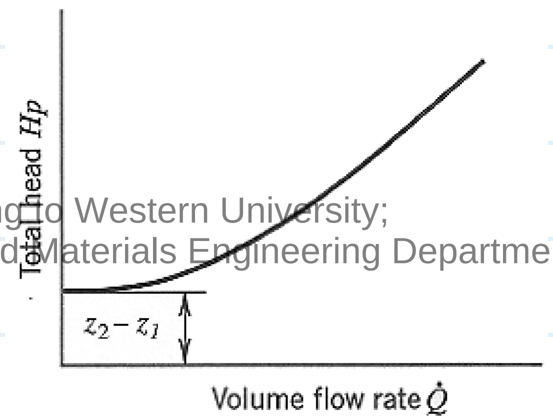
regular loss

Figure 10-22 Friction loss due to flow of water in commercial steel pipe (schedule 40). (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1989)



Simplified Eq.:

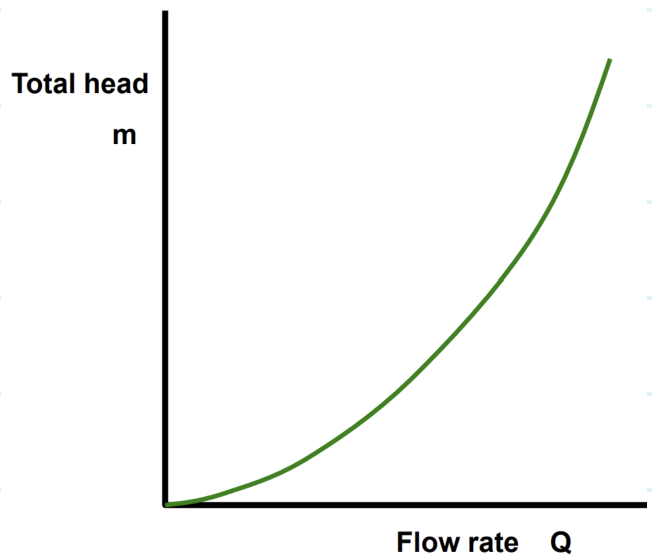
$$\Delta h_{1-2} \equiv \Delta z + a\dot{Q}^2$$



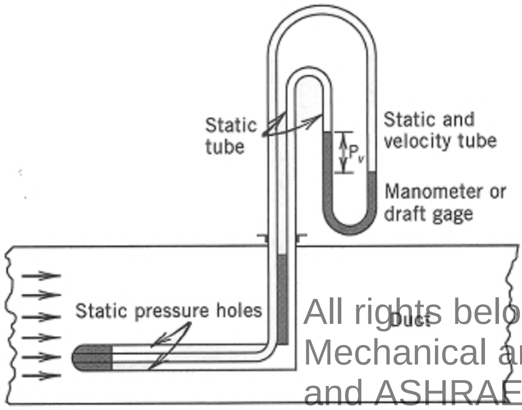
in open-loop systems

in closed-loop systems

$$\Delta h_{1-2} \equiv c\dot{Q}^2$$

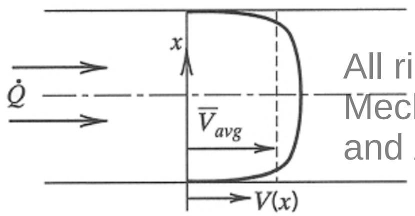


Pitot Pipe



$$P_v = \frac{\overline{V_1^2}\rho}{2g_c} = P_t - P_1$$

Figure 10-8 Pitot tube in a duct.



All rights belong to Western University; Mechanical and Materials Engineering Department and ASHRAE

For practical cases $V_{avg}/V_{max} = 0.82$

Figure 10-9 Velocity profile in a pipe or duct.

we should put pitot tube at the center of the duct to measure the V_{max}. Then we can calculate V_{avg}.

for both pressure & flow rate.

$P_t = P_1 + P_v$ = static P + dynamic P

$$P_t = P_1 + \frac{\rho v^2}{2g}$$

Orifice plate

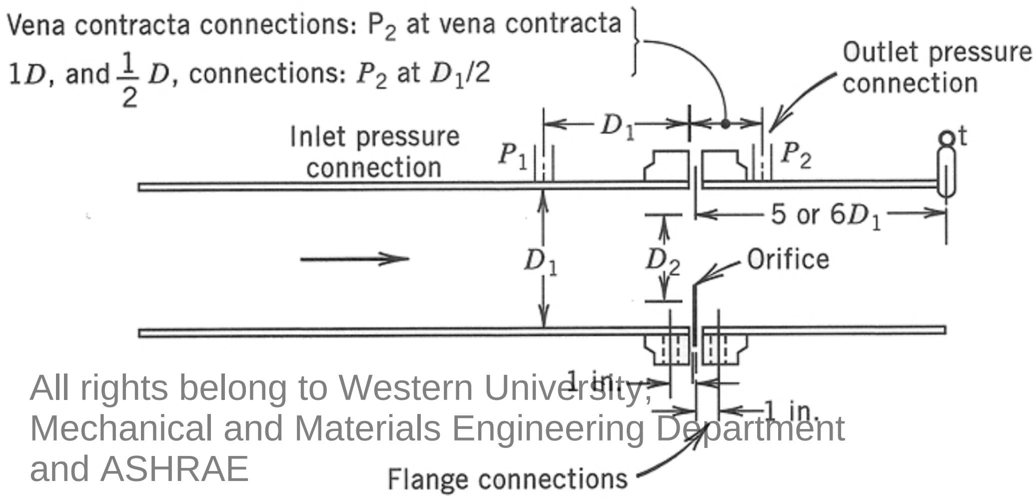


Figure 10-11 Recommended location of pressure taps for use with thin-plate and square-edged orifices according to reference 3.

$$P_1 - P_2 = \left(\frac{Q}{C_d A_2}\right)^2 \frac{\rho}{2g_c}$$

for measuring flow rate.

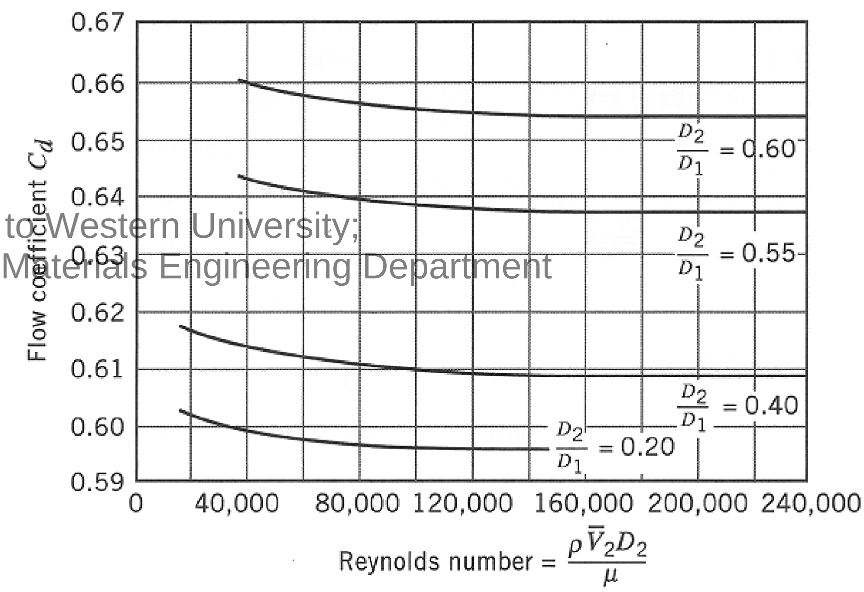


Figure 10-12 Flow coefficients for square-edged orifices.

The effect of RPM on the output parameters of the pump

$$Q_n = Q_o \frac{RPM_n}{RPM_o}$$

$$H_{pn} = H_{pn} \left[\frac{RPM_n}{RPM_o} \right]^2$$

$$W_n = W_o \left[\frac{RPM_n}{RPM_o} \right]^3$$

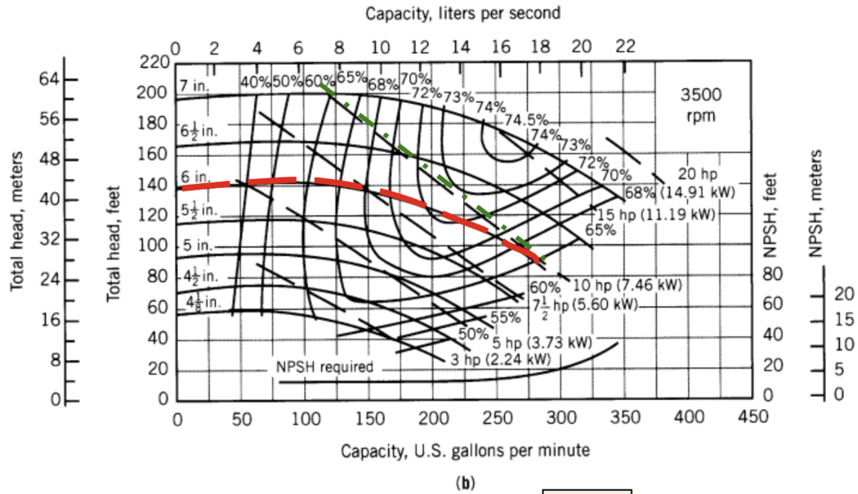


Figure 10-14*b* Centrifugal pump performance data for 3500 rpm.

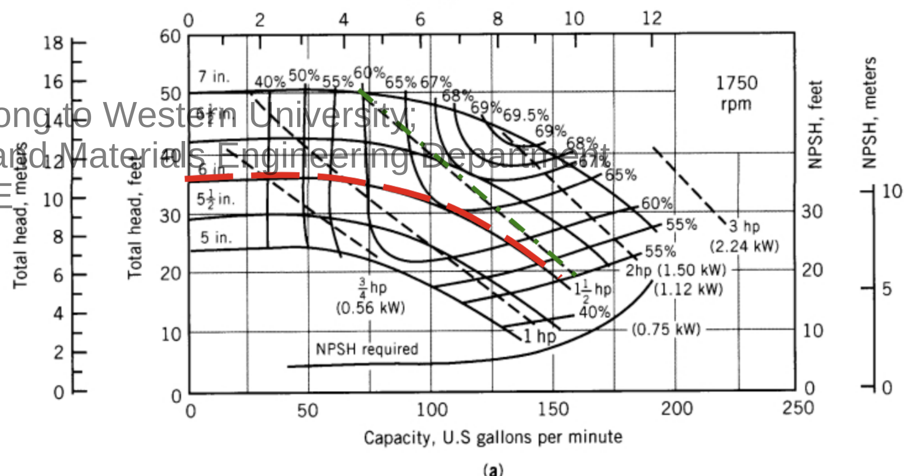
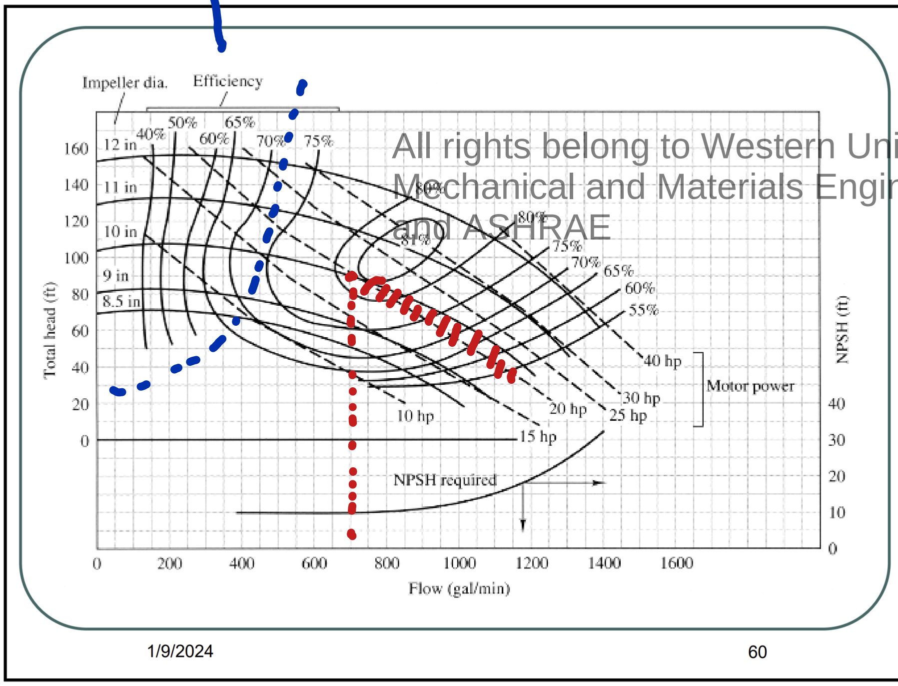
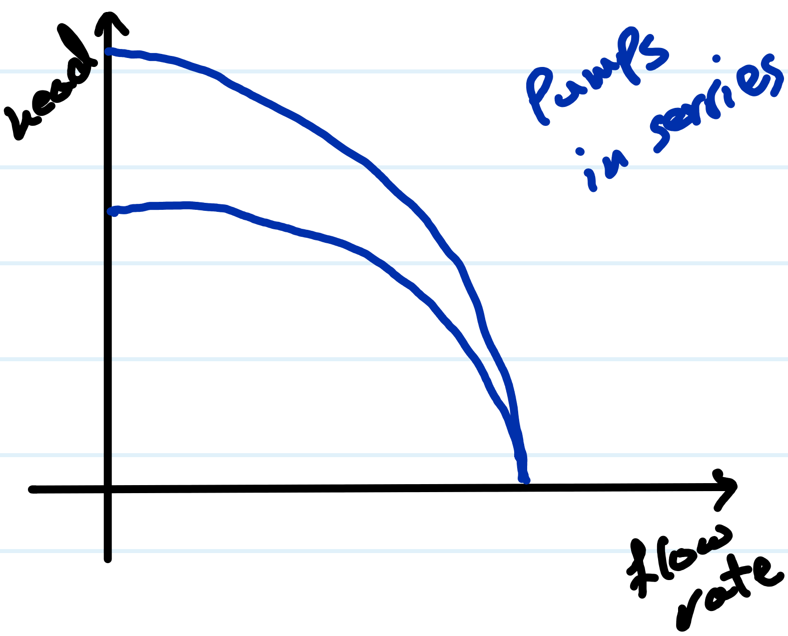


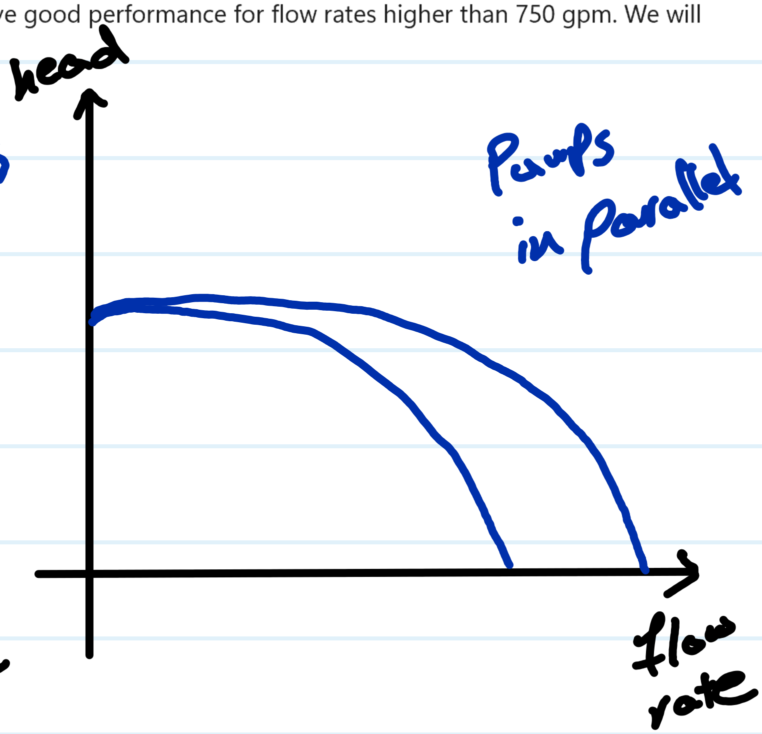
Figure 10-14*a* Centrifugal pump performance data for 1750 rpm.

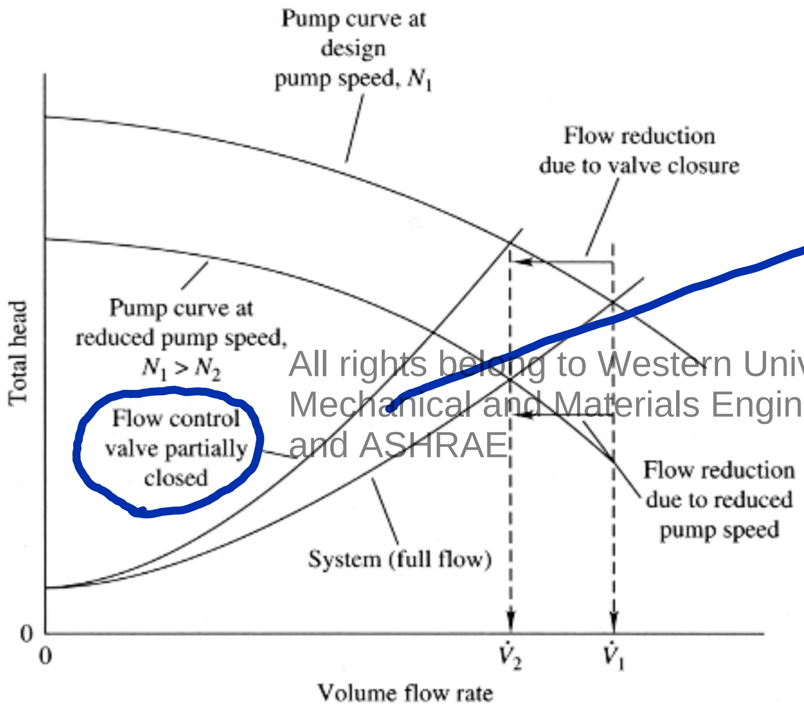
This area is not a proper area to select the pump. Because any small variations in the head will cause to a large variations in the flow rate.



We should choose the line of electric motor a larger size without having any intersection with the pump line. For example for the point selected, 20 hp is not good selection. Because it can not have good performance for flow rates higher than 750 gpm. We will choose 25 hp.







System curve using control valve

To reduce the flow rate of sys →

(1) reduce the Pump's dia. or speed ✓

(2) close the control valve

↳ more ↓ hp → more economical

Cavitation:

NPSHR: minimum required head of pump to avoid Cavitation

NPSHA: minimum available head of system to avoid Cavitation

$\boxed{NPSHA > NPSHR}$

$$NPSHA = \frac{P_s g_c}{\rho g} + \frac{V_s^2}{2g} - \frac{P_v g_c}{\rho g}$$

Where:

- $g_c P_s/\rho g$= Static head at pump inlet, ft or m, absolute
- $V_s^2/2g$ = velocity head at the pump inlet, ft or m
- $g_c P_v/\rho g$= statis pressure head of the liquid at the pumping temperature, ft or m

Water Hammer

$$\Delta p_h = \rho c_s V / g_c \quad (10)$$

where

Δp_h = pressure rise caused by water hammer, lb_f/ft^2

ρ = fluid density, lb_m/ft^3

c_s = velocity of sound in fluid, fps

V = fluid flow velocity, fps

$g_c = 32.2$

The c_s for water is 4720 fps, although the elasticity of the pipe reduces the effective value.

Expansion Tank

Free surface type tank equation:

$$V_T = \frac{V_w\left[\left(\frac{v_2}{v_1}-1\right)-3\alpha\Delta t\right]}{\frac{P_a}{P_1}-\frac{P_a}{P_2}}$$

Where:

- V_T: expansion tank volume ft3 or m3
- V_w: volume of water in the system ft3 or m3
- P_a: local barometric pressure psia or kpa
- P_1: pressure at lower temperature, t_1, psia or kpa
- P_2: pressure at high temperature, t_2 , psia or kpa
- t_1: lower temperature, F or C
- t_2: higher temperature, F or C
- V_1: specific volume of water at t1, ft^3/lbm or m^3/kgm
- V_2: specific volume of water at t1, ft^3/lbm or m^3/kgm
- ά: linear coefficient of thermal expansion of the piping, F^{-1} or C^{-1}: 6.5 x 10^{-6} F^{-1} (11.7 X 10^{-6} C^{-1}) for steel pipe, and 9.3 x 10-5 F-1 (16.74 x 10-6 C-1) for copper pipe
- Δt: higher temperature minus lower temperature F of C $\rightarrow t_2 - t_1$

more Common

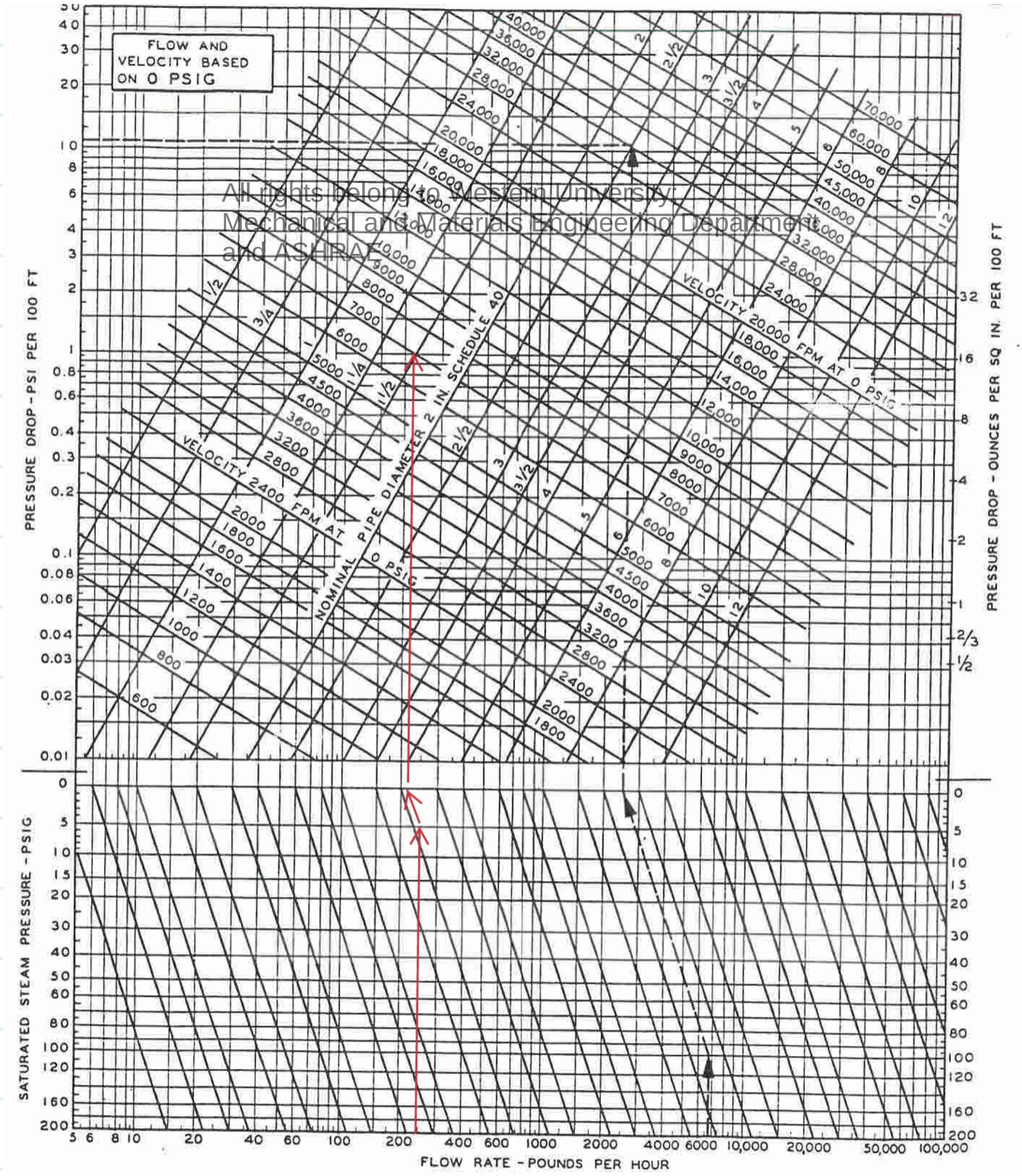
Bladder type tank equation:

$$V_T = \frac{V_w\left[\left(\frac{v_2}{v_1}-1\right)-3\alpha\Delta t\right]}{1-\frac{P_1}{P_2}}$$

- Where:
  - V_T: expansion tank volume ft^3 or m^3
  - V_w: volume of water in the system ft^3 or m^3
  - P_1: pressure at lower temperature, t_1, psia or kpa
  - P_2: pressure at high temperature, t_2 , psia or kpa
  - t_1: lower temperature, F or C
  - t_2: higher temperature, F or C
  - V_1: specific volume of water at t_1, ft^3/lbm or m^3/kgm
  - V_2: specific volume of water at t_1, ft^3/lbm or m^3/kgm
  - ά: linear coefficient of thermal expansion of the piping, F^{-1} or C^{-1}: 6.5 x 10^{-6} F^{-1} (11.7 X 10^{-6} C^{-1}) for steel pipe, and 9.3 x 10-5 F-1 (16.74 x 10-6 C-1) for copper pipe
  - Δt: higher temperature minus lower temperature F of C

Steam Systems

1 lb steam ≃ carries almost 1000 $\frac{Btu}{hr}$



For a certain flow rate and boiler pressure, we will locate and normalize the flow rate for 0 psi/100 ft. Then, for a specific value of pressure drop, we will find the pipe size and the velocity. Then using Fig. 11, we will correct the velocity for the same boiler pressure. min of 1 psi/100ft

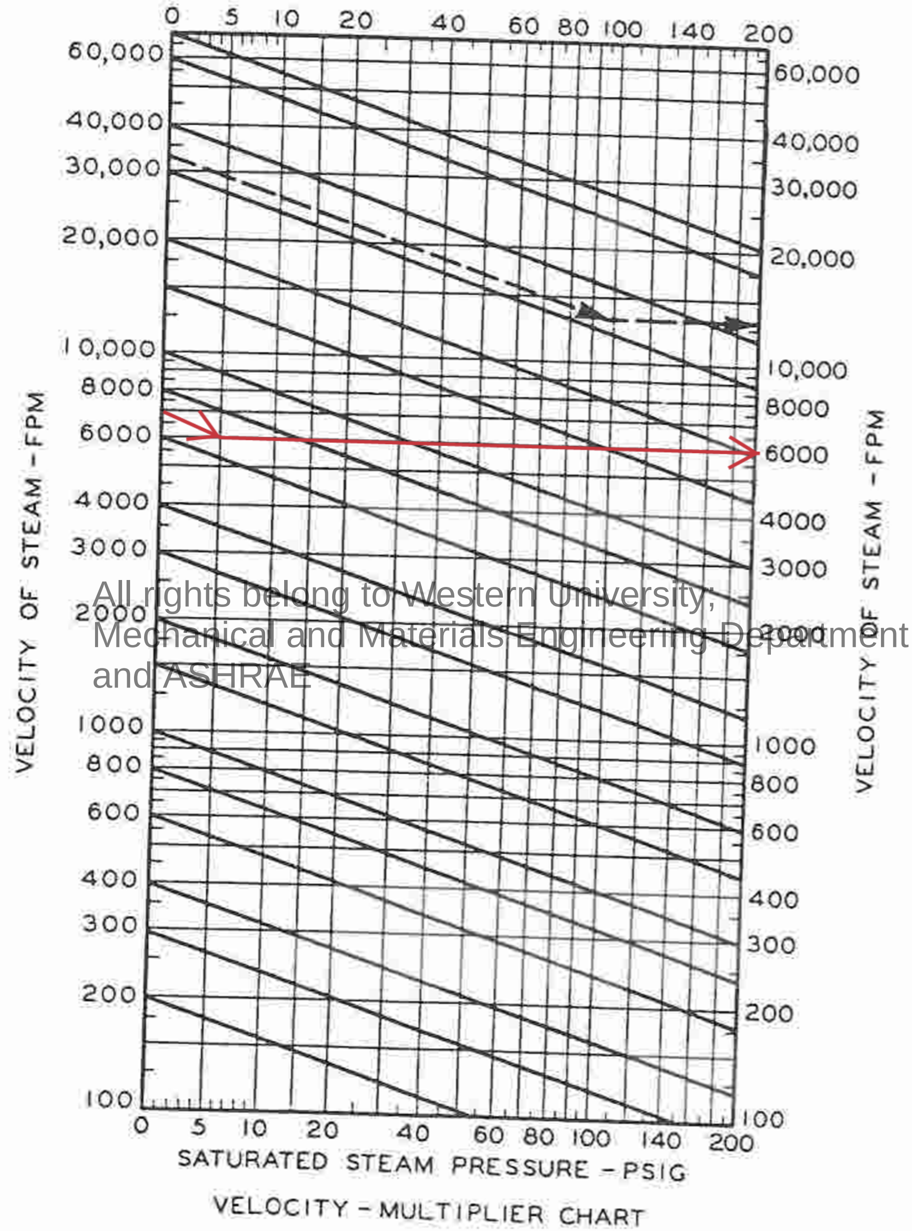


Fig. 11 Velocity Multiplier Chart for Figure 10

Sizing condensate pipe

Water Carrying Capacity Of Pipes

Use Table 14 directly for water. For condensate from steam systems up to 30 psig return line size should be determined on the startup load which is about twice the running load to allow for flash steam and avoid high back pressure. Use frictional resistance of 10 in. Wg. per 100 ft. of travel.

FOR EXAMPLE: A plant has a running load of 1000 lbs. steam per hour so unless we have any specific information to the contrary, the starting load may be assumed to be about 2000 lb./hr.

A quick reference to the 10″ Wg column will show that a 1¼″ pipe will be required.

TABLE 14 WEIGHT OF CONDENSATE, Lb/Hr

Steel Pipe Size	Approximate Frictional Resistance in inches Wg per 100 ft of Travel: 1	5	7	10	14	16	20
½″	100	240	290	350	430	460	520
¾″	230	560	680	820	990	1000	1200
1″	440	1070	1200	1550	1800	2000	2260
1¼″	950	2300	2700	3300	4000	4300	4800
1½″	1400	3500	4200	5000	6100	6600	7100
2″	2800	6800	8100	9900	11800	12700	14200
2½″	5700	13800	16500	20000	23900	25700	28900
3″	9000	21500	25800	31000	37000	39800	44700
4″	18600	44000	52000	63400	75500	81000	90900

for the worst case scenario:

1. the flow rate of condensate will be twice the flow rate of steam.

example: 1000 lbs/hr steam → 2000 lbs/hr

10 in WG/100 ft = 0.833 ft/100 ft.

for the slope of 1%. (10/100 ft) → pipe size: 1¼″

Gas system

TABLE 8.3.2.1

Maximum Capacity of Schedule 40 Pipe Including a Reasonable Number of Fittings In Cubic Feet of Gas Per Hour for a Gas Supply Pressure of Less than 7 inches Water Column and a Maximum Loss in Pressure of 0.5-inch Water Column (Based on a 0.60 Relative Density Gas)

gas pressure at boiler

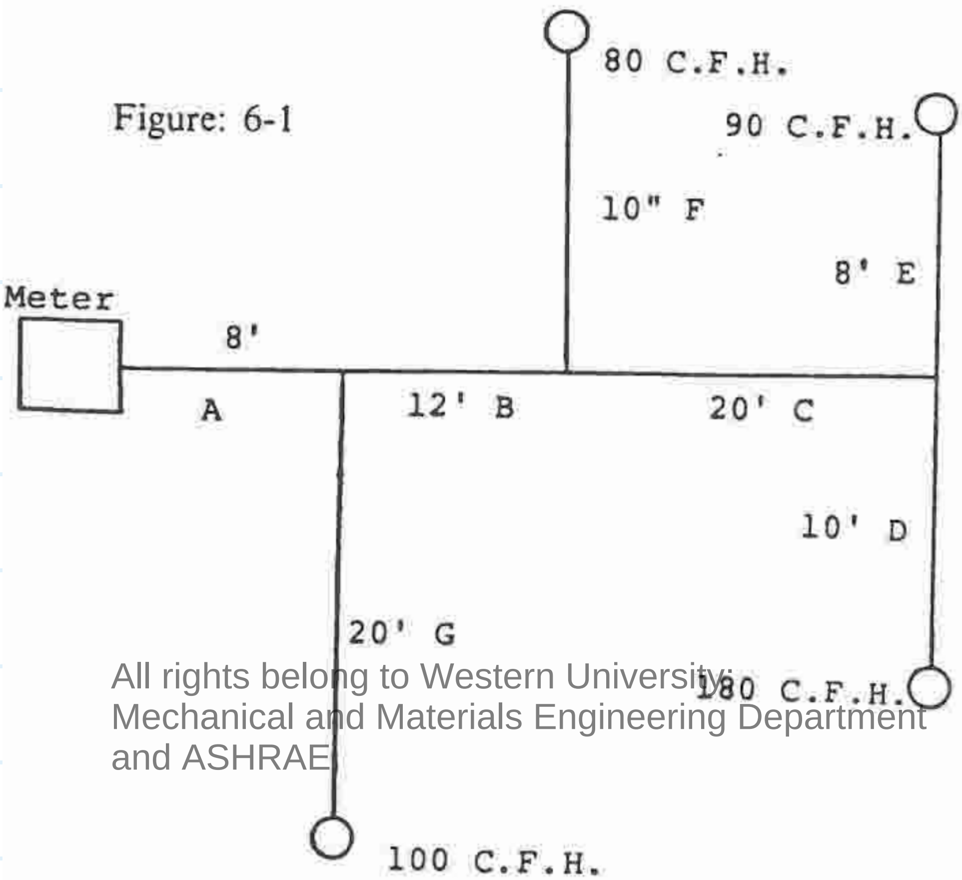
Length of Pipe (Feet)	Nominal Pipe Size Inches											
	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	4	5	6	8
10	120	271	545	1,201	1,862	3,766	6,165	10,500	22,020	38,690	61,240	124,700
20	85	192	385	848	1,316	2,663	4,358	7,426	15,580	27,360	43,310	88,210
30	70	156	315	693	1,074	2,174	3,559	6,063	12,720	22,330	35,360	72,020
40	59	136	272	600	931	1,884	3,082	5,250	11,010	19,340	30,620	62,370
50	54	121	244	537	833	1,685	2,756	4,697	9,852	17,300	27,390	55,780
60	49	111	222	489	759	1,538	2,516	4,287	8,993	15,790	25,000	50,930
70	45	102	205	453	704	1,424	2,330	3,969	8,326	14,630	23,150	47,150
80	43	96	192	425	658	1,331	2,179	3,713	7,789	13,670	21,650	44,100
90	40	90	182	400	620	1,255	2,055	3,500	7,343	12,900	20,410	41,580
100	37	86	172	380	589	1,192	1,949	3,320	6,966	12,230	19,360	39,450
125	34	76	154	340	527	1,065	1,743	2,971	6,230	10,940	17,330	35,280
150	31	70	141	310	480	972	1,592	2,711	5,688	9,987	15,810	32,200
175		65	130	287	445	900	1,473	2,510	5,266	9,247	14,640	29,820
200		61	121	269	416	842	1,379	2,348	4,925	8,652	13,700	27,900
250		54	108	240	372	753	1,233	2,100	4,406	7,738	12,250	24,950
300		49	99	219	340	688	1,126	1,917	4,021	7,063	11,180	22,770
350		46	92	203	315	636	1,042	1,775	3,723	6,539	10,350	21,080
400		43	86	190	294	595	975	1,660	3,483	6,117	9,684	19,730
450		40	81	179	278	562	919	1,566	3,284	5,768	9,131	18,600
500		37	77	169	263	533	871	1,485	3,115	5,471	8,661	17,630
550		36	74	161	250	507	831	1,416	2,971	5,217	8,258	16,820
600		35	70	155	240	487	795	1,356	2,844	4,996	7,909	16,110

(cfh) ft^3/hr

$1\ ft^3/hr$ of gas : $1000\ Btu/hr$ energy

PIPING SYSTEM
OPERATING AT LESS THAN ½ PSIG

Figure: 6-1

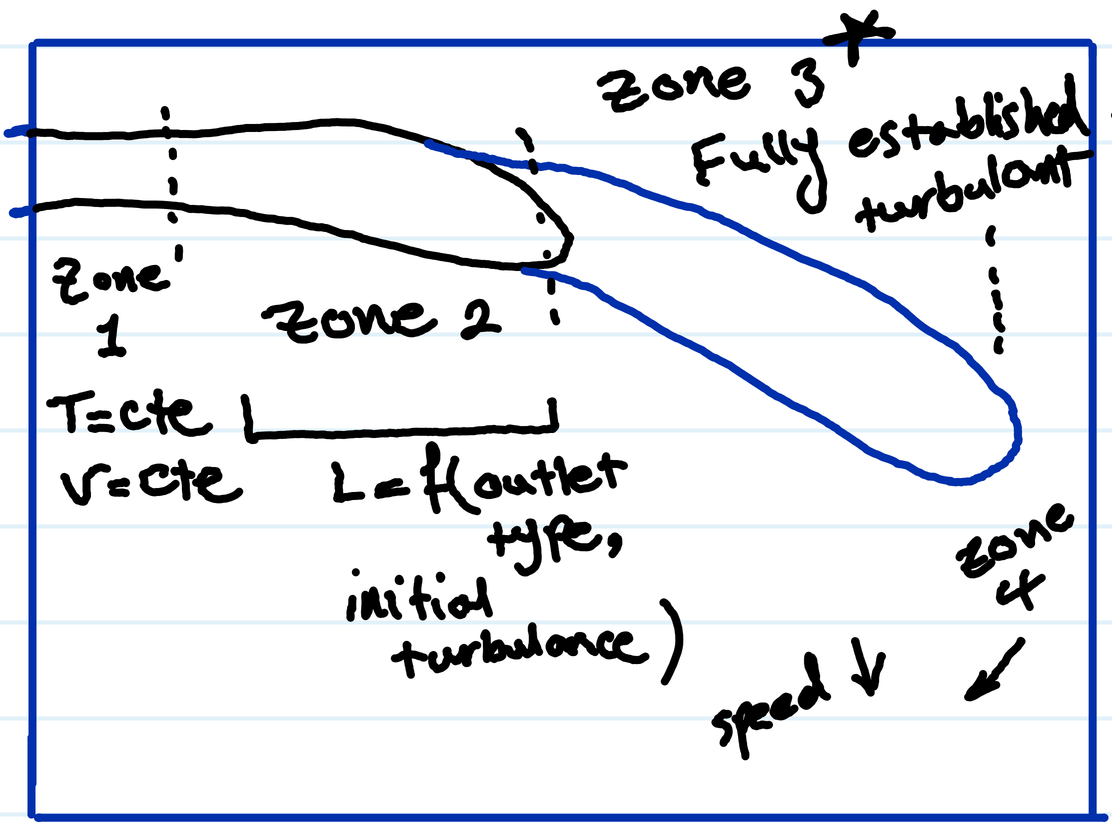


Less than ½ PSIG System

✓ 1. Total volume of gas in C.F.H. is 100 + 80 + 90 + 180 = 450 C.F.H.
✓ 2. Working pressure is less than ½ psig.
✓ 3. Longest run of pipe is 8′ + 12′ + 20′ + 10′ = 50′
✓ 4. Pipe size necessary is:

Pipe	Load	Pipe Size
A	450 C.F.H.	1¼″
B	350 C.F.H.	1¼″
C	270 C.F.H.	1¼″
D	180 C.F.H.	1″
E	90 C.F.H.	¾″
F	80 C.F.H.	¾″
G	100 C.F.H.	¾″

Space Air Diffusion



→ in the occupied zone

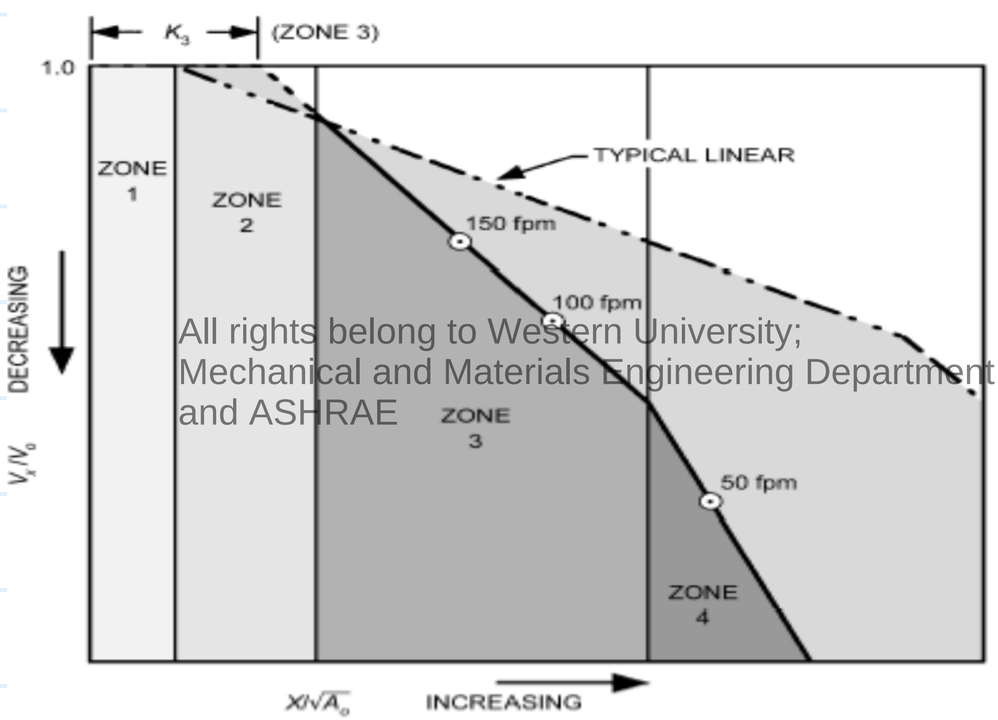
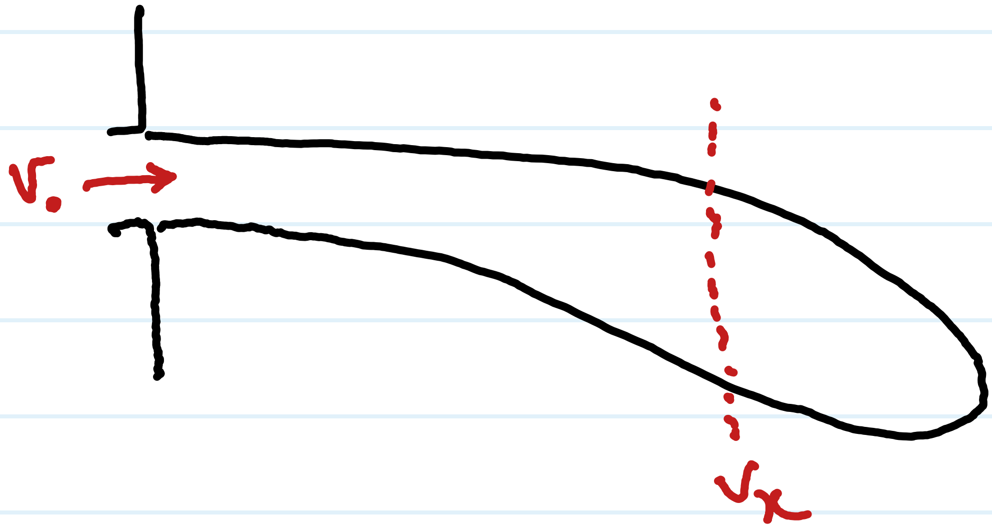


Fig. 2 Chart for Determining Centerline Velocities of Axial and Radial Jets



- The relation between the center-line velocity and the initial velocity (zone 3) is given by:

$$\frac{\overline{V}_x}{\overline{V}_0} = K\frac{D_0}{x} = 1.13K\frac{\sqrt{A_0}}{x}$$

throw

$$\overline{V}_x = \frac{1.13K\,Q_0}{x\sqrt{A_0}} \quad \text{or} \quad V_x = \frac{1.13KQ}{X\sqrt{A_c C_d R_{fa}}}$$

Where:

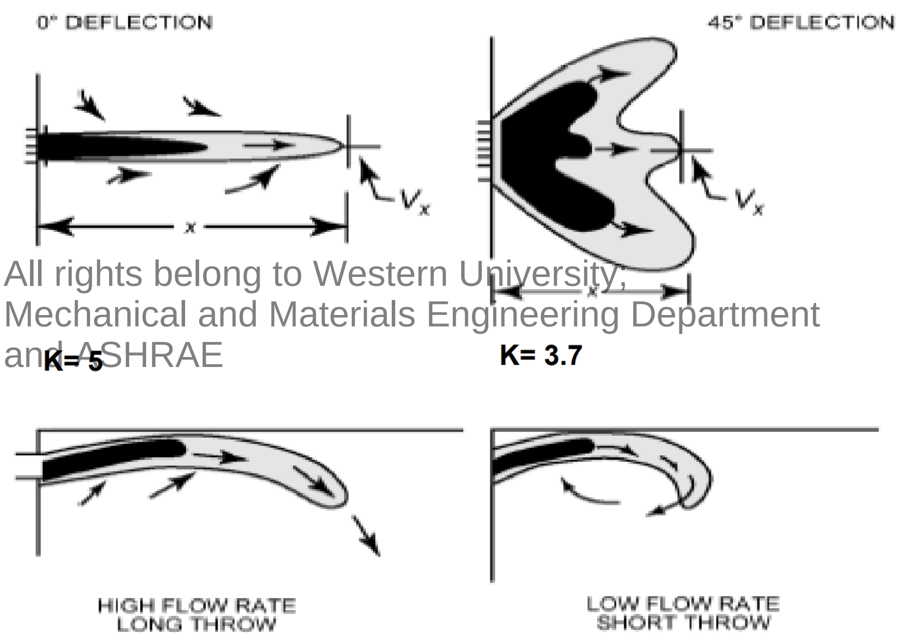
$\boldsymbol{V_x}$ = centerline velocity at distance X from outlet, fpm
$\boldsymbol{V_o}$ = V_c/C_dR_{fa} = average initial velocity at discharge from open-ended duct or across contracted stream at vena contracta of orifice or multiple-opening outlet, fpm
$\boldsymbol{V_c}$ = nominal velocity of discharge based on core area, fpm
$\boldsymbol{C_d}$ = discharge coefficient (usually between 0.65 and 0.90)
$\boldsymbol{R_{fa}}$ = ratio of free area to gross (core) area
$\boldsymbol{Q_o}$ = air-flow rate at outlet, cfm

There are tables specificaly provided for each diffuser type. Throw & Noise conditions are provided as well.

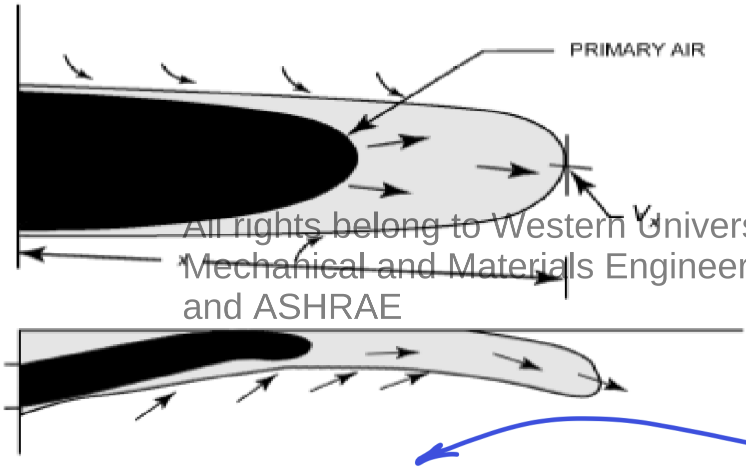
Table 1 Recommended Values for Centerline Velocity Constant *K* for Commerical Supply Outlets

Outlet Type	Discharge Pattern	Area *A*	*K*[a]
High sidewall grilles (Figure 1A)	0° deflection [b]	Core	5.0
	Wide deflection	Core	3.7
High sidewall linear (Figure 1B)	Core less than 4 in. high [c]	Core	3.9
	Core more than 4 in. high	Core	4.4
Low sidewall (Figure 1C)	Up and on wall, no spread	Core	4.4
	Wide spread [c]	Core	2.6
Baseboard (Figure 1C)	Up and on wall, no spread	Duct	3.9
	Wide spread [e]	Duct	1.8
Floor (Figure 1C)	No spread [c]	Core	4.1
	Wide spread	Core	1.4
Ceiling circular directional (Fig. 1D)	360° horizontal [d]	Duct	1.0
	Four-way—little spread	Duct	3.3
Ceiling linear (Figure 1E)	One-way—horizontal along ceiling [c]	Core	4.8

[a] These values are representative for commercial outlets with discharge patterns as shown in Figure 1.
[b] Free area is about 80% of core area.
[c] Free area is about 50% of core area.
[d] Cone free area is greater than duct area.
[e] Face free area is greater than duct area.



A. HIGH SIDEWALL GRILLES



Core less than 4 in high ➔ K= 3.9

Core more than 4 in high ➔ K= 4.4

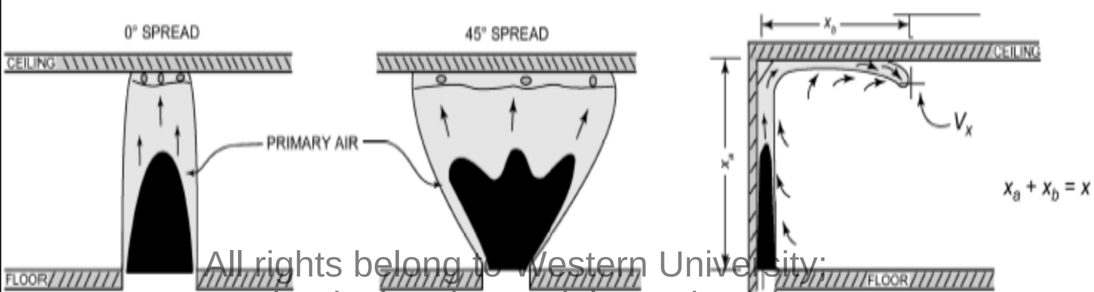
} hight

core length

Note: Throw values based on 4 ft long, isothermal active section. Correction multipliers for other lengths are as follows:

Active length, ft	2	4	8	10
Throw correction	0.72	1	1.5	1.7

B. HIGH SIDEWALL LINEAR

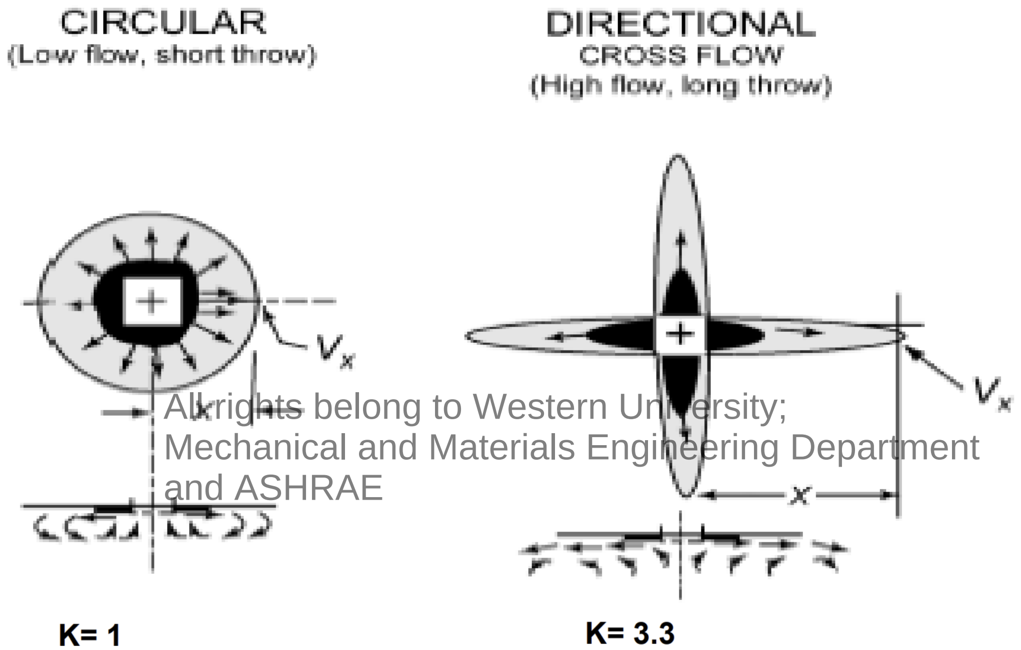


All rights belong to Western University; Mechanical and Materials Engineering Department and ASHRAE

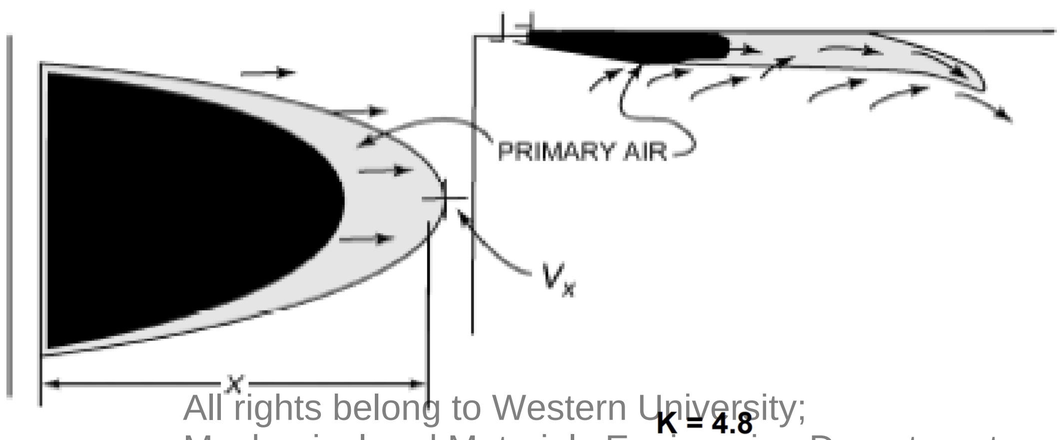
C. LOW SIDEWALL BASEBOARD OR FLOOR WITH VERTICAL DISCHARGE

K = 4.1

K = 1.4



D. CEILING DIFFUSERS



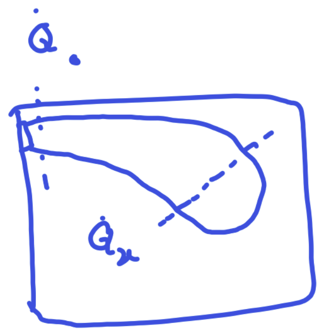
Note: Throw values based on 4 ft long, isothermal active section. Correction multipliers for other lengths are as follows:

Active length, ft	2	4	8	10
Throw correction	0.72	1	1.5	1.7

E. CEILING LINEAR

Behavior of the jets[cont'd]

- The jet expands because of entrainment of room air; the air beyond zone 2 is a mixture of primary air and induced air
- The following equations are used to calculate the induction ratio for two different kind of jet:
  - Circular jet: $$\frac{\dot{Q}_x}{\dot{Q}_0} = 2\frac{\overline{V}_0}{\overline{V}_x}$$
  - Continues slot up to 10 ft in length and by at least 2 ft: $$\frac{\dot{Q}_x}{\dot{Q}_0} = \sqrt{2}\frac{\overline{V}_0}{\overline{V}_x}$$



Room Air Motion

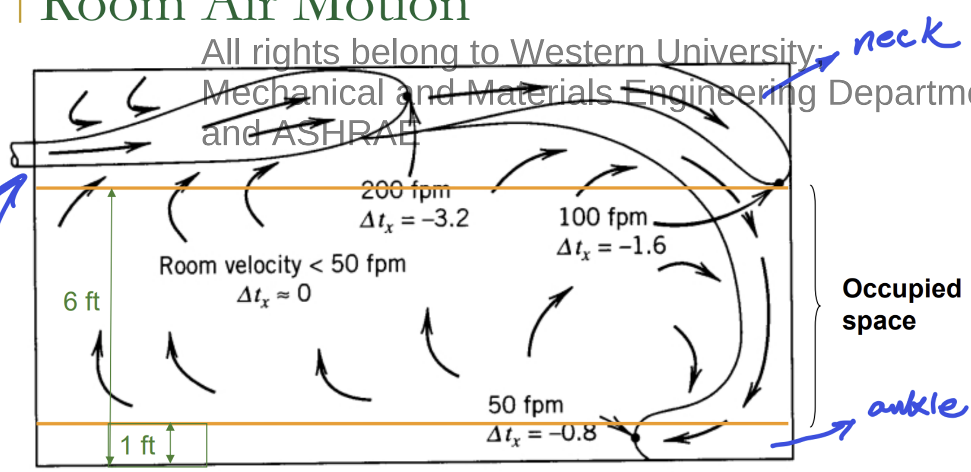


Figure 11-3 Jet and room air velocities and temperatures for $\overline{V}_0 = 1000$ ft/min and $\Delta t_o = -20$ F.

$$\Delta t_x = 0.8\Delta t_0 \frac{\overline{V}_x}{\overline{V}_0}$$

$t_{jet} - t_{room}$

t_c

$t_o - t_{x,jet}$

15

50 fpm < V_x < 100 fpm in occupied space

Draft. Koestel and Tuve (1955) and Reinmann et al. (1959) defined draft as any localized feeling of coolness or warmth of any portion of the body caused by both air movement and air temperature, with humidity and radiation considered constant. The warmth or coolness of a draft was measured above or below a controlled room condition of 76°F db at the center of the room, 30 in. above the floor, with air moving at about 30 fpm.

temp person feel

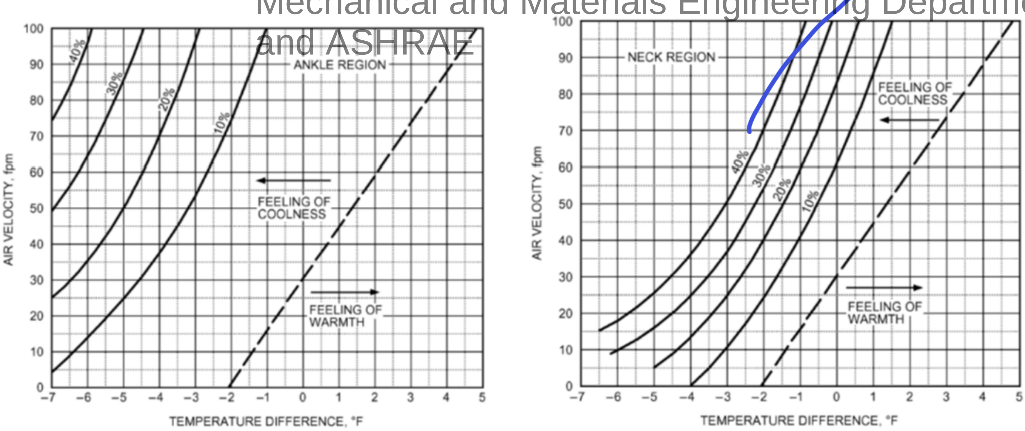
$$\theta = (t_x - t_c) - 0.07(V_x - 30)$$

in occupied zone

where

θ = effective draft temperature, °F
t_x = local airstream dry-bulb temperature, °F
t_c = average (control) room dry-bulb temperature, °F
V_x = local airstream centerline velocity, fpm

How many people do not feel comfortable



$t_x - t_c$