Understanding Ducted Systems

AMCA & O'Dell Associates Education Series | Session 7 | December 16, 2021

Lisa Cherney

Education Manager, AMCA International

Webinar Moderator

- Joined AMCA in February 2019
- Responsible for development of AMCA's education programs; staff liaison for the Education & Training Committee
- Projects include webinars, AMCA's online learning platform programming, presentations at trade shows, PDH/RCEP account management, and AMCA's Speakers Network

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Pat Brooks

Senior Project Manager, SMACNA

- Over 35 years experience in HVAC ductwork design and manufacturing
- Bachelor’s & master's degrees in mechanical engineering, and masters’ degree in business
- Member of ASHRAE and SPIDA technical committees on duct design; recently named ASHRAE Distinguished Lecturer

Understanding Ducted Systems

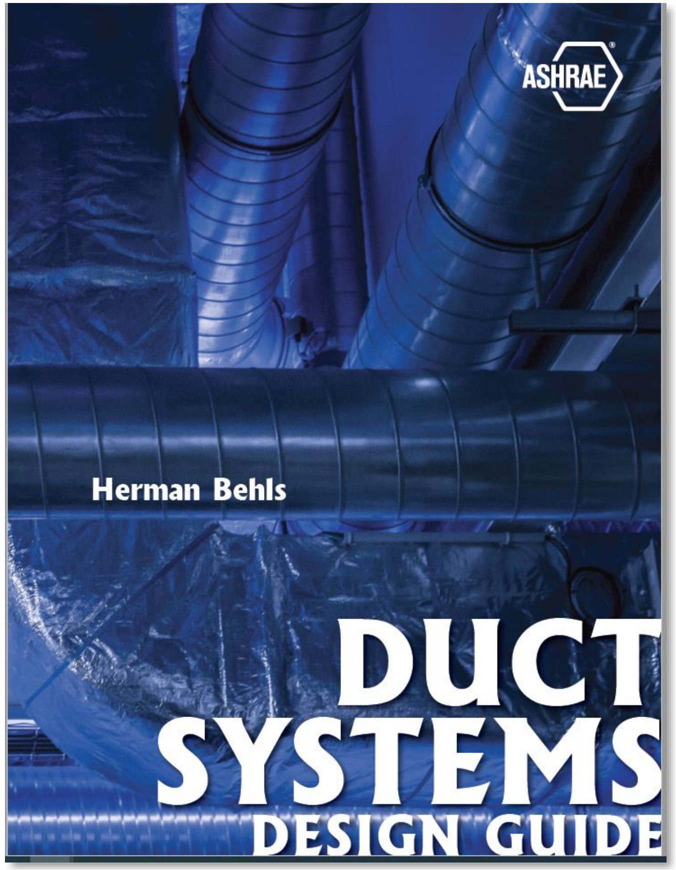
Purpose and Learning Objectives

The purpose of this presentation is to teach the fundamentals of supply duct design & pressure losses.

At the end of this presentation you will be able to:

1. Describe friction and dynamic losses, and fitting selections.
2. Identify the various duct shapes and how to convert duct shapes.
3. Calculate initial duct size.
4. Explain duct system layout, use of loss coefficients & friction calculations.

Duct System Design Guide

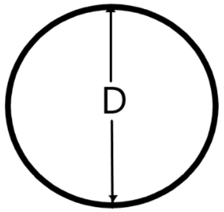


Goals of a High-Performance Air System

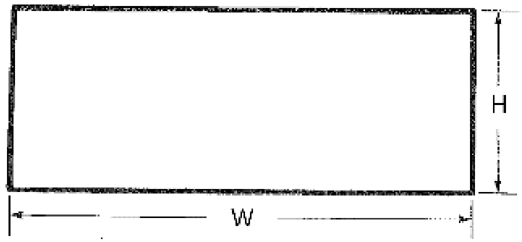
- Design energy efficient HVAC systems that deliver the proper amount of air to specific areas of the building
- Design balanced systems
- Minimize fan energy use
- Minimize first cost
- Minimize the maintenance cost
- Keep noise levels within the required NC/RC levels
- Provide a comprehensive design to the owner per the Owner’s Project Requirements (OPRS)

Cross-sectional Areas

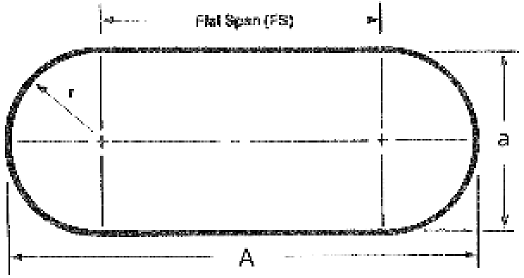
Round: $A_d = \dfrac{\pi D^2}{4}$



Rectangular : $A_d = WH$



Flat Oval: $A_d = \left(\dfrac{\pi a^2}{4}\right) + a\,(A-a)$



Basic Equations

$$\textit{Velocity } V = \frac{Q}{A_d} \rightarrow A_d = \frac{Q}{V}$$

If Q (cfm[L/s]) and A (ft² [m²]) are known, the duct velocity, V (fpm, m/s) can be calculated.

Example 1: If the volume flow rate in a 22 in. (560 mm) duct is, Q = 5000 cfm (2360 L/s), what is the average velocity of air in the duct?

D = 22 inch (1.83 ft) [560 mm]

$$A_d = \frac{\pi(D)^2}{4}$$

$$A_d = \frac{\pi(1.83)^2}{4} = 2.64\ ft^2\ (.25\ m^2)$$

V = 5000 / 2.64 = 1894 fpm [(2360/.25/ 1000) = 9.6 m/s]

Basic Equations

$$\text{Velocity } V = \frac{Q}{A_d} \rightarrow A_d = \frac{Q}{V}$$

Example 2: If the design volume flow rate is 13,000 cfm (6135 L/s) and the velocity is 4000 fpm (20.3 m/s), what is the H dimension in a rectangular duct if the W dimension is 14 inches (355 mm)?

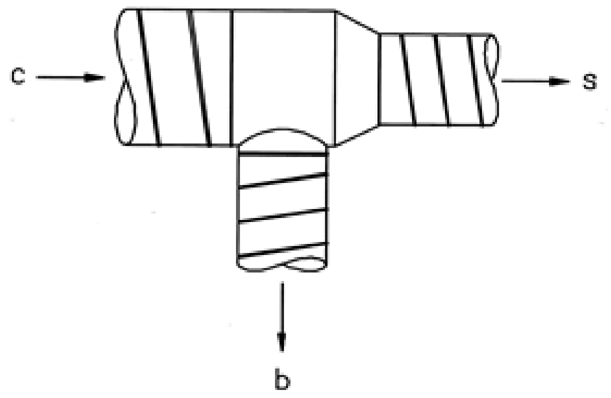
A_d = Q / V = 13,000 / 4000 = 3.25 ft^2 (Multiply by 144 to get in^2) = *468 in^2*
[A_d = Q / V = 6135 / 20.3/1000 = 0.30 m^2]

$$A_d = WH \;\; \text{->} \;\; H = A_d \,/\, W$$

H = 468 / 14 = 33.4 inches
[H = .30 x 1000^2/355 = 845 mm]

Basic Equations – Diverging Flow

According to the law of conservation of mass, the volume flow rate before flow divergence is equal to the sum of the flows after divergence.



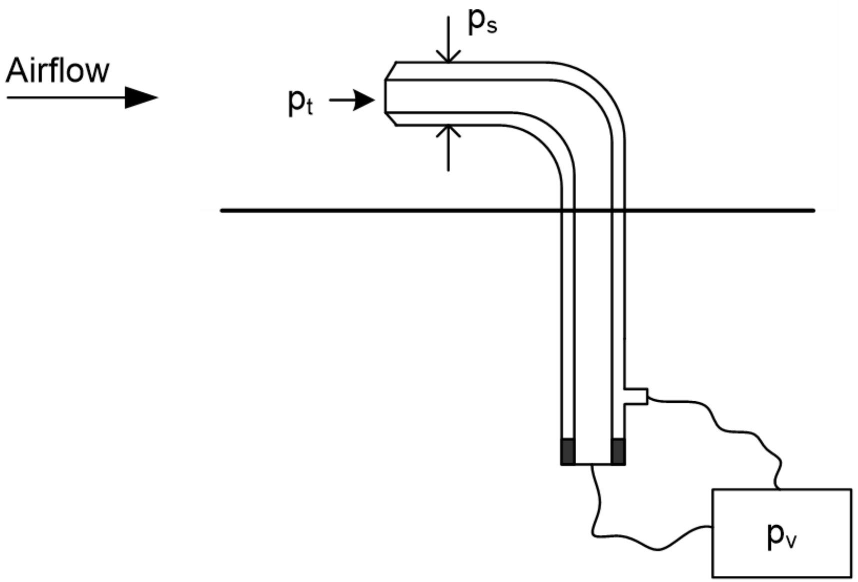
$$Q_c = Q_b + Q_s$$

Where:

Q_c = common (upstream) volume flow rate, cfm (L/s)
Q_b = branch volume flow rate, cfm (L/s)
Q_s = straight-through volume flow rate, cfm (L/s)

Basic Equations - Pressure

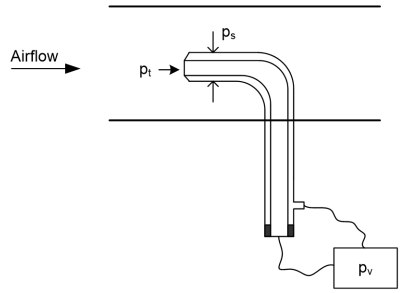
$$p_t = p_s + p_v$$



Pitot-static tube

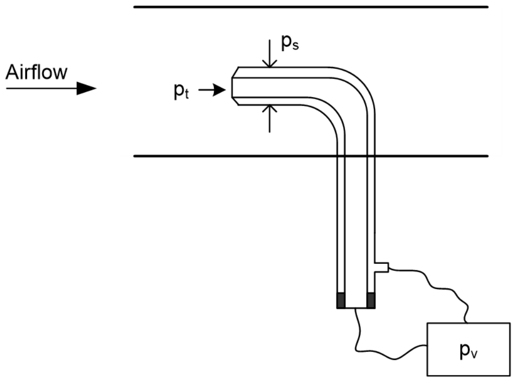
Total Pressure

- Total pressure (pt) represents the total energy of the air flowing in a duct system.
- Energy cannot be created or increased except by adding work or heat (typically at the fan).
- Energy and thus total pressure must always decrease once the air leaves the fan.
- Total pressure losses represent the irreversible conversion of static and kinetic energy to internal energy in the form of heat.
- These losses are classified as either friction losses or dynamic losses.



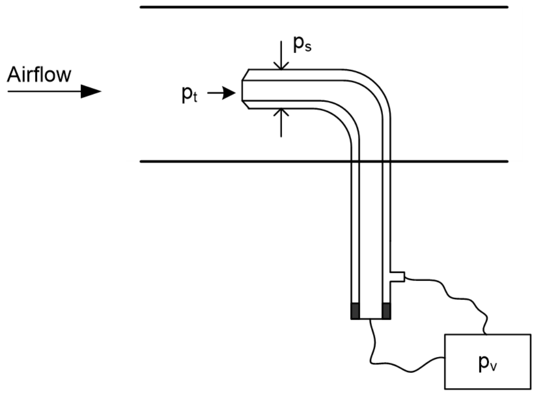
Static Pressure

- Static Pressure is a measure of the static energy of air flowing
- Air which fills a balloon is a good example of static pressure
- Equally exerted in all directions
- The atmospheric pressure of air is a static pressure = 14.696 psi at sea level. One psi ~ 27.7 in. of water, 1 atm~ 407 in. of water [101.325 kPa]
- Static pressure will decrease with an increase of velocity pressure
- Static pressure can increase if there is a decrease in velocity pressure (static regain)



Velocity Pressure

- Velocity pressure (pv) is always a positive number in the direction of flow
- Will increase if duct cross-sectional area decreases
- Will decrease if duct cross-sectional area increases
- When velocity pressure increases, static pressure must decrease
- When velocity pressure decreases, there can be a gain in static pressure, commonly called **STATIC REGAIN**



Velocity Pressure

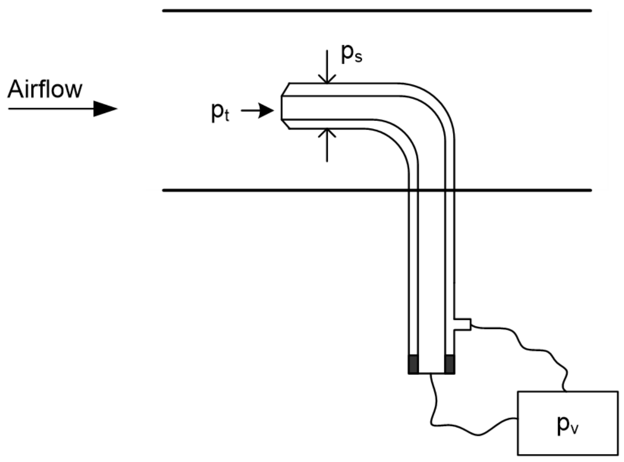
I-P $$p_v = \rho \left(\frac{V}{1097}\right)^2$$

Where:

p_v = velocity pressure, in. of water (Pa)
V = velocity, ft/min (m/s)
ρ = density, lb_m/ft^3 (Kg/m^3)

SI $$p_v = \rho\, V^2/2$$

For standard conditions, $\rho = 0.075\ lb_m/ft^3$ ($1.204\ kg/m^3$)



Pressure – Changes in Pressure

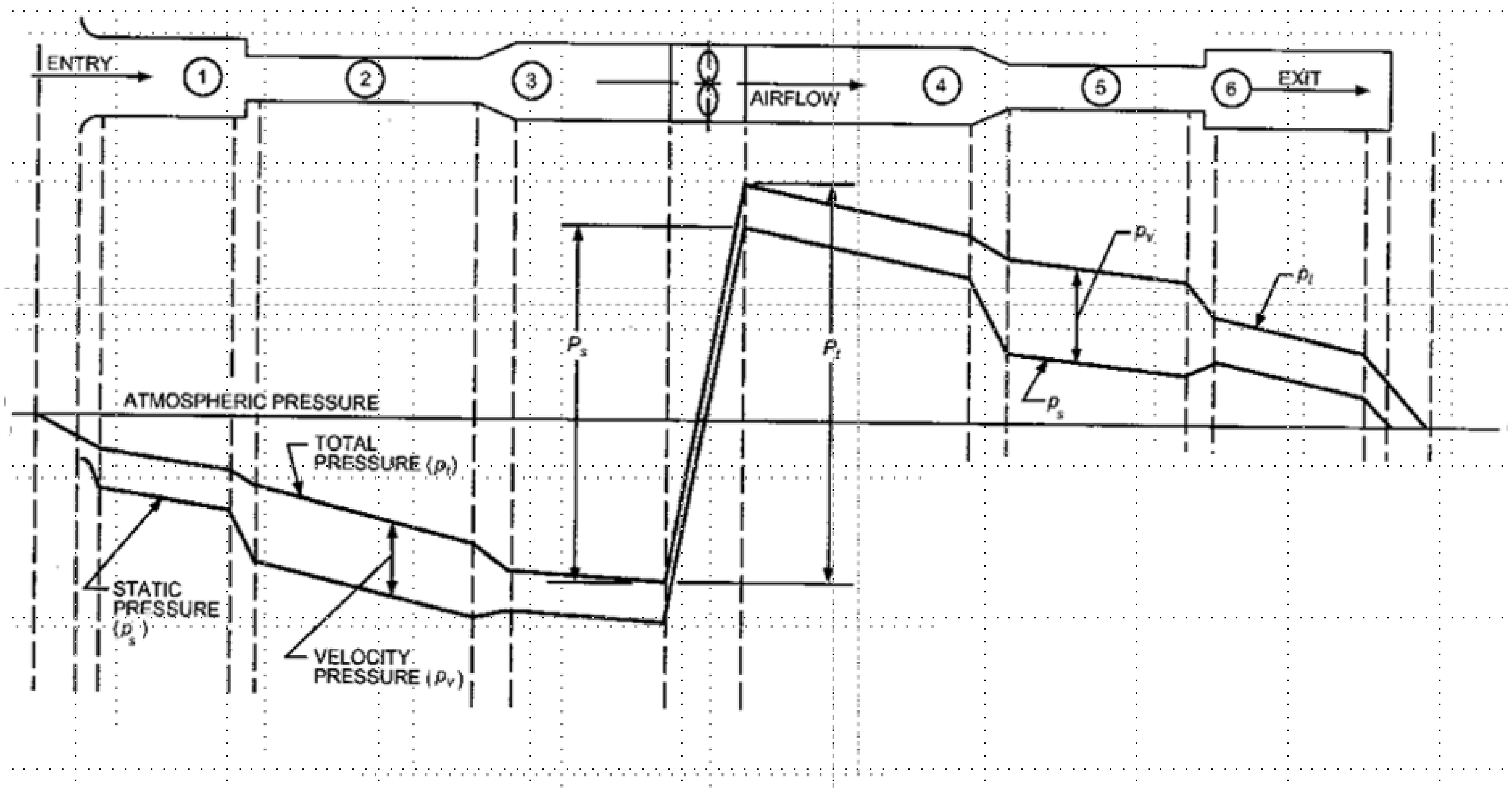
$$\Delta p_t = \Delta p_s + \Delta p_v$$

Derived from the Bernoulli Equation

$$p_{s1} + \frac{\rho_1 V_1^2}{2g_c} + \frac{g}{g_c}\rho_1 z_1 = p_{s2} + \frac{\rho_2 V_2^2}{2g_c} + \frac{g}{g_c}\rho_2 z_2 + \Delta p_{t,1-2}$$

(ASHRAE 2017 Handbook, Chapter 21)

Pressure – Changes in Pressure



Pressure Losses

Friction Losses

Dynamic Losses

Pressure Losses

Darcy-Weisbach Equation (ASHRAE 2017 Handbook, Chapter 21)

$$\Delta p_t = \left(\frac{f\,L}{D_h} p_v\right) + \sum(C) * p_v$$

Where:

f = friction factor

L = Length, ft (m)

D_h = hydraulic diameter, ft (m)

pv = velocity pressure, in. (Pa)

C = loss coefficient

Left hand side is the Darcy Equation for the friction losses.

Right Hand Side is the Weisbach Equation for fittings or other dynamic losses.

The ASHRAE Duct Fitting Database Determines Friction Losses and Fitting Losses and Coefficients and includes over 200 types of fittings.

Pressure Losses

Friction – Colebrook Equation

$$\frac{1}{\sqrt{f}} = -2\,log\left(\frac{\varepsilon}{3.7\,Dh} + \frac{2.51}{Re\sqrt{f}}\right)$$

The Colebrook equation was developed to calculate the friction factor, *f*; requires you to also know the Reynolds Number, Re and the absolute roughness, *ε (ft[mm])*, which is determined experimentally.

Pressure Losses

(From ASHRAE 2021 Handbook)

Table 1 Duct Roughness Factors

1	2	3
	Absolute Roughness ε, ft {mm}	
Duct Type/Material	**Range**	**Roughness Category**
Drawn tubing (Madison and Elliot 1946)	0.0000015 {0.00046}	Smooth 0.0000015 {0.00046}
PVC plastic pipe (Swim 1982)	0.00003 to 0.00015 {0.009 to 0.046}	Medium smooth 0.00015 {0.046}
Commercial steel or wrought iron (Moody 1944)	0.00015 {0.046}	
Aluminum, round, longitudinal seams, crimped slip joints, 3 ft {0.91 m} spacing (Hutchinson 1953)	0.00012 to 0.0002 {0.037 to 0.061}	
Friction chart:		
Galvanized steel, round, longitudinal seams, variable joints (Vanstone, drawband, welded. Primarily beaded coupling), 4 ft {1.22 m} joint spacing (Griggs et al. 1987)	0.00016 to 0.00032 {0.049 to 0.098}	Average 0.0003 {0.09}
Galvanized steel, spiral seams, 10 ft {3.05 m} joint spacing (Jones 1979)	0.0002 to 0.0004 {0.061 to 0.12}	
Galvanized steel, spiral seam with 1, 2, and 3 ribs, beaded couplings, 12 ft {3.66 m} joint spacing (Griggs et al. 1987)	0.00029 to 0.00038 {0.088 to 0.116}	
Galvanized steel, rectangular, various type joints (Vanstone, drawband, welded. Beaded coupling), 4 ft {1.22 m} spacing[a] (Griggs and Khodabakhsh-Sharifabad 1992)	0.00027 to 0.0005 {0.082 to 0.15}	
Phenolic duct, aluminum foil on the interior face, sections connected with a four-bolt flange and cleat joint (Idem and Paruchuri 2018)		
5 ft {1.52 m} spacing:	0.00049 to 0.00128 {0.149 to 0.391}	
10 ft {3.05 m} spacing:	0.00025 to 0.00098 {0.075 to 0.298}	
Wright Friction Chart:		
Galvanized steel, round, longitudinal seams, 2.5 ft {0.76 m} joint spacing, ε = 0.0005 ft {0.15 mm}	Retained for historical purposes [See Wright (1945) for development of friction chart]	
Flexible duct, nonmetallic and wire, fully extended (Abushakra et al. 2004; Culp 2011)	0.0003 to 0.003 {0.09 to 0.9}	Medium rough 0.003 {0.9}
Galvanized steel, spiral, corrugated,[b] Beaded slip couplings, 10 ft {3.05 m} spacing (Kulkarni et al. 2009)	0.0018 to 0.0030 {0.54 to 0.91}	
Fibrous glass duct, rigid (tentative)[c]	—	
Fibrous glass duct liner, air side with facing material (Swim 1978)	0.005 {1.52}	
Fibrous glass duct liner, air side spray coated (Swim 1978)	0.015 {4.57}	Rough 0.01 {3.0}
Flexible duct, metallic corrugated, fully extended	0.004 to 0.007 {1.2 to 2.1}	
Concrete (Moody 1944)	0.001 to 0.01 {0.30 to 3.0}	

[a]Griggs and Khodabakhsh-Sharifabad (1992) showed that ε values for rectangular duct construction combine effects of surface condition, joint spacing, joint type, and duct construction (cross breaks, etc.), and that the ε-value range listed is representative.
[b]Spiral seam spacing was 4.65 in. {119 mm} with two corrugations between seams. Corrugations were 0.75 in. {19 mm} wide by 0.23 in. {6 mm} high (semicircle).
[c]Subject duct classified "tentatively medium rough" because no data available.

Pressure Losses

Dynamic

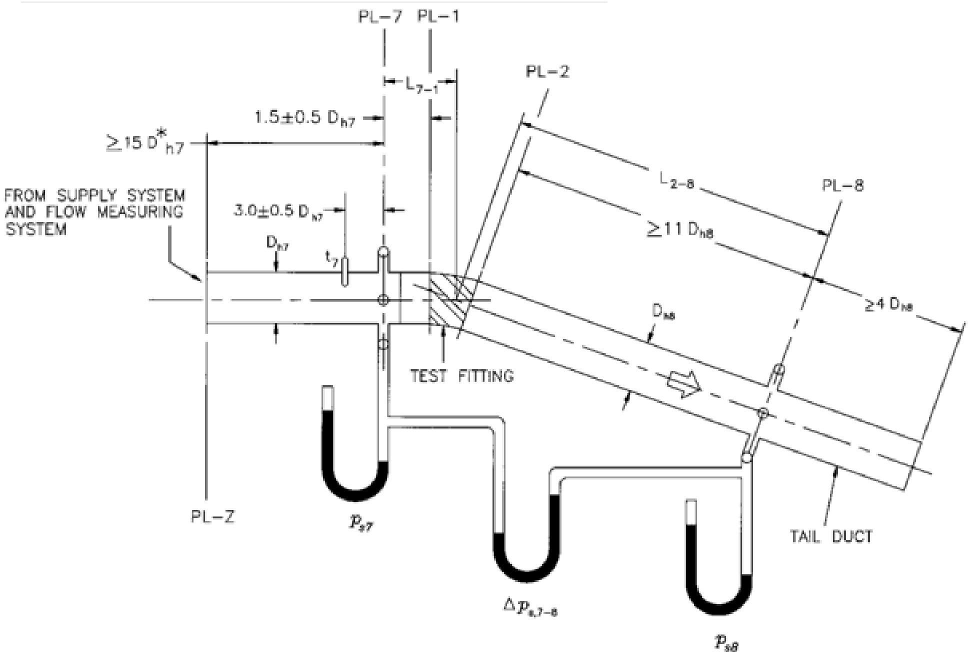
The right-hand side of the Darcy-Weisbach Equation is the Weisbach Equation

$$p_{t,fittings} = \sum (C) * p_v$$

Pressure Losses

Dynamic -How Loss Coefficients are Determined

$$\Delta p_{t,fitting} = C * pv, \quad C = \frac{\Delta p_{t,fitting}}{p_v}$$



$$\Delta p_{t,1\text{-}2} = \Delta p_{s,7\text{-}8} + (p_{v7} - p_{v8}) - (L_{7\text{-}1}\Delta p_{f,7\text{-}1} + L_{2\text{-}8}\Delta p_{f,2\text{-}8})$$

$$C = \frac{\Delta p_{t,1-2}}{p_{v8}}$$

Pressure Losses

Dynamic – Loss Coefficients , ASHRAE Duct Fitting Database

$$\Delta p_{t,fitting} = C * pv_{,} \quad C = \frac{\Delta p_{t,fitting}}{p_v}$$

ASHRAE Duct Fitting Database (DFDB)

- Has 232 Fittings
- Calculates Loss of Round, Rectangular and Flat Oval Duct and Fittings
- Calculates and Takes into Account Density – Can Change Air Properties
- Determines Pressure Loss Base on Input Dimensions and Flow Rates
- Can Look at Complete Fitting Loss Coefficient Table Data, Print it or Export it to Excel
- Can Lookup Fittings in Table View by Filters
- Can create projects of duct/fitting pressure losses
- Results in I-P or SI

Pressure Losses

Example Using ASHRAE Duct Design Database I-P

Friction Loss, 10” Diameter, Airflow is 1000 cfm, L = 100 ft, ε = 0.0003 ft

CD11-1 Straight Duct, Round
(Colebrook 1939)

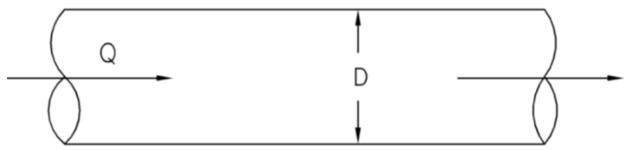
INPUT

Diameter (D)	in.	10
Length (L)	ft	100
Absolute Roughness (ei)	ft	0.0003
Flow Rate (Q)	cfm	1000
Density (RHO)	lbm/ft^3	0.075

Calculate

OUTPUT

Velocity (V)	fpm	1,833
Velocity Presure (Pv)	in. wg	0.21
Reynolds Number (Re)		156,017
Friction Factor (f)		0.0186
Pressure Loss (Po)	in. wg	0.47



Pressure Losses

Example Using ASHRAE Duct Design Database SI

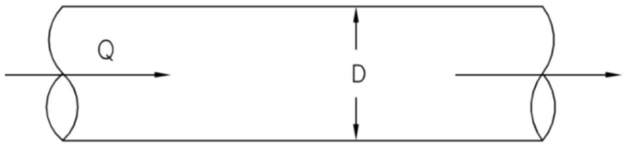
Friction Loss, 254 mm Diameter, Airflow is 472 L/s, L = 30 m ,ε = 0.09 mm

CD11-1 Straight Duct, Round
(Colebrook 1939)

INPUT		
Diameter (D)	mm	254
Length (L)	m	30
Absolute Roughness (ei)	mm	.09
Flow Rate (Q)	L/s	472
Density (RHO)	kg/m^3	1.204

Calculate

OUTPUT		
Velocity (V)	m/s	9.3
Velocity Presure (Pv)	Pa	52
Reynolds Number (Re)		156,719
Friction Factor (f)		0.0185
Pressure Loss (Po)	Pa	114.4



Pressure Losses

Example Using ASHRAE Duct Design Database I-P

Example: 10” Dia, 90° Smooth Radius Elbow, R/D = 1.5. Airflow is 1000 acfm. Elevation is 5000 ft.

CD3-1 Elbow, Die Stamped, 90 Degree, r/D = 1.5
(UMC 1985, Report SRF785)

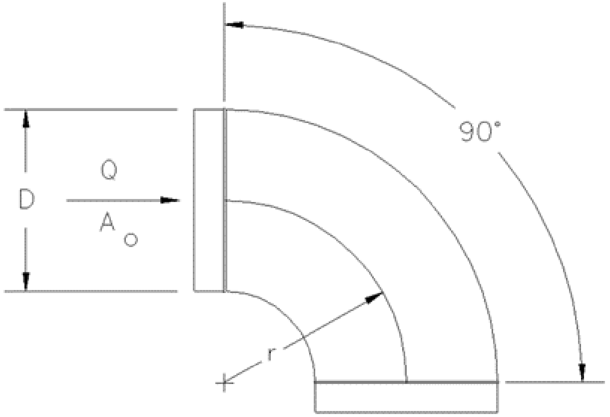
INPUT

Diameter (D)	in.	10
Flow Rate (Q)	cfm	1000
Density (RHO)	lbm/ft^3	0.062

Calculate

OUTPUT

Velocity (Vo)	fpm	1,833
Vel Pres at Vo (Pv)	in. wg	0.17
Loss Coefficient (Co)		0.11
Pressure Loss (Po)	in. wg	0.02



Pressure Losses

Example Using ASHRAE Duct Design Database   S-I

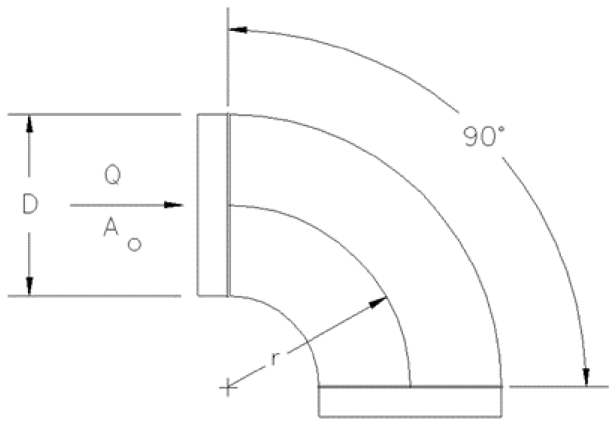
Example: 250 mm Dia, 90° Smooth Radius Elbow, R/D = 1.5. Airflow is 472 L/s. Elevation is 1524 m.

CD3-1 Elbow, Die Stamped, 90 Degree, r/D = 1.5
(UMC 1985, Report SRF785)

INPUT		
Diameter (D)	mm	250
Flow Rate (Q)	L/s	472
Density (RHO)	kg/m^3	0.996

Calculate

OUTPUT		
Velocity (Vo)	m/s	7.6
Vel Pres at Vo (Pv)	Pa	29
Loss Coefficient (Co)		0.11
Pressure Loss (Po)	Pa	3.1



Pressure Losses

Friction Efficiency – Roughness vs Velocity, I-P

Example: 24” x 32” Rectangular Duct, L = 100 ft, Standard Density using ASHRAE DFDB

Using ASHRAE Databse

			Standard Galvanized (ε = 0.0005 ft)	Lined Duct, Corrugated (ε = 0.003 ft)
Velocity (fpm)	Velocity Pressure p_v (inch water)	Q = AV Flow Rate (cfm)	Δp_f Friction Loss (inch water)	Δp_f Friction Loss (inch water)
1000	0.06	5333	0.05	0.06
2000	0.25	10667	0.17	0.23
3000	0.56	16000	0.37	0.52
4000	1	21333	0.65	0.93

Pressure Losses

Friction Efficiency – Roughness vs Velocity SI

Example: 610 mm x 815 Rectangular Duct, L = 30 m, Standard Density

Using ASHRAE Database, SI

			Standard Galvanized (ε = 0.15)	Lined Duct, Corrugated (ε = 0.9 mm)
Velocity (m/s)	Velocity Pressure p_v (Pa)	Q = AV Flow Rate (L/s)	Δp_f Friction Loss (Pa)	Δp_f Friction Loss (Pa)
5.1	13	2516	9.5	12.1
10.1	50	5000	34.5	46.7
15.2	115	7550	76.0	105.4
20.3	204	10070	$\boxed{132.4}$	$\boxed{186.6}$

Pressure Losses

Friction Efficiency – Roughness vs Velocity

Example: 24" x 32" (610 mm x 815 mm) Rectangular Duct, L = 100 ft (30 m), Standard Density using ASHRAE DFDB

Observations:

- ❑ Factor of 13+!! Increase in Pressure Loss when Velocity is Increased by a Factor of 4, From 1000 to 4000 fpm (5 to 20 m/s)
  - ❖ 0.05 in wg (9.5 Pa) increased to 0.65 in wg (132.4)
- ❑ Factor of only 1.2 to 1.4 Increase in Pressure Loss When Roughness (ε) is Increased by a Factor of 10
  - ❖ At 1000 fpm (5 m/s) , 0.05 in wg (9.5 Pa) increased to 0.06 in wg (12.1 Pa)
  - ❖ At 4000 fpm (20 m/s), 0.65 in wg (132.4 Pa) increased to 0.93 in wg (186.6 Pa)

Pressure Losses

Equivalent Round for Rectangular and Flat Oval Duct – Converting Duct Sizes

Rectangular, D_e $$= \frac{1.30(WH)^{0.625}}{(W+H)^{0.250}}$$

Flat Oval, D_e $$= \frac{1.55\left[\frac{\pi}{4}a^2 + a(A-a)\right]^{0.625}}{[\pi a + 2(A-a)]^{0.250}}$$

D_e= Equivalent Round, in (mm)
W= Rectangular Width, in (mm)
H = Rectangular Height, in (mm)
A = Flat Oval Major Dimensions, in (mm)
a = Flat Oval Minor Dimensions, in (mm)

Pressure Losses

Equivalent Round for Rectangular and Flat Oval Duct – Converting Duct Sizes

Example: W= 14 in (355 mm), H = 24 in(600 mm) for Rectangular, a = 14 in (355 mm) and A = 24 in (600 mm) for Flat Oval. What are the Equivalent Round Dimensions

Solution:

Rectangular:

$D_e = 1.3 \times ((14 \times 24)^{0.625}) /(14 + 24)^{0.25} = 19.9$ in (I-P)

$D_e = 1.3 \times ((355 \times 600)^{0.625}) /(355+600)^{0.25} = 500$ mm(SI)

Flat Oval:

$AR = (\pi \times 14^2 /4 + 14 \times (24 - 14)) = 294\ in^2$ (I-P)

$AR = (\pi \times 355^2 /4 + 355 \times (600 - 355)) = 185954\ mm^2$ (SI)

$P = \pi \times 14 + 2 \times (24-14) = 64$ in (I-P)

$P = \pi \times 355 + 2 \times (600-355) = 1605$ mm (SI)

$D_e = 1.55 \times 294^{0.625} / 64^{0.25} = 19$ in (I-P)

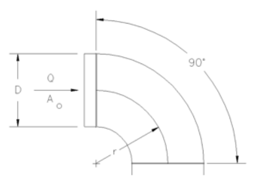
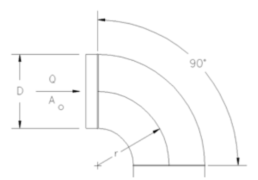
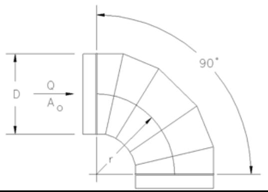
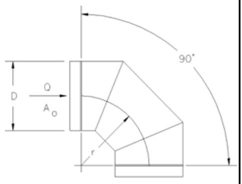
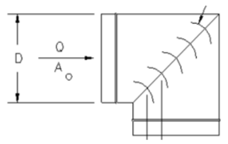
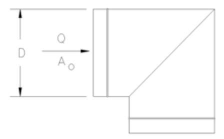
$D_e = 1.55 \times 185955^{0.625} / 1605^{0.25} = 481$ mm (SI)

If the round size is known, then knowing one of the rectangular or flat oval dimensions, the solution must be solved iteratively or use a chart.

Pressure Losses

Fitting Efficiency – Round Elbows, I-P

Example: Diameter = 10 inch, Standard Density using ASHRAE DFDB

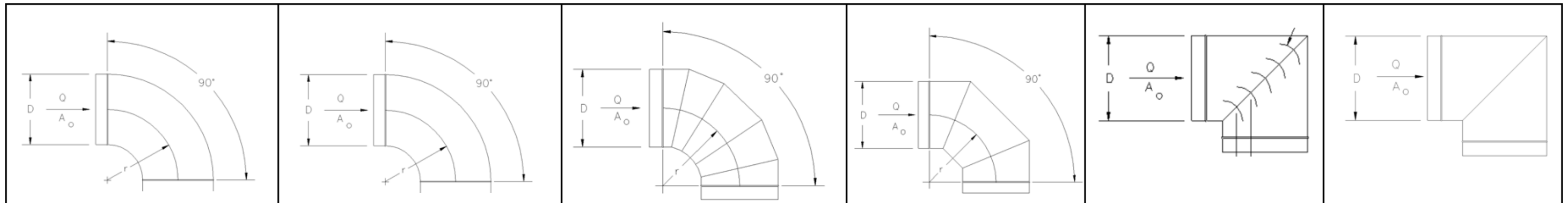
From ASHRAE DFDB														
			Smooth Radius, R/D = 1.5		Smooth Radius, R/D = 1.0		5 Piece, R/D = 1.5		3 Piece, R/D = 1.5 (Table		Mitered w Vanes		Mitered without Vanes	
Velocity (fpm)	Velocity Pressure p_v (inch water)	Q = AV Flow Rate (cfm)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)
1000	0.06	545	0.11	0.01	0.24	0.01	0.20	0.01	0.34	0.02	0.48	0.03	1.19	0.07
2000	0.25	1090	0.11	0.03	0.24	0.06	0.20	0.05	0.34	0.09	0.48	0.12	1.19	0.30
3000	0.56	1635	0.11	0.06	0.24	0.13	0.20	0.11	0.34	0.19	0.48	0.27	1.19	0.67
4000	0.99	2175	0.11	0.11	0.24	0.24	0.20	0.20	0.34	0.34	0.48	0.48	1.19	1.18
			Best		Better		Better		Good		Good		BAD	

Pressure Losses

Fitting Efficiency – Round Elbows, SI

Example: Diameter = 250 mm, Standard Density using ASHRAE DFDB

From ASHRAE DFDB, SI



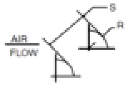
			Smooth Radius, R/D = 1.5		Smooth Radius, R/D = 1.0		5 Piece, R/D = 1.5		3 Piece, R/D = 1.5 (Table		Mitered w Vanes		Mitered without Vanes	
Velocity (m/s)	Velocity Pressure p_v (Pa)	Q = AV Flow Rate (L/s)	Loss Coefficient C	Δp_t (Pa)	Loss Coefficient C	Δp_t (Pa)	Loss Coefficient C	Δp_t (Pa)	Loss Coefficient C	Δp_t (Pa)	Loss Coefficient C	Δp_t (Pa)	Loss Coefficient C	Δp_t (Pa)
5.2	17	257	0.11	1.87	0.24	4.08	0.20	3.40	0.34	5.78	0.48	8.16	1.19	20.23
10.5	66	514	0.11	7.26	0.24	15.84	0.20	13.20	0.34	22.44	0.48	31.68	1.19	78.54
15.7	149	771	0.11	16.39	0.24	35.76	0.20	29.80	0.34	50.66	0.48	71.52	1.19	177.31
20.9	263	1026	0.11	28.93	0.24	63.12	0.20	52.60	0.34	89.42	0.48	126.24	1.19	312.97
			Best		Better		Better		Good		Good		BAD	

Pressure Losses

Fitting Efficiency – Rectangular Mitered Elbows, I-P

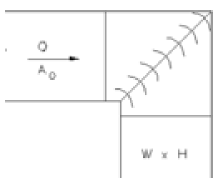
Example: Diameter = 12-inch x 12-inch , Standard Density using ASHRAE DFDB

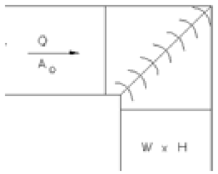
From ASHRAE DFDB

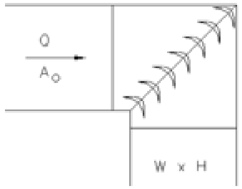


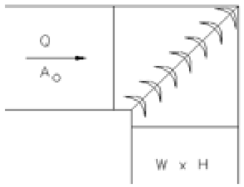
Single Thickness

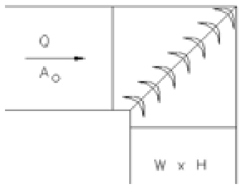
Double Thickness

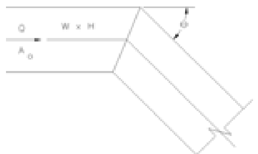












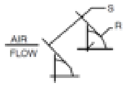
			Single Thickness, Mitered w Turing Vanes, R=2.0, S=1.5		Single Thickness, Mitered w Turing Vanes, R=4.5, S=3.25		Double Thickness, Mitered w Turing Vanes, R=4.5, S=3.25		Double Thickness, Mitered w Turing Vanes, R=2.0, S=1.5		Double Thickness, Mitered w Turing Vanes, R=2.0, S=2.25		Mitered without Vanes, Theta = 90 Deg	
Velocity (fpm)	Velocity Pressure p_v (inch water)	Q = AV Flow Rate (cfm)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)
1000	0.06	1000	0.11	0.01	0.33	0.02	0.41	0.03	0.38	0.02	0.25	0.02	1.18	0.07
1500	0.14	1500	0.11	0.02	0.33	0.05	0.41	0.06	0.38	0.05	0.25	0.04	1.18	0.17
2000	0.25	2000	0.11	0.03	0.33	0.08	0.41	0.10	0.38	0.09	0.25	0.06	1.18	0.29
2500	0.39	2500	0.11	0.04	0.33	0.13	0.41	0.16	0.38	0.15	0.25	0.10	1.18	0.46
			Best								Good		Bad	

Pressure Losses

Fitting Efficiency – Rectangular Mitered Elbows, SI

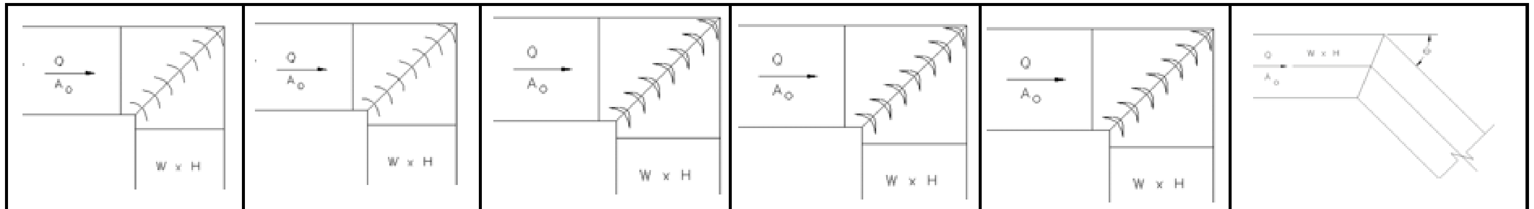
Example: Diameter = 305 mm x 305 mm, Standard Density using ASHRAE DFDB

From ASHRAE DFDB



Single Thickness

Double Thickness



Velocity (m/s)	Velocity Pressure p_v (Pa)	Q = AV Flow Rate (L/s)	Single Thickness, Mitered w Turing Vanes, R=50, S=40: Loss Coefficient C	Single Thickness, Mitered w Turing Vanes, R=50, S=40: Δp_t (Pa)	Single Thickness, Mitered w Turing Vanes, R=110, S=80: Loss Coefficient C	Single Thickness, Mitered w Turing Vanes, R=110, S=80: Δp_t (Pa)	Double Thickness, Mitered w Turing Vanes, R=110, S=80: Loss Coefficient C	Double Thickness, Mitered w Turing Vanes, R=110, S=80: Δp_t (Pa)	Double Thickness, Mitered w Turing Vanes, R=50, S=40: Loss Coefficient C	Double Thickness, Mitered w Turing Vanes, R=50, S=40: Δp_t (Pa)	Double Thickness, Mitered w Turing Vanes, R=50, S=60: Loss Coefficient C	Double Thickness, Mitered w Turing Vanes, R=50, S=60: Δp_t (Pa)	Mitered without Vanes, Theta = 90 Deg: Loss Coefficient C	Mitered without Vanes, Theta = 90 Deg: Δp_t (Pa)
5.1	15.00	472	0.11	1.65	0.33	4.95	0.41	6.15	0.38	5.70	0.25	3.75	1.18	17.70
7.6	35.00	708	0.11	3.85	0.33	11.55	0.41	14.35	0.38	13.30	0.25	8.75	1.18	41.30
10.2	62.00	945	0.11	6.82	0.33	20.46	0.41	25.42	0.38	23.56	0.25	15.50	1.18	73.16
12.79	97.00	1180	0.11	10.67	0.33	32.01	0.41	39.77	0.38	36.86	0.25	24.25	1.18	114.46
			Best								Good		Bad	

Pressure Losses

Fitting Efficiency – Rectangular Mitered Elbows

SMACNA Research ***HVAC Systems Duct Design – Fourth Edition*** Shows:

- vanes with trailing edges have higher loss coefficients than standard construction
- removing every other vane can more than double the pressure loss
- turning vanes are 90°; if used in elbows of other angle the pressure loss will increase

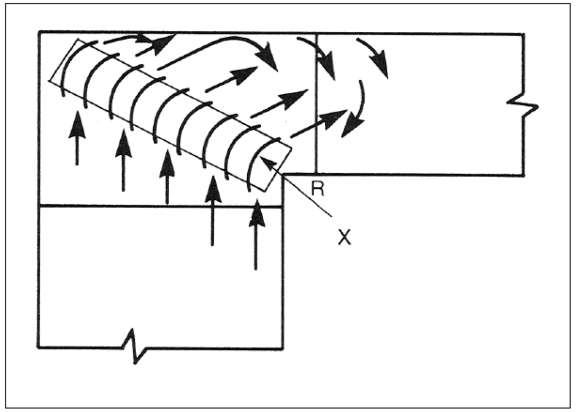


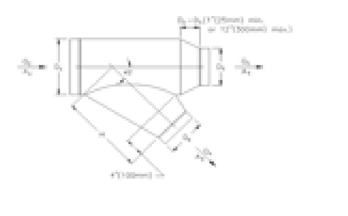
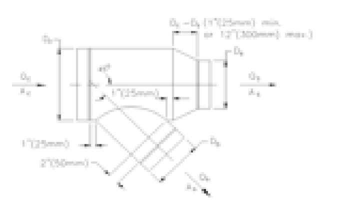
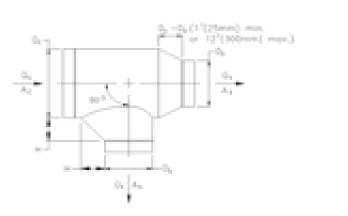
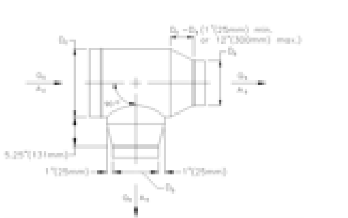
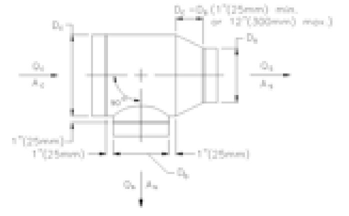
FIGURE 5–14 TURBULENCE CAUSED BY IMPROPER MOUNTING AND USE OF TURNING VANES

Pressure Losses

Fitting Efficiency – Round Taps, I-P

Comparison of Round Diverging Flow Fittings | **Loss Coefficients fromr ASHRAE Duct Fittng Database** | **Standard Conditions** | **ρ = 0.075 lb_m/ft^3**

D_c =	12 inch	A_c =	0.79 ft^2	Q_c =	1000 cfm	V_c =	1273 fpm
D_b =	8.5 inch	A_b =	0.39 ft^2	Q_b =	500 cfm	V_b =	1269 fpm
D_s =	8.5 inch	A_s =	0.39 ft^2	Q_s =	500 cfm	V_s =	1269 fpm

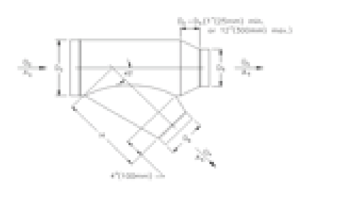
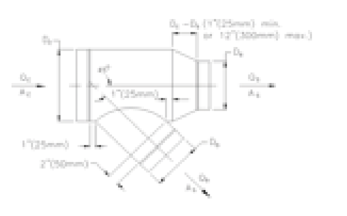
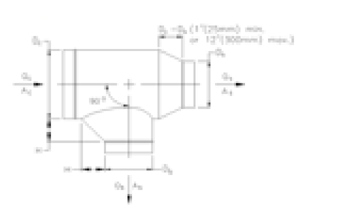
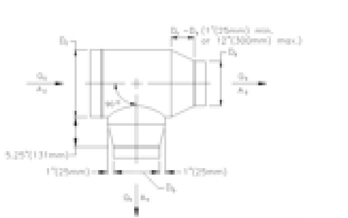
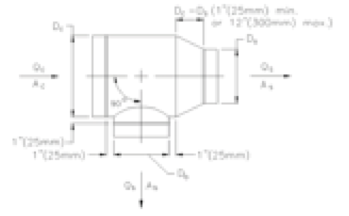






		Wye, 45° Conical		Wye, 45°		45° Entry		Conical Tee		Tee	
Branch Velocity (fpm)	Velocity Pressure p_v (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)	Loss Coefficient C	Δp_t (inch water)
1269	0.10	0.15	0.02	0.38	0.04	0.35	0.04	0.44	0.04	1.21	0.12
		Best		Good		Best		Good		Worst	

Pressure Losses

Fitting Efficiency – Round Taps, SI

Comparison of Round Diverging Flow Fittings			Loss Coefficients fromr ASHRAE Duct Fittng Database		Standard Conditions		ρ = 0.075 lb_m/ft^3
D_c =	305 mm	A_c =	0.07 m^2	Q_c =	472 cfm	V_c =	6.5 m/s
D_b =	215 inch	A_b =	0.04 m^2	Q_b =	235 cfm	V_b =	6.5 m/s
D_r =	215 inch	A_r =	0.04 m^2	Q_r =	235 cfm	V_r =	6.5 m/s

		Wye, 45° Conical		Wye, 45°		45° Entry		Conical Tee		Tee	
											
Velocity (m/s))	Pressure p_v (Pa)	Loss Coefficient C	Δp_t Pa)	Loss Coefficient C	Δp_t Pa)	Loss Coefficient C	Δp_t Pa)	Loss Coefficient C	Δp_t Pa)	Loss Coefficient C	Δp_t Pa)
6.5	25	0.15	3.75	0.38	9.50	0.35	8.75	0.43	10.75	1.21	30.25
		Best		Good		Best		Good		Worst	

Designing the Duct System Overview

- Step 1 - Determine air volume requirements. Include an allowance for leakage.
- Step 2 - Locate duct runs. Avoid unnecessary directional changes.
- Step 3 - Locate balancing dampers if necessary.
- Step 4 - Determine the allowable noise (NC) levels.
- Step 5 - Select design method.
- Step 6 - Select the initial duct size.
- Step 7 - Determine duct sizes based on the design methodology. Use efficient fittings.
- Step 8 - Keep aspect ratios as close to 1 as possible.
- Step 9 - Determine system pressure requirements. Include total pressure losses of components.

Designing the Duct System Overview

- Step 10 - Determine the design leg(s) for both Supply and Return
- Step 11 - Determine the required fan operating pressure
- Step 12 - Analyze the design to improve balancing and reduce material cost.
- Step 13 - Select fan according to proper guidelines for the operating pressure and maximum total volume flow rate
- Step 14 - Analyze the design to make sure it meets the acoustical requirements.
- Step 15 - Select materials that minimize cost and meet SMACNA Duct Construction Standards.
- Step 16 - Analyze the life-cycle cost of the design.
- Step 17 - Commission the design to make sure it meets the Owner's Project Requirements (OPRS)

Pressure Losses – The Design Leg

Critical Path

Critical paths are the duct sections from a fan outlet to the terminal device with the highest total pressure drop for supply systems or from the entrance to the fan inlet with the highest total pressure drop for return or exhaust systems.

Designing the Duct System Overview

- Selecting the Design Method

Design Method	Pros	Cons
Equal Friction	Easier to Use	Does not account for varying lengths, uses same friction loss rate to size 1 ft. length or 100 ft. for example.
	Can Use a Ductulator to Determine Sizes	Fittings don't affect the design only the analysis. The design or size is a function of the friction rate used. Fittings losses must be included in the analysis.
	Good for quickly designing small systems.	The system will not be balanced without additional work or use of dampers.
	Can design return/exhaust or supply systems.	Optimum friction rate is not known. Choosing a friction rate is from experience by rule of thumb.

Designing the Duct System Overview

- Selecting the Design Method

Design Method	Pros	Cons
Static Regain	Larger duct sizes may be used, but offset by smaller sizes in non-critical paths	Sizing ducts is cumbersome and may require may interations which are best suited by the use of a computerized design program
	System will be more balanced than equal friction, depending on the available duct sizes allowed	
	Often can use smaller sizes or less efficient and lower cost tyes of fittings in the non-design legs	Can only be used on Supply Systems
		Must choose an initial velocity based on guidelines

Designing the Duct System Overview

Equal Friction Rate Design

- Size all main and branch duct at a constant friction rate per 100 ft (1 meter) including the initial section
- Calculate the total pressure loss for each section, both supply and return ductwork including all fitting losses
- If by hand, a spreadsheet will be helpful
- For each main and branch of a junction be sure to account for the straight-through and branch loss coefficients
- Tabulate the total pressure required for each path from the fan to the supply terminal (and return grill for return systems)
- Determine the critical path and maximum operating pressure
- Determine the excess pressure for each non-critical paths

Designing the Duct System Overview

Equal Friction Rate Design

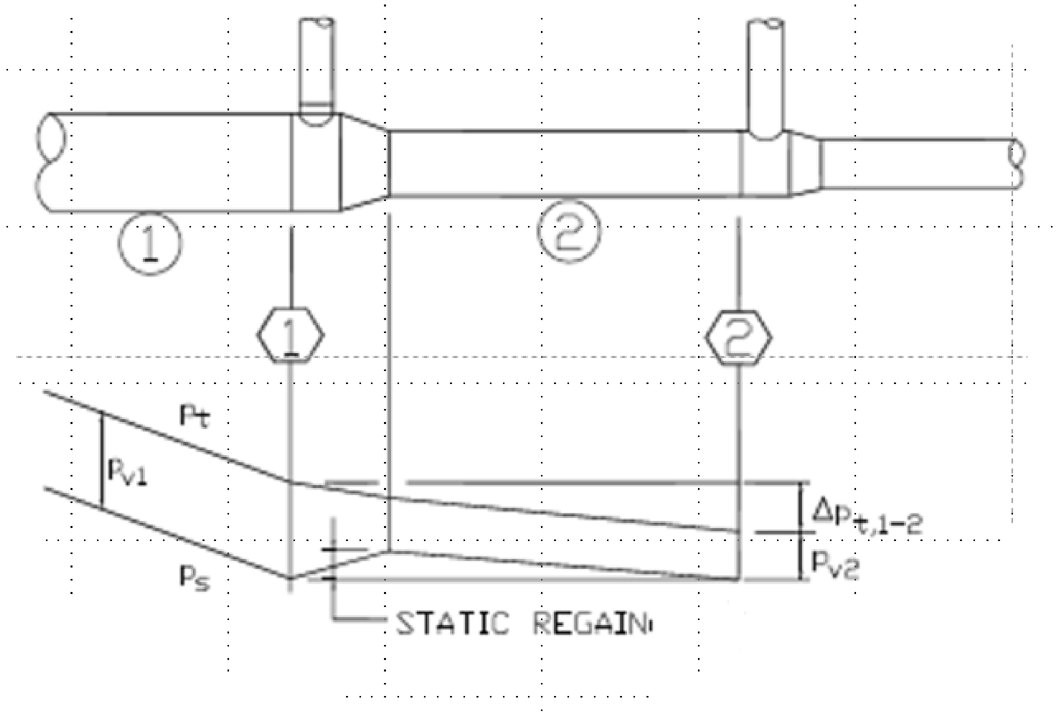
- For Total Pressure Design, balance the system with smaller duct sizes then less efficient fittings in non-design legs. (This is usually not done with equal friction designs) calculate the excess total pressure for the path to each terminal
- If excess pressure is greater than 0.1 in. of water (25 Pa), consider using a higher friction rate in non-design legs with smaller duct sizes in that area
- Perform an acoustical analysis of the system. Provide lined duct or silencers as necessary

Designing the Duct System Overview

Static Regain Design

- Determine the initial size . Use the same size as you would for the Equal Friction Rate or use the Maximum Recommended Duct Airflow Velocities to Achieve Specified Acoustic Design Criteria table
- Size the straight-through and branch sections using
- $\Delta p_{s,1\text{-}2} = p_{v1} - p_{v2} - \Delta p_{t,1\text{-}2} = 0$. Use the junction upstream velocity to determine p_{v1}. Use efficient fittings
- Size ductwork downstream of VAV terminal units by the equal friction method
- Tabulate the total pressure required for each path from the fan to the supply terminal

Duct Design –Static Regain



$$\Delta p_{t,1\text{-}2} = \Delta p_{s,1\text{-}2} + \Delta p_{v,1\text{-}2}$$

$$\Delta p_{s,1\text{-}2} = \Delta p_{t,1\text{-}2} - \Delta p_{v,1\text{-}2}$$

$$\Delta p_{s,1\text{-}2} = 0$$

$$\Delta p_{t,1\text{-}2} = \Delta p_{v,1\text{-}2}$$

Satisfid When:

$$\Delta p_{v,1\text{-}2} - \Delta p_{t,1\text{-}2} - = 0$$

or

$$p_{v1} - p_{v2} - \Delta p_{t,1\text{-}2} - = 0$$

Designing the Duct System

Static Regain Design

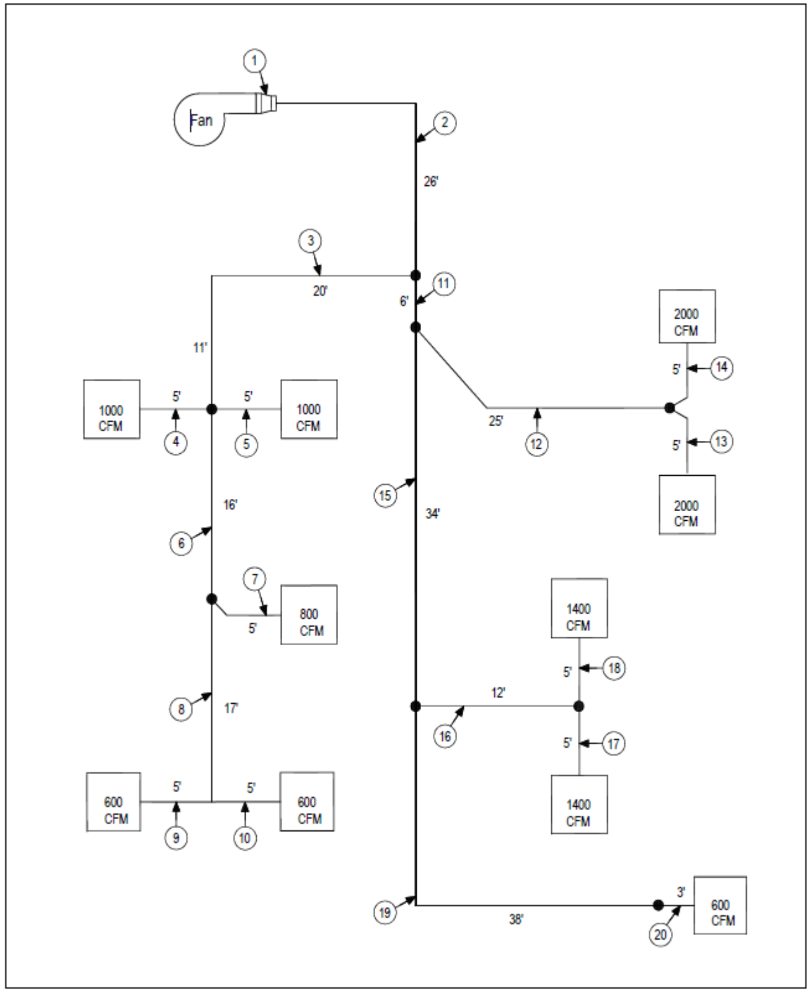
- Determine the critical path and maximum operating pressure
- Determine the excess pressure for each non-critical paths
- Design should be reasonably in balance, but for Total Pressure Design, adjust appropriate branches using smaller duct sizes
- If there is still excess pressure, consider using some less efficient fittings
- Unbalance of 0.1 in. of water (25 Pa) is acceptable (well within the accuracy of the fitting loss coefficients)
- Perform an acoustical analysis of the system. Provide lined duct or sound attenuators where necessary

Designing the Duct System

Example

Size the system shown using Equal Friction and Static Regain. The design air temperature is 69 °F (21°C), located in Denver. Density (ρ) is 0.061 lb_m/ft^3 (0.983 kg/m^3), zero duct air leakage, Ducts are round spiral galvanized steel. The diffuser and distribution ductwork downstream of the VAV box has a pressure loss of 0.05 in. of water (12 Pa). The VAV terminal units have loss coefficients according to the following Table.

VAV Terminal Outlets			
Section	Box Size, in (mm)	Airflow, cfm (L/s)	Loss Coefficient (C)
4 & 5	10 (249)	1000 (472)	2.58
7	9 (229)	800 (378)	2.31
9 & 10	8 (206)	600 (283)	2.49
13 & 14	14 (324)	2000 (944)	2.56
17 & 18	12 (661)	1400 (661)	2.65
20	8 (206)	600 (283)	2.49



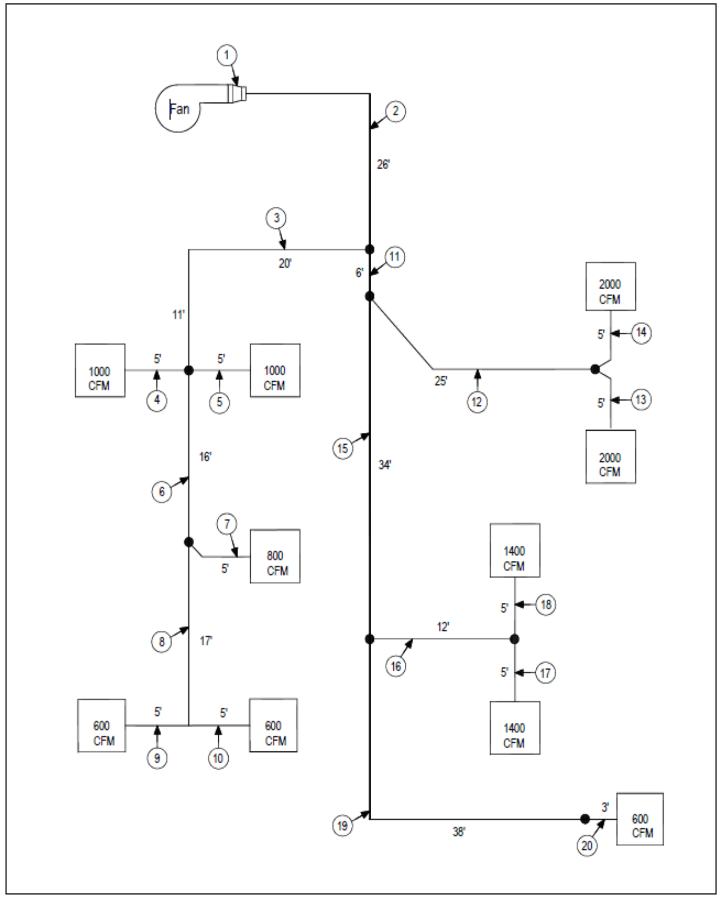
Designing the Duct System

Example

Assume the first section is in a shaft and the RC requirement is 35 maximum.

Maximum Recommended Duct Airflow Velocities to Achieve Specified Acoustic Design Criteria[1]			
Duct Location	RC or NC Rating in Adjacent Occupancy	Maximum Airflow Velocity, fpm (m/s)	
		Rectangular Duct	Round Duct
In shaft or above drywall ceiling	45	3500 (17.8)	5000 (25.4)
	35	2500 (12.7)	3500 (17.8)
	25 or less	1700 (8.6)	2500 (12.7)
Above suspended acoustic ceiling	45	2500 (12.7)	4500 (22.9)
	35	1750 (8.9)	3000 (15.2)
	25 or less	1200 (6.1)	2000 (10.2)
Duct located within occupied space	45	2000 (10.2)	3900 (19.8)
	35	1450 (7.4)	2600 (13.2)
	25 or less	950 (4.8)	1700 (8.6)

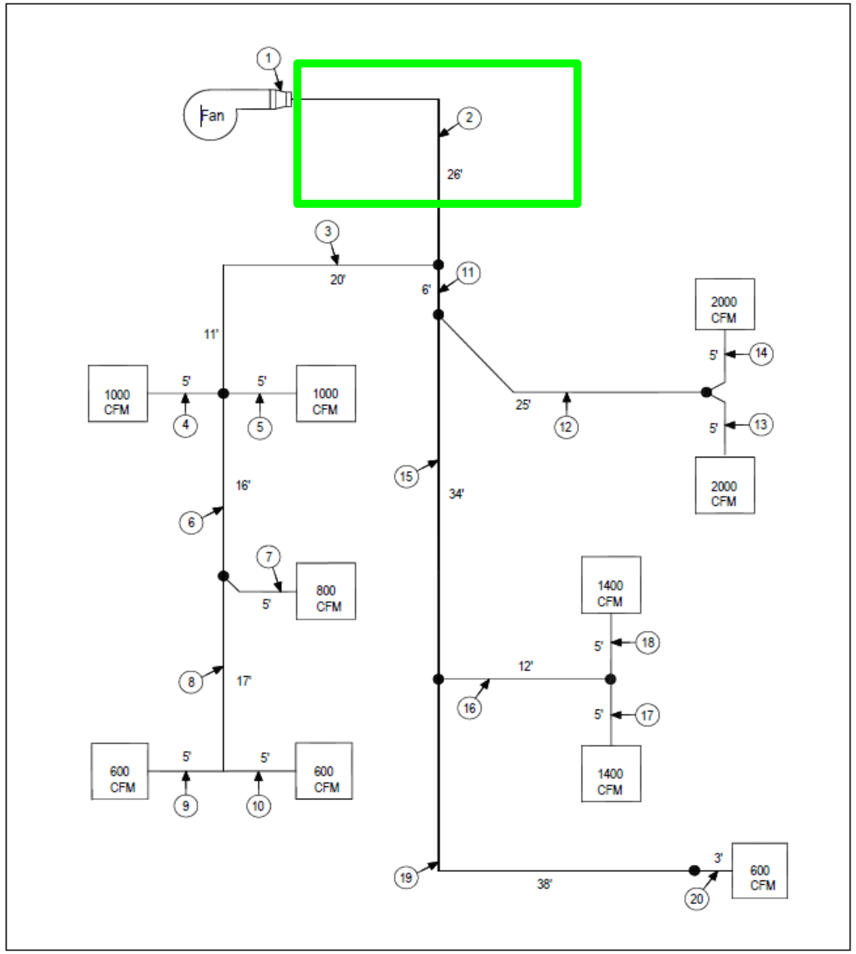
[1]Table 4-1 [Schaffer 2005 (2011)] [Table 8 from ASHRAE 2015 – HVAC Applications Chapter 48, Noise and Vibration Control]



Designing the Duct System

Example: Sizing the First Section

- Sum the airflow requirements at the terminal VAV boxes. Assume no diversity or leakage
- The total fan airflow is 11,400 cfm (5381 l/s)
- Sizing the first section for the maximum velocity of 3500 (17.8 m/s) results in a diameter size of 25 inches (625 mm) off the fan transition. Use CD11-3 for initial size and friction rate
- The initial velocity in the round section is 3,334 fpm (17.5 m/s)



Designing the Duct System

Example: Sizing the First Section, I-P

CD11-3 Straight Duct, Round, Maximum Velocity (Haaland 1983)
[Knowing Flow Rate (Q) and Target Velocity (V), determine Diameter (D)]

INPUT

Flow Rate (Q)	cfm	11400
Target Velocity (VT)	fpm	3500
Absolute Roughness (ei)	ft	0.00040
Density (RHO)	lbm/ft^3	0.061

Calculate   Load Defaults

OUTPUT

Diameter (D)	in.	25.0
Velocity (V)	fpm	3,344
Velocity Pressure (Pv)	in. wg	0.57
Reynolds Number (Re)		584,044
Fricton Factor (f)		0.0150
Friction Rate (Po)	in. wg per 100 ft	0.41

Designing the Duct System

Example: Sizing the First Section, SI

CD11-3 Straight Duct, Round, Maximum Velocity (Haaland 1983)
[Knowing Flow Rate (Q) and Target Velocity (V), determine Diameter (D)]

INPUT

Flow Rate (Q)	L/s	5381
Target Velocity (VT)	m/s	17.8
Absolute Roughness (ei)	mm	0.12
Density (RHO)	kg/m^3	0.983

Calculate Load Defaults

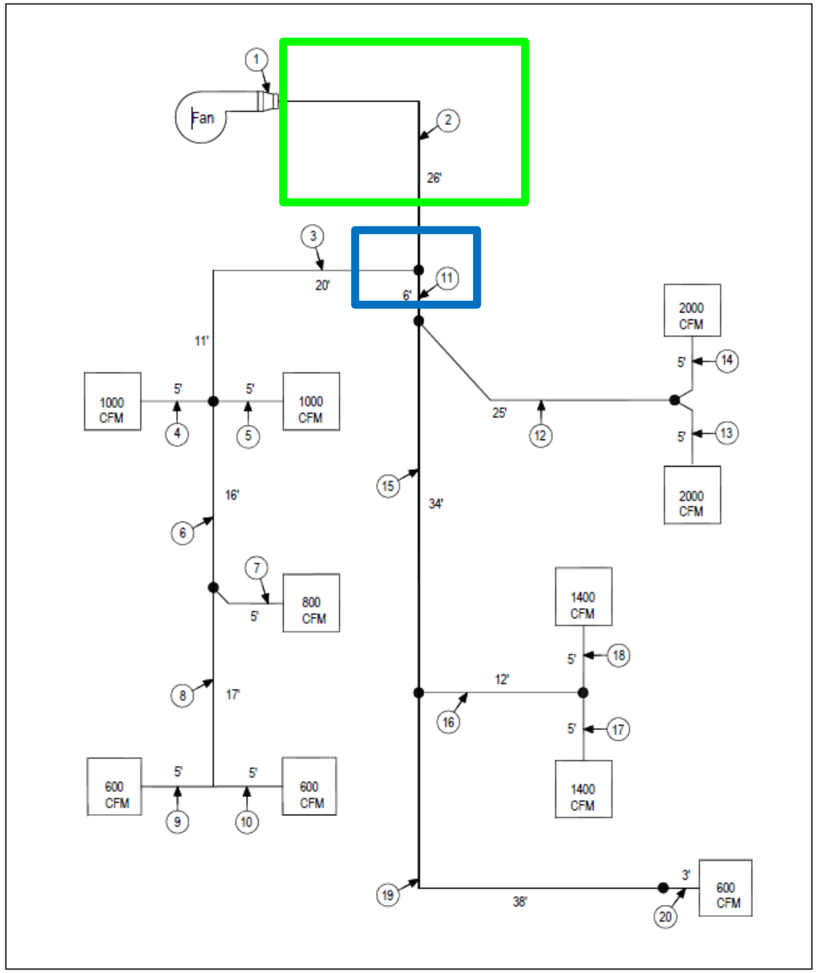
OUTPUT

Diameter (D)	mm	625
Velocity (V)	m/s	17.5
Velocity Pressure (Pv)	Pa	151
Reynolds Number (Re)		590,894
Fricton Factor (f)		0.0150
Friction Rate (Po)	Pa per m	3.63

Designing the Duct System

Example: Equal Friction

- That has a friction loss rate of 0.41 inch water per 100 ft (3.63 Pa/m)
- That rate will be used to size the other sections
- Use DFDB CD11-4 to size the Duct and CD11-1 to calculate the pressure loss
- We must also account for fitting losses, so a spreadsheet is used to calculate the data for each section



Designing the Duct System

Example: Equal Friction Section 11, I-P

CD11-4 Straight Duct, Round, Friction Rate Constant
[Knowing Flow Rate (Q) and Target Friction Rate, determine Diameter (D)]

INPUT

Flow Rate (Q)	cfm	7400
Length (L)	ft	11
Target Friction Rate (dpol)	in.wg/100 ft	.41
Absolute Roughness (e)	ft	.0004
Density (RHO)	lbm/ft^3	0.061

Calculate Load Defaults

OUTPUT

Calculated Diameter (D)	in.	21.27
Nominal Duct Diameter (Dnom)	in.	21.00
Air Velocity (V)	ft/min	3,077
Velocity Pressure (PV)	in. wg	0.483
Reynolds Number (Re)		451,548
Friction Factor (F)		0.0158
Friction Rate (DpolNom)	in. wg/100 ft	0.437
Pressure Loss (DP)	in.wg	0.048

Designing the Duct System

Example: Equal Friction Section 11, SI

CD11-4 Straight Duct, Round, Friction Rate Constant
[Knowing Flow Rate (Q) and Target Friction Rate, determine Diameter (D)]

INPUT

Flow Rate (Q)	L/s	3492
Length (L)	m	3.4
Target Friction Rate (dpol)	pa/m	3.63
Absolute Roughness (e)	mm	0.120
Density (RHO)	kg/m^3	0.983

Calculate Load Defaults

OUTPUT

Calculated Diameter (D)	mm	531
Nominal Duct Diameter (Dnom)	mm	531
Air Velocity (V)	m/s	15.8
Velocity Pressure (PV)	Pa	122
Reynolds Number (Re)		451,354
Friction Factor (F)		0.0158
Friction Rate (DpolNom)	Pa/m	3.637
Pressure Loss (DP)	Pa	12.365

Designing the Duct System

Example: Equal Friction Spreadsheet, I-P

Sizes per CD11-4

Equal Friction Example Problem (DDG) I-P | Friction Rate = 0.41in wg/100 ft per CD11-3, V target = 3500 fpm for Acoustics

Air Temperature, °F	69	Relative Humidity, %	0
Elevation, ft	5430	Air Density, lbm/ft^3	0.061
Barometric Pressures, psia	12.032	Viscosity (μ), lbm/(ft-min)	0.00073245

Upstream Section	Section	Fitting	ASHRAE Fitting Code	Air Quantity (cfm)	Duct Size (in.)	Velocity (fpm)	Duct Length (ft)	Velocity Pressure, p_v (in. wg)	Loss Coefficient, C	Total Pressure Loss (in. wg)
		Source		Source					Source	
		Drawings	DFDB	Drawing	DFDB	DFDB	Drawing	DFDB	DFDB	∑
1	2	Duct	CD11-3/CD11-1	11400	25	3344	26			0.11
		Elbow, 90°	CD3-9						0.13	
		Transition: H1=27.0",W1=20.0",L=24" (Theta1=5°,Theta2=12°)	SD4-2						0.01	
		Sized at Maximum Velocity of 3500 fpm						0.57	0.14	0.08
Section Total										0.19
2	11	Duct	CD11-1	7400	21	3077	11			0.05
		Tee, 45° Entry, Main (Dc=25, Ds=21, Db=17)	SD5-12						0.14	
								0.48	0.14	0.07
Section Total										0.12

Designing the Duct System

Example: Equal Friction Spreadsheet, SI

Sizes per CD11-4

Equal Friction Example Problem (DDG) — Friction Rate = 3.63 Pa/m per CD11-3, V target = 17.8 for Acoustics

Air Temperature, °C	21	Relative Humidity, %	0
Elevation, m	1655	Air Density, kg/m³	0.983
Barometric Pressures, Kpa	83	Viscosity (μ), Kg/m -s	1.82E-05

Upstream Section	Section	Fitting	ASHRAE Fitting Code	Air Quantity (L/s)	Duct Size (mm)	Velocity (m/s)	Duct Length (m)	Velocity Pressure, p_v (Pa)	Loss Coefficient, C	Total Pressure Loss (Pa)
		Source		Source					Source	
		Drawings	DFDB	Drawing	DFDB	DFDB	Drawing	DFDB	DFDB	∑
1	2	Duct	CD11-3,CD11-1	5380	625	17.5	7.9			28.1
		Elbow, 90°	5380						0.13	
		Transition: H1=685 mm",W1=508 mm,L=610 mm (Theta1=5°,Theta2=12°)	SD4-2						0.01	
		Sized at Maximum Velocity of 17.8 m/s						150.00	0.14	21.0
Section Total										49.1
2	11	Duct	CD11-1	3492	533	15.8	3.4			10.60
		Tee, 45° Entry, Main	SD5-12						0.14	
								122.00	0.14	17.08
Section Total										27.68

Designing the Duct System

Example: Equal Friction Unbalance

Unbalance I-P:

Path to Terminal Box	TP (in. wg)	Excess Pressure (in. wg)	% Deviation
(4/5)	1.28	0.02	1.3%
(7)	1.27	0.03	2.1%
(9/10)	1.30	0.00	0.0%
(13/14)	1.26	0.04	3.1%
(17/18)	1.20	0.09	7.1%
(20)	1.15	0.14	11.1%

Unbalance SI:

Equal Friction			
Path to Terminal Box	TP (Pa)	Excess Pressure (Pa)	% Deviation
(4/5)	330	39	10.6%
(7)	314	55	15.0%
(9/10)	322	47	12.6%
(13/14)	369	0	0.0%
(17/18)	348	21	5.8%
(20)	312	57	15.6%

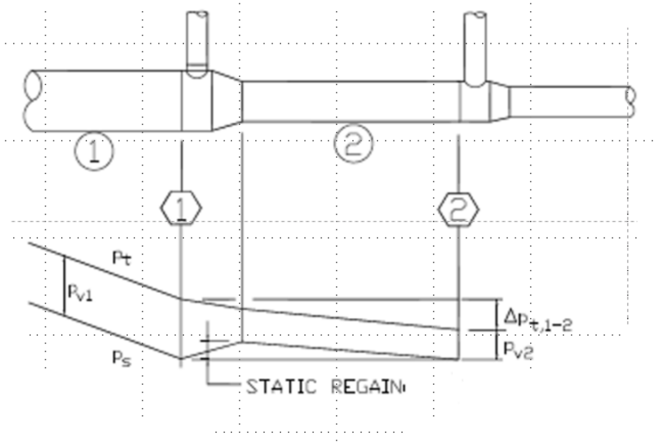
Designing the Duct System

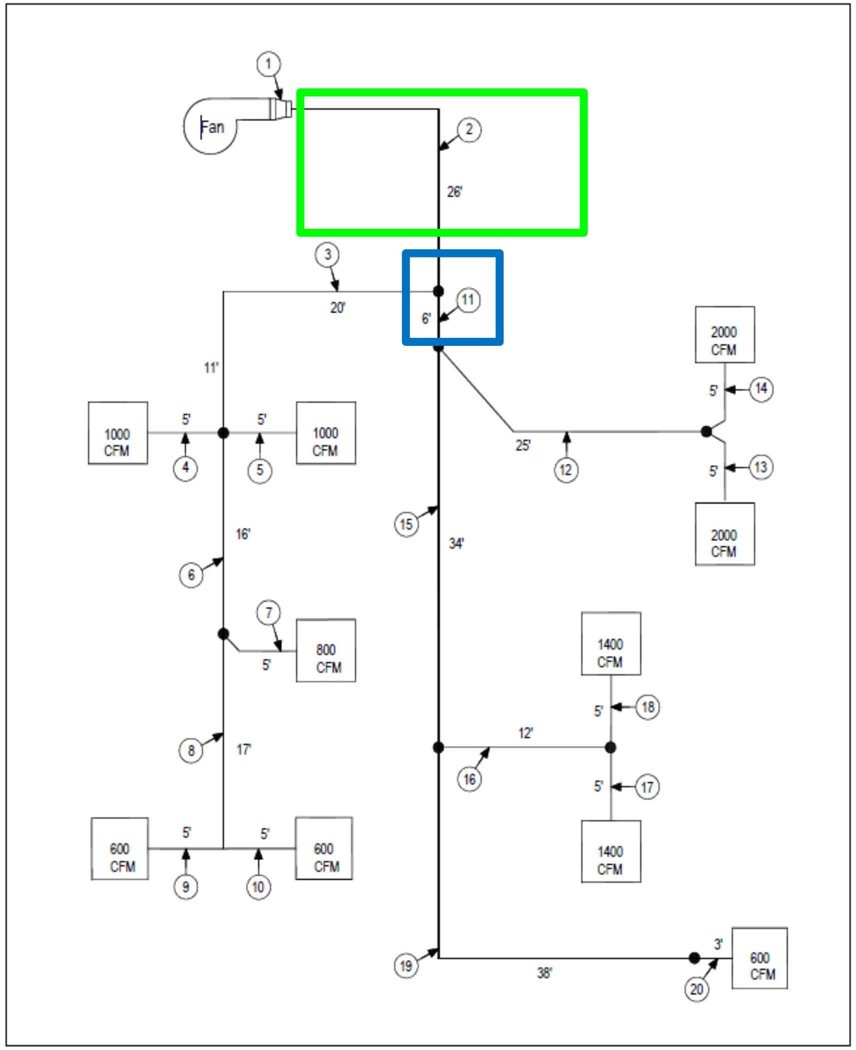
Example: Static Regain

- Size the first section the same as the Equal Friction Method using the Maximum Recommended Duct AirFlow Velocities to Achieve Specified Acoustic Design Criteria table.

- Use the static regain equation for other sections:

$$\Delta\, p_{s,1\text{-}2} = p_{v1} - p_{v2} - \Delta p_{t,1\text{-}2} = 0$$





Designing the Duct System

Example: Static Regain, I-P

Static Regain Spreadsheet

Air Temperature, °F	69	Relative Humidity, %	0
Elevation, ft	5430	Air Density, lbm/ft³	0.061
Barometric Pressures, psia	12.032	Viscosity (μ), lbm/(ft-min)	0.00073245

Upstream Section	Section	Fitting	ASHRAE Fitting Code	Air Quantity (cfm)	Duct Size (in.)	Velocity (fpm)	Duct Length (ft)	Velocity Pressure, p_v (in. wg)	Loss Coefficient, C	Total Pressure Loss (in. wg)	Regain (In. wg) $[p_{v1} - p_{v2}] - \Delta p_t$
Source											
		Drawings	DFDB	Drawings	Iteration	DFDB	Drwing	DFDB	DFDB	Σ	Static Regain Calc
		Duct	CD11-1				26			0.11	
1	2*	Elbow	CD3-9	11400	25	3334			0.13		
		Transition: H1= 20", W1= 27", L=24" (Theta1=17°, Theta2=0°)	SD4-2						0.01		
		Sized at Maximum Velocity of 3500 fpm						0.57	0.14	0.08	
Section Total										0.19	

Designing the Duct System

Example: Static Regain, SI

Example Static Regain Spreadsheet SI

Air Temperature, °C		21	Relative Humidity, %	0							
Elevation, m		1655	Air Density, kg/m^3	0.983							
Barometric Pressures, Kpa		83	Viscosity (μ), Kg/m -s	1.82E-05							
Upstream Section	Section	Fitting	ASHRAE Fitting Code	Air Quantity (L/s)	Duct Size (mm)	Velocity (m/s)	Duct Length (m)	Velocity Pressure, p_v (Pa)	Loss Coefficient, C	Total Pressure Loss (Pa)	Regain (Pa) $[p_{v1} - p_{v2}] - \Delta p_t$
		Source		Source				Source			Source
		Drawings	DFDB	Drawings	Iteration	DFDB	Drawings	DFDB	DFDB	Σ	Static Regain Calculation
1	2*	Duct	CD11-1	5360	625	17.5	7.9			28.1	
		Elbow	CD3-9						0.13		
			SD4-2						0.01		
		Sized at Maximum Velocity of 3500 fpm						150.00	0.14	21.00	
Section Total										49.10	

Designing the Duct System

Example: Static Regain, I-P

Upstream Section	Section	Fitting	ASHRAE Fitting Code	Air Quantity (cfm)	Duct Size (in.)	Velocity (fpm)	Duct Length (ft)	Velocity Pressure, p_v (in. wg)	Loss Coefficient, C	Total Pressure Loss (in. wg)	Regain (In. wg) $[p_{v1}-p_{v2}]-\Delta p_t$
Source		Drawings	DFDB	Drawings	Iteration	DFDB	Drwing	DFDB	DFDB	Σ	Static Regain Calc
2	11d*	Duct	CD11-1	7400	22	2803	11			0.04	
		Tee, 45° Entry, Main: Dc=25, Ds=22, Db=22	SD5-12						0.14		
		4rth iteration						0.40	0.14	0.06	(0.57-0.40)-0.10
Section Total										0.10	0.07
2	11e	Duct	CD11-1	7400	21	3077	11			0.04	
		Tee, 45° Entry, Main: Dc=25, Ds=21, Db=21	SD5-12						0.14		
		5th iteration						0.48	0.14	0.07	(0.57-0.48)-0.11
Section Total										0.11	-0.02

Designing the Duct System

Example: Static Regain, SI

Example Static Regain Spreadsheet SI

Air Temperature, °C	21	Relative Humidity, %	0
Elevation, m	1655	Air Density, kg/m^3	0.983
Barometric Pressures, Kpa	83	Viscosity (μ), Kg/m -s	1.82E-05

stream ection	ection	Fitting	ASHRAE Fitting Code	Air Quantity (L/s)	Duct Size (mm)	Velocity (m/s)	Duct Length (m)	Velocity Pressure, p_v (Pa)	Loss Coefficient, C	Total Pressure Loss (Pa)	Regain (Pa) $[p_{v1} - p_{v2}] - \Delta p_t$
2	11d*	Tee, 45° Entry, Main: 4th iteration	SD5-12	3492	559	14.2			0.14		
								99.00	0.14	13.86	
Section Total										23.16	27.84
2	11e	Duct	CD11-1	3492	533	15.7	3.4			11.7	
		Tee, 45° Entry, Main: 5th iteration	SD5-12						0.14		
								120.00	0.14	16.80	
Section Total										28.50	1.50

Designing the Duct System

Example: Static Regain

Critical Paths

I-P

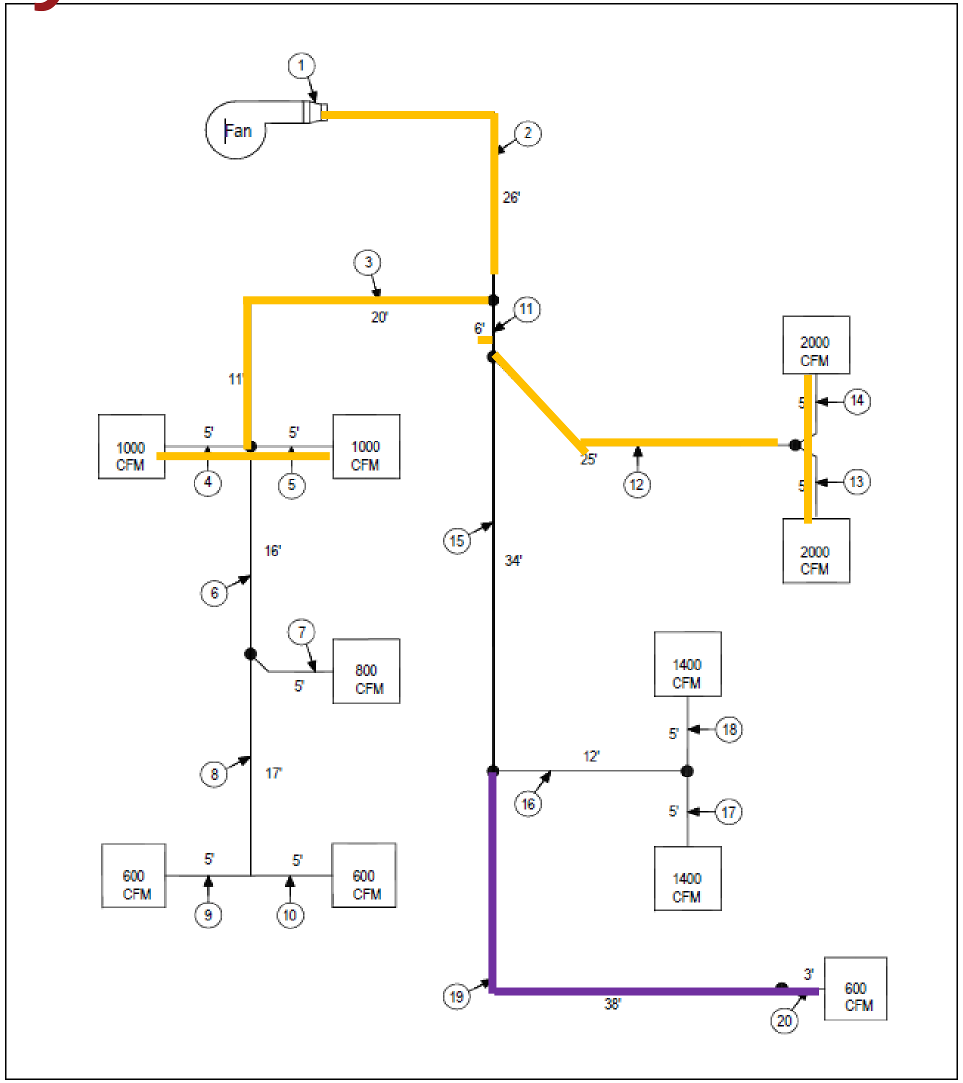
Static Regain Design			
Path to Terminal Box	TP (in. wg)	Excess Pressure (in. wg)	% Deviation
(4/5)	1.16	0.00	0.0%
(7)	1.12	0.04	3.9%
(9/10)	1.09	0.07	6.1%
(13/14)	1.16	0.00	0.1%
(17/18)	1.06	0.11	9.3%
(20)	1.04	0.13	11.1%
		Average % Deviation	5.1%

SI

Static Regain Design			
Terminal Box	TP (Pa)	Excess Pressure (Pa))	% Deviation
(4/5)	295	4	1.4%
(7)	279	20	6.6%
(9/10)	280	19	6.3%
(13/14)	299	0	0.0%
(17/18)	274	25	8.4%
(20)	232	67	22.5%
		Average % Deviation	7.5%

Designing the Duct System

Critical Paths and
Excess Pressure



Designing the Duct System

Critical Paths and Excess Pressure
Static Regain vs Equal Friction

Unbalance I-P

Static Regain Design			
Path to Terminal Box	TP (in. wg)	Excess Pressure (in. wg)	% Deviation
(4/5)	1.16	0.00	0.0%
(7)	1.12	0.04	3.9%
(9/10)	1.09	0.07	6.1%
(13/14)	1.16	0.00	0.1%
(17/18)	1.06	0.11	9.3%
(20)	1.04	0.13	11.1%

Average % Deviation 5.1%

Unbalance I-P:

Equal Friction			
Path to Terminal Box	TP (in. wg)	Excess Pressure (in. wg)	% Deviation
(4/5)	1.28	0.02	1.3%
(7)	1.27	0.03	2.1%
(9/10)	1.30	0.00	0.0%
(13/14)	1.26	0.04	3.1%
(17/18)	1.20	0.09	7.1%
(20)	1.15	0.14	11.1%

Average % Dev 4.7%

Designing the Duct System

Critical Paths and Excess Pressure
Static Regain vs Equal Friction

Unbalance SI

Static Regain Design			
Terminal Box	TP (Pa)	Excess Pressure (Pa))	% Deviation
(4/5)	295	4	1.4%
(7)	279	20	6.6%
(9/10)	280	19	6.3%
(13/14)	299	0	0.0%
(17/18)	274	25	8.4%
(20)	232	67	22.5%

Average % Deviation 7.5%

Unbalance SI:

Equal Friction			
Path to Terminal Box	TP (Pa)	Excess Pressure (Pa)	% Deviation
(4/5)	330	37	10.2%
(7)	314	54	14.6%
(9/10)	322	45	12.2%
(13/14)	367	0	0.0%
(17/18)	346	21	5.8%
(20)	310	57	15.7%

Average % Dev 9.6%

Designing the Duct System

Total Pressure Method Comparison with Static Regain

	I-P				SI				
	Static Regain Design		Total Pressure Design			Static Regain Design		Total Pressure Design	
Section	ΔP_t (in wg)	Size (inch)	ΔP_t (in wg)	Size (inch)	Section	ΔPt (Pa)	Size (mm)	Pt (in wg)	Size (inch)
1	0.0	20 x 27 to 25 Transition	0.00	20 x 27 to 25 Transition	1	0.0	685 x 508 x610 Transition	0.0	20 x 27 to 25 Transition
2	0.19	25	0.19	25	2	49.1	625	49.1	625
3	0.37	20	0.37	20	3	93.6	483	93.6	483
4 & 5 T	0.61	10	0.61	10	4 & 5 T	152.3	254	152.3	254
6	0.04	16	0.04	16	6	9.5	406	9.5	406
7 T	0.53	9	0.53	9	7 T	127.2	229	127.2	229
8	0.03	14	0.03	14	8	6.6	356	6.6	356
9 &10 T	0.48	8	0.48	8	9 &10 T	121.4	203	121.4	203
11	0.10	22	0.11	22	11	28.5	559	27.2	559
12	0.26	21	0.31	20	12	64.6	533	75.7	533
13 & 14 T	0.62	14	0.55	14	13 & 14 T	133.2	356	142.2	356
15	0.10	17	0.11	17	15	24.2	432	24.2	432
16	0.11	17	0.11	17	16	33.6	432	33.6	432
17 & 18	0.56	12	0.56	12	17 & 18	138.4	305	148.6	305
19	0.15	9	0.25	8	19	37.1	229	76.9	203
20	0.49	8	0.46	8	20	92.84	229	105.5	229

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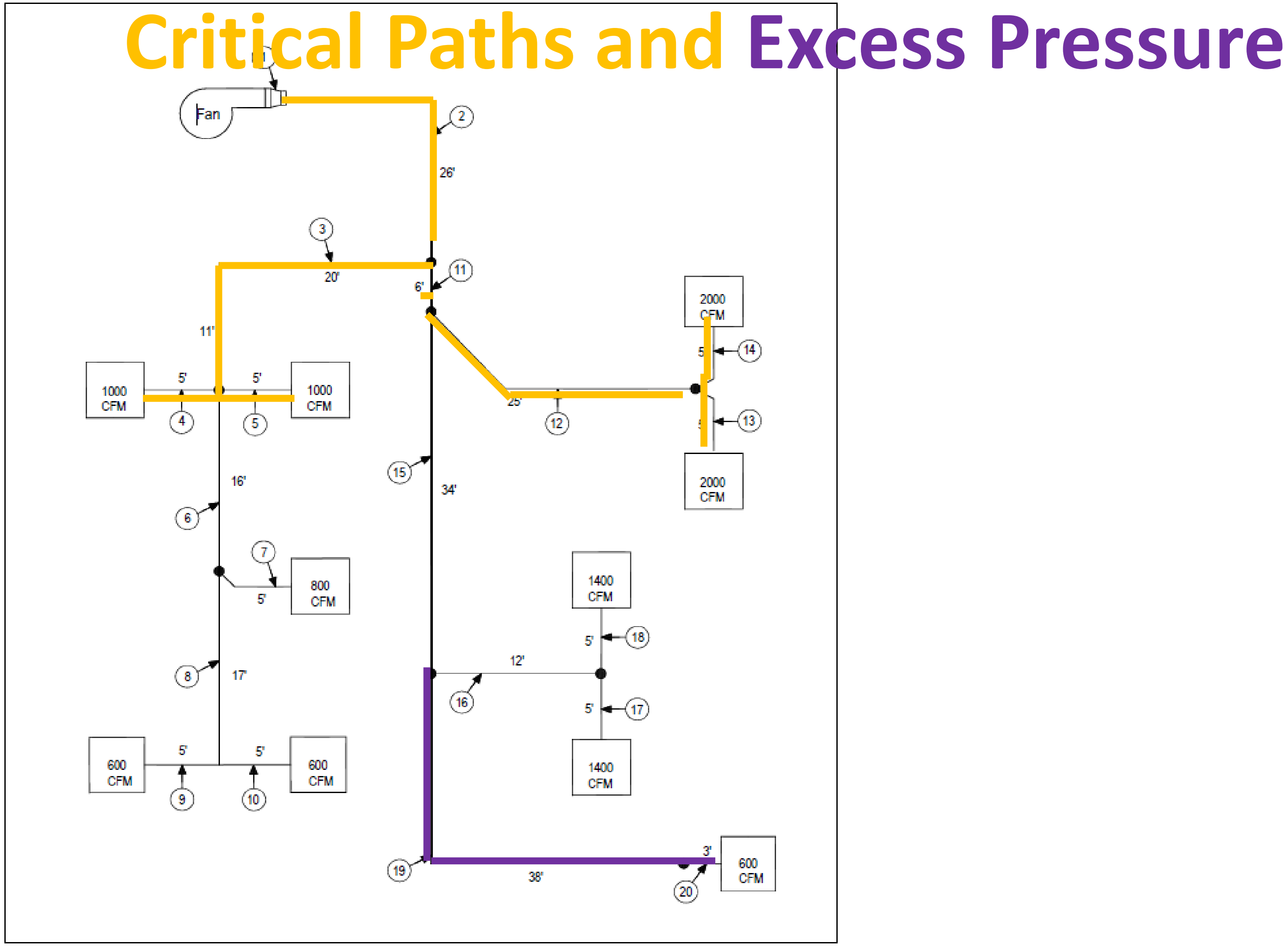

Designing the Duct System

Total Pressure Method Comparison with Static Regain, I-P

I-P

Static Regain Design			
Path to Terminal Box	TP (in. wg)	Excess Pressure (in. wg)	% Deviation
(4/5)	1.16	0.00	0.0%
(7)	1.12	0.04	3.9%
(9/10)	1.09	0.07	6.1%
(13/14)	1.16	0.00	0.1%
(17/18)	1.06	0.11	9.3%
(20)	1.04	0.13	11.1%
		Average % Deviation	5.1%

Total Pressure Design			
Path to Terminal Box	TP (in. wg)	Excess Pressure (in. wg)	% Deviation
(4/5)	1.16	0.00	0.0%
(7)	1.12	0.04	3.9%
(9/10)	1.09	0.07	6.1%
(13/14)	1.15	0.01	1.0%
(17/18)	1.07	0.09	8.0%
(20)	1.12	0.05	3.9%
		Average % Deviation	3.8%

Designing the Duct System

Total Pressure Method Comparison with Static Regain, SI

SI

Static Regain Design			
Terminal Box	TP (Pa)	Excess Pressure (Pa))	% Deviation
(4/5)	295	4	1.4%
(7)	279	20	6.6%
(9/10)	280	19	6.3%
(13/14)	299	0	0.0%
(17/18)	274	25	8.4%
(20)	232	67	22.5%
		Average % Deviation	7.5%

Total Pressure Design			
Terminal Box	TP (Pa)	Excess Pressure (Pa))	% Deviation
(4/5)	295	0.00	0.0%
(7)	279	15.57	5.3%
(9/10)	280	14.67	5.0%
(13/14)	294	0.70	0.2%
(17/18)	283	12.17	4.1%
(20)	283	12.09	4.1%
		Average % Deviation	3.1%

Conclusion

- ✓ Equal Friction Designs May Not be Well Balanced
- ✓ Static Regain Designs Should be Better Balanced and for the Same First Section, Should Have a Lower Operating Cost than Equal Friction
- ✓ First Sections can be Sized to Meet Acoustical Objectives
- ✓ Efficient Fittings Should be Used in the Initial Design
- ✓ Smaller Duct Sizes or Less Efficient Fittings can be Used After the Initial Design to help Balance the non-design legs which should Lower First Cost… This is the Total Pressure Method of Design and can be Applied to Equal Friction or Static Regain Design.

Thank you for your time!

*To receive PDH credit for today's program, you **must** complete the online evaluation, which will be sent via email 1 hour after the conclusion of this session.*

PDH credits and participation certificates will be issued electronically within 30 days, once all attendance records are checked and online evaluations are received.

Attendees will receive an email at the address provided on your registration, listing the credit hours awarded and a link to a printable certificate of completion.

Questions?

NEXT PROGRAM

Join us for our next AMCA & O’Dell Associates Education Session:

- Tuesday, December 21
- 10:00-11:00am ET
- ***Topic: Stall Detection & Control in Commercial and Industrial Fans***
- Presenter: Geoff Sheard, President, AGS Consulting

>> For additional session details please contact Sarah Johnson, Marketing Manager, O’Dell Associates (sjohnson@odellassoc.com)