Duct Design

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Why duct design is important

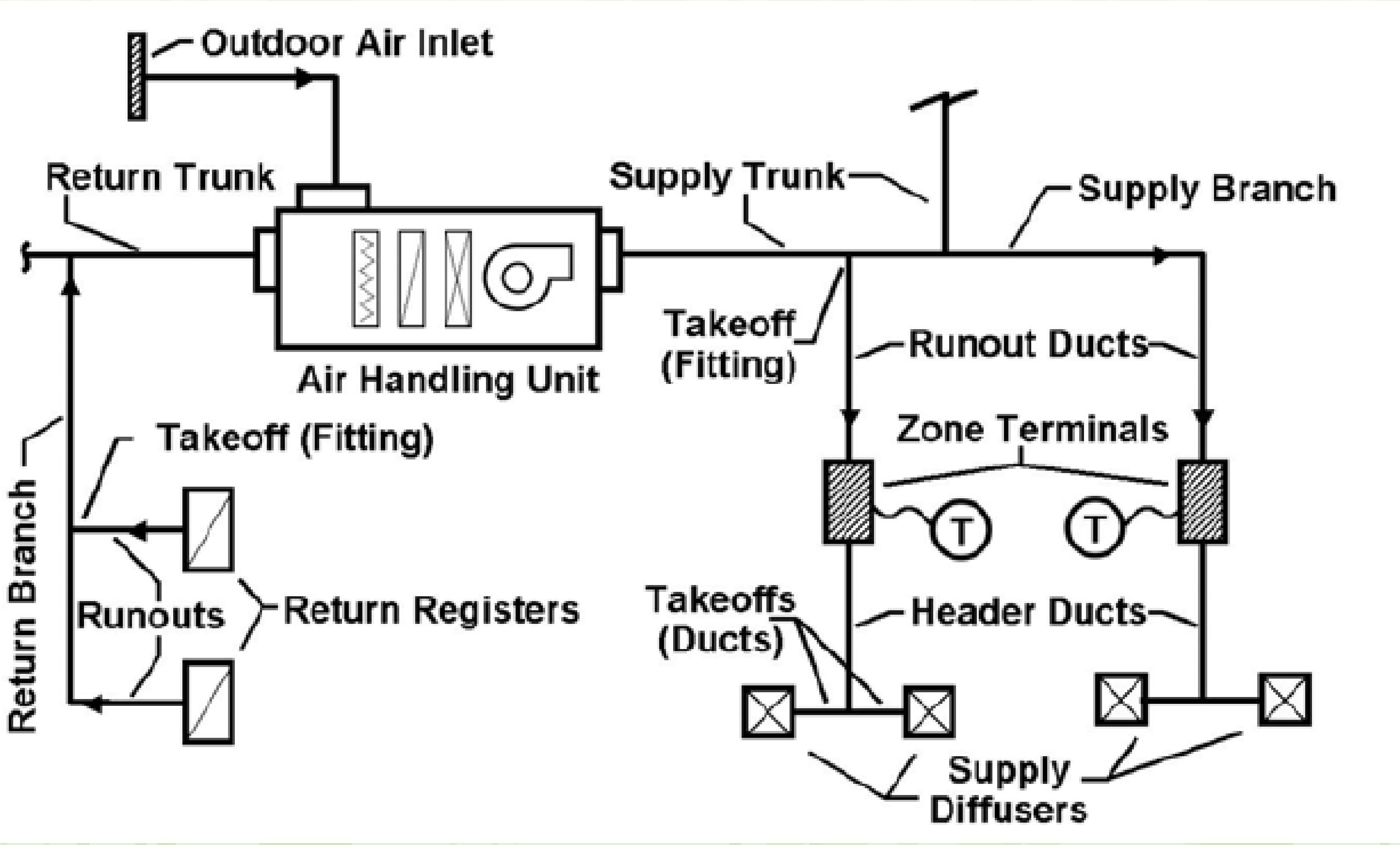
- The efficiency of air distribution systems has been found to be 60-75% or less in many houses because of insufficient and/or poorly installed duct insulation and leaks in the duct system. Properly designed and installed duct systems can have efficiencies of 80% or more for little or no additional cost.

- Duct systems that are undersized, are pinched, or have numerous bends and turns may lead to low air flow rates and high air velocities.
- Low air flow rates cause the heating and cooling equipment to operate inefficiently.
- High air velocities increase noise.

Duct Design Objective

- The objectives of good duct design are occupant comfort, proper air distribution, economical heating and cooling system operation, and economical duct installation.
- The outcome of the duct design process will be a duct system (supply and return plenums, ducts, fittings, boots, grilles, and registers) that provides conditioned air to meet all room heating and cooling loads.

Duct Terms..


Outdoor Air Inlet
Return Trunk
Supply Trunk
Supply Branch
Takeoff (Fitting)
Runout Ducts
Air Handling Unit
Zone Terminals
Takeoff (Fitting)
Return Branch
T
T
Runouts
Return Registers
Takeoffs (Ducts)
Header Ducts
Supply Diffusers

Duct materials

- Sheet metals
- Glass fibers
- Flexible nonmetallic

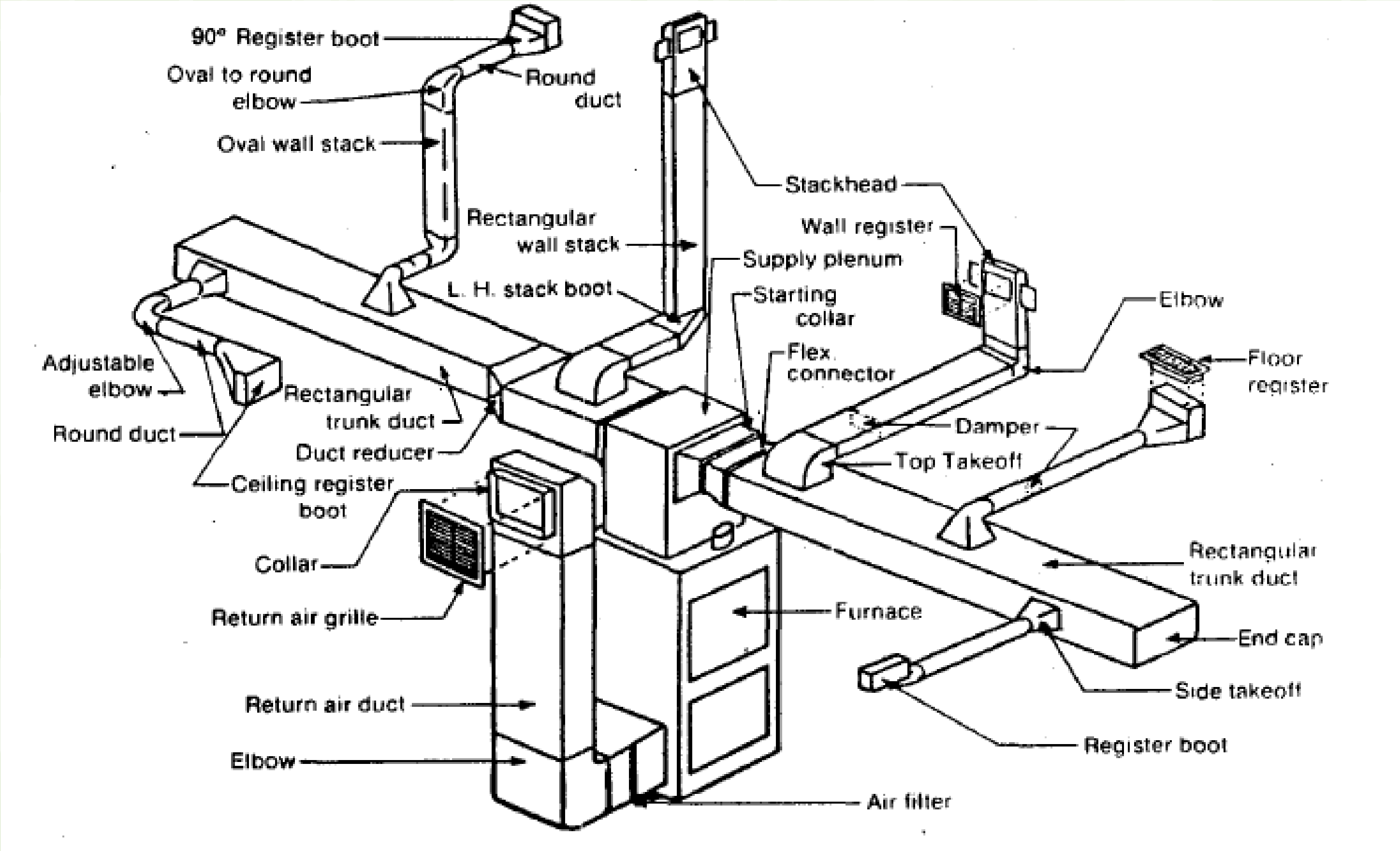
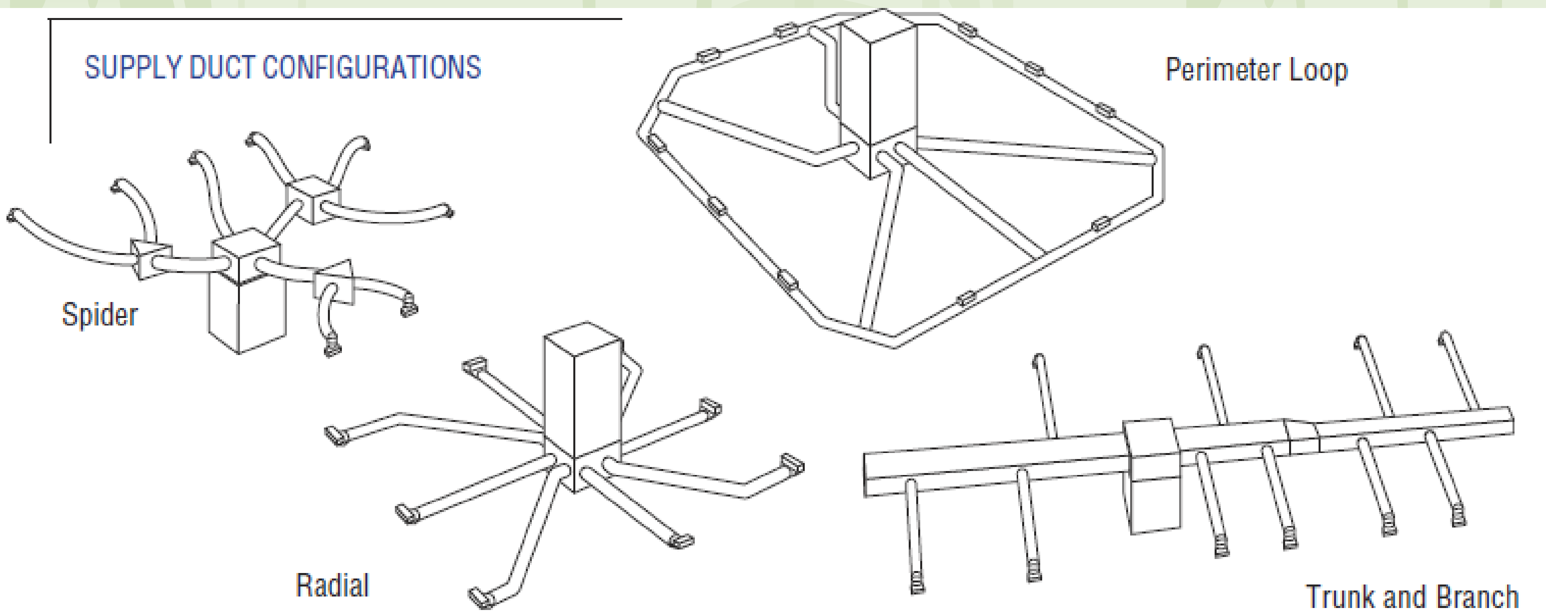


Fig. 5. Composite duct system

Supply Duct Systems

- Supply ducts deliver air to the spaces that are to be conditioned. The two most common supply duct systems for residences are the trunk and branch system and the radial system because of their versatility, performance, and economy.
- The spider and perimeter loop systems are other options.



The trunk and branch system

- A large main supply trunk is connected directly to the air handler or its supply plenum and serves as a supply plenum or an extension to the supply plenum.
- Smaller branch ducts and runouts are connected to the trunk.
- The trunk and branch system is adaptable to most houses, but it has more places where leaks can occur. It provides air flows that are easily balanced and can be easily designed to be located inside the conditioned space of the house.

- The principal design limitation of the extended Plenum system is the length of single-size trunk duct. To maintain reasonably uniform air Pressures in the air-distribution system, the length of a single-size trunk duct should be limited to about 24 feet. When this length is exceeded. pressure tends to build up toward the end of the duct, resulting in too much airflow in branches near the ends. And insufficient airflow in branches closer to the equipment.

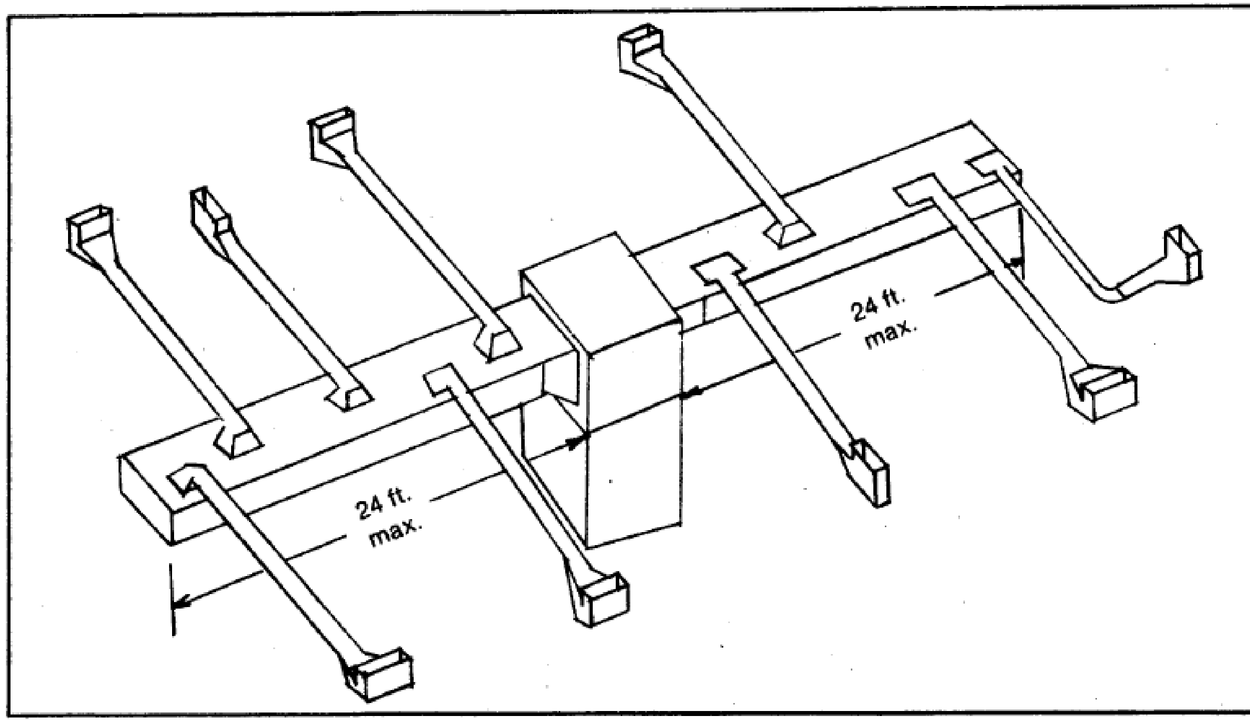


Fig. 41. Extended plenum duct system, equipment centrally located

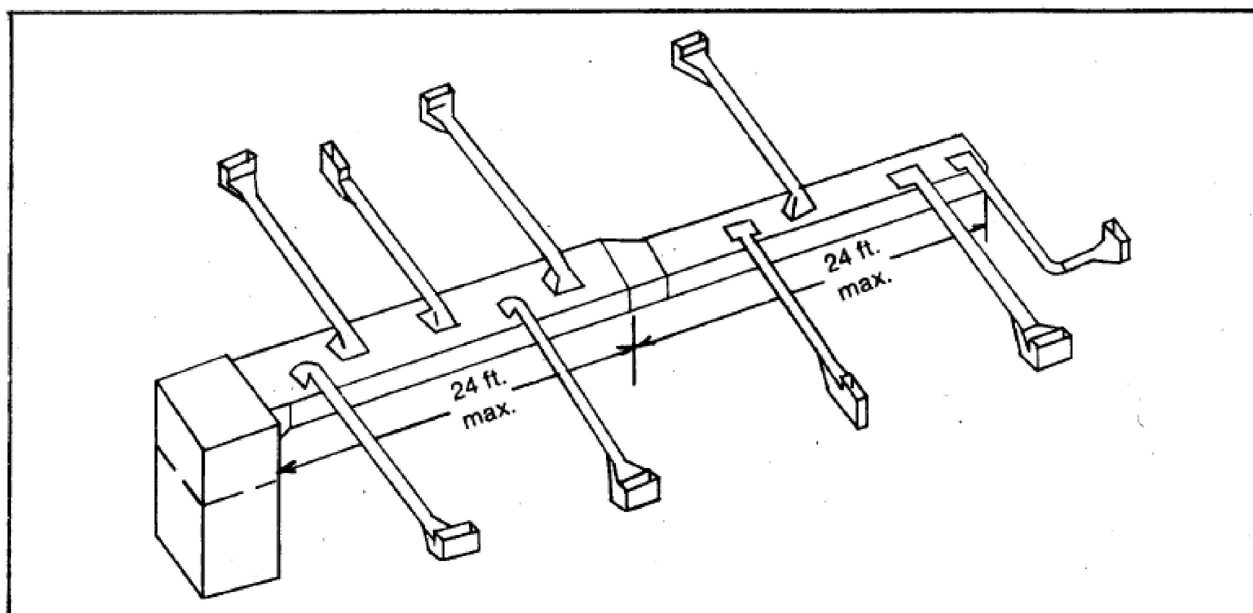


Fig. 42. Reduced extended plenum duct system, equipment at one end of structure

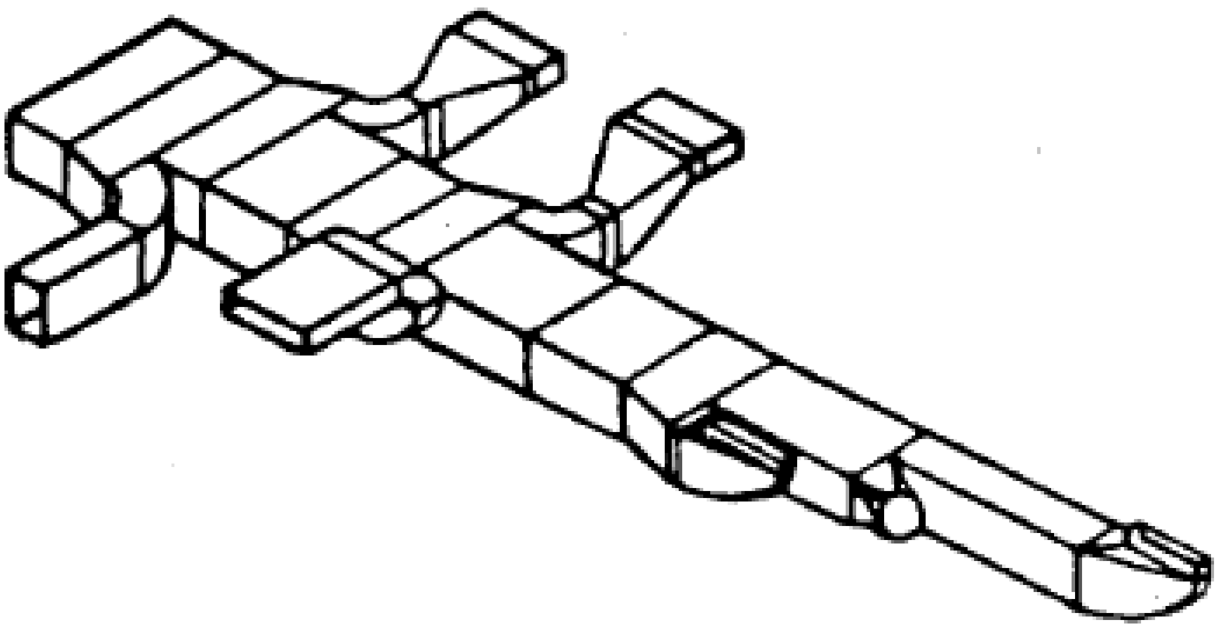


Fig. 60 Typical section of reducing trunk system

SPIDER SYSTEM

- A spider system is a more distinct variation of the trunk and branch system. Large supply trunks (usually large-diameter flexible ducts) connect remote mixing boxes to a small, central supply plenum. Smaller branch ducts or runouts take air from the remote mixing boxes to the individual supply outlets. This system is difficult to locate within the conditioned space of the house.

RADIAL SYSTEM

- No main supply trunk; branch ducts or runouts that deliver conditioned air to individual supply outlets are essentially connected directly to the air handler, usually using a small supply plenum. The short, direct duct runs maximize air flow. The radial system is most adaptable to single-story homes. This system is normally associated with an air handler that is centrally located so that ducts are arranged in a radial pattern. However, symmetry is not mandatory, and designs using parallel runouts can be designed so that duct runs remain in the conditioned space (e.g., installed above a dropped ceiling).

PERIMETER LOOP SYSTEM

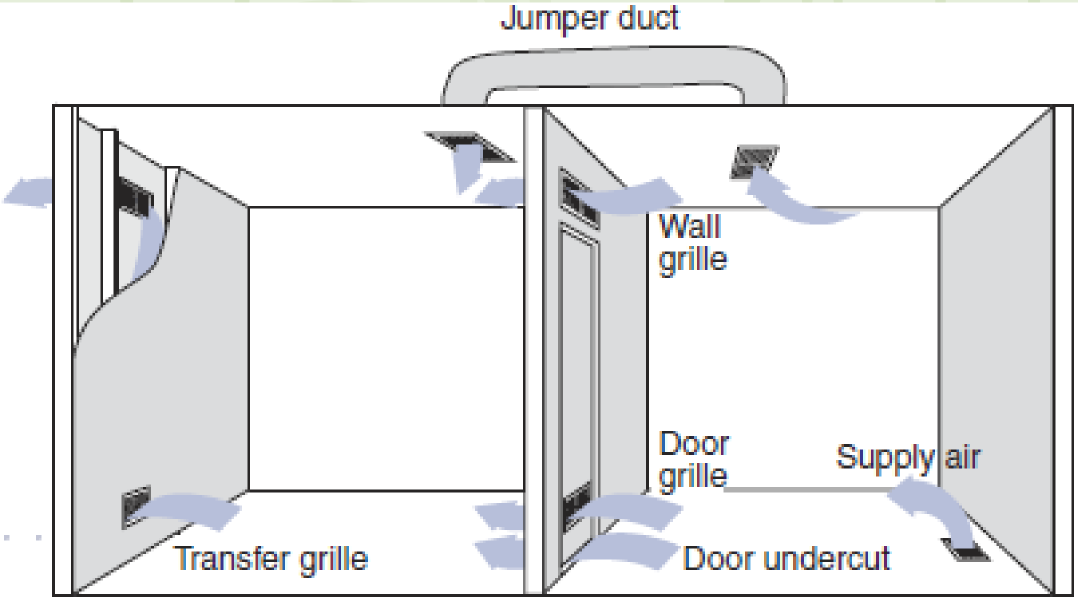
- A perimeter loop system uses a perimeter duct fed from a central supply plenum using several feeder ducts. This system is typically limited to houses built on slab in cold climates and is more difficult to design and install.

RETURN DUCT SYSTEMS

- Return ducts remove room air and deliver it back to the heating and cooling equipment for filtering and reconditioning. Return duct systems are generally classified as either central or multiple-room return.

RETURN AIR TECHNIQUES

Closed interior doors create a return-air blockage in systems with only one or two returns. Grilles through doors or walls or jumper ducts can reduce house pressures and improve circulation.



MULTIPLE-ROOM RETURN SYSTEM

- Return air from each room supplied with conditioned air, especially those that can be isolated from the rest of the house (except bathrooms and perhaps kitchens and mechanical rooms). The ultimate return duct system ensures that air flow is returned from all rooms (even with doors closed), minimizes pressure imbalances, improves privacy, and is quiet. However, design and installation costs of a multi-room return system are generally higher than costs for a central return system, and higher friction losses can increase blower requirements.

CENTRAL RETURN SYSTEM

- Consists of one or more large grilles located in central areas of the house (e.g., hallway, under stairway) and often close to the air handler. To ensure proper air flow from all rooms, especially when doors are closed, transfer grilles or jumper ducts must be installed in each room.
- Transfer grilles are through-the-wall vents that are often located above the interior door frames, although they can be installed in a full wall cavity to reduce noise transmission. The wall cavity must be well sealed to prevent air leakage. Jumper ducts are short ducts routed through the ceiling to minimize noise transfer.

Central return - Advantages

- Ductwork minimal: usually one large duct witha relatively short run
- Allows for sufficient air flow with a minimum of air friction loss, thus minimizing blower requirements
- Easy to install
- Often preferred with an open plan Permits convenient air-filter servicing if a filtergrille is used, especially if equipment is in an attic or crawl space
- Generally less costly

Central return - Disadvantages

- Generally noisier unless special acoustical pro-Doors to individual rooms must be undercut to Large duct may require a special chase Large grille can be unattractive visions Doors to individual rooms must be undercut are made
- Doors to individual rooms must be undercut to permit proper airflow
- Large duct may require a special chase
- Large grille can be unattractive

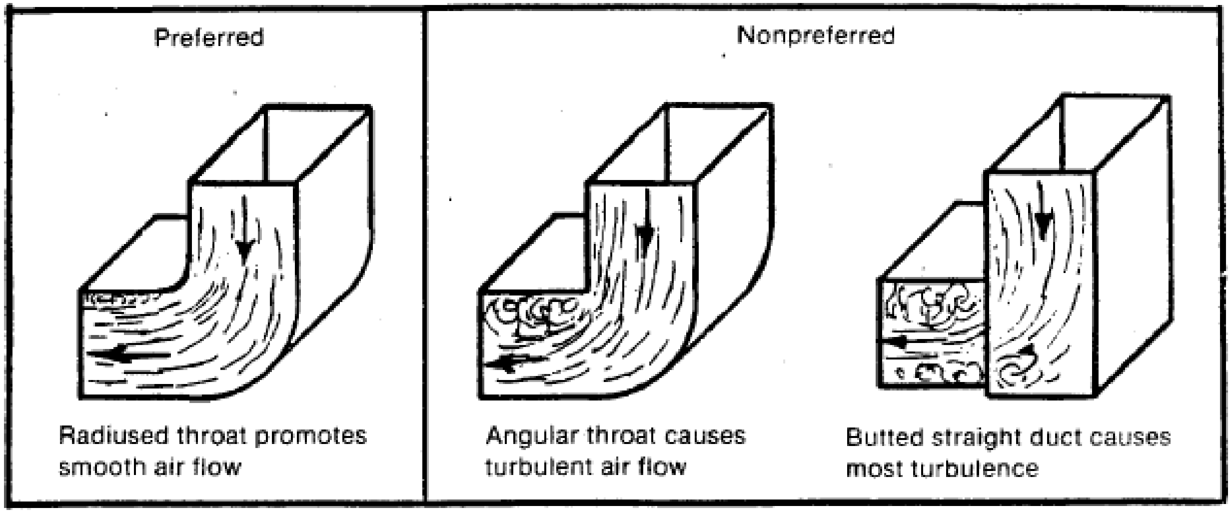


Fig. 63. Effect of fitting design on air flow

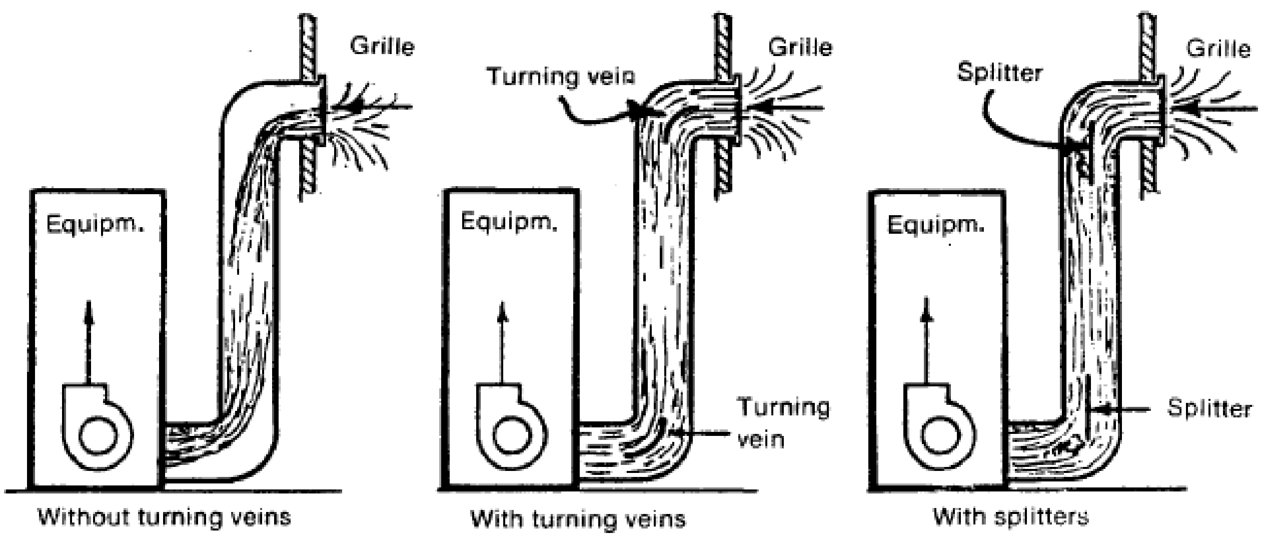


Fig. 64. Air flow in central returns

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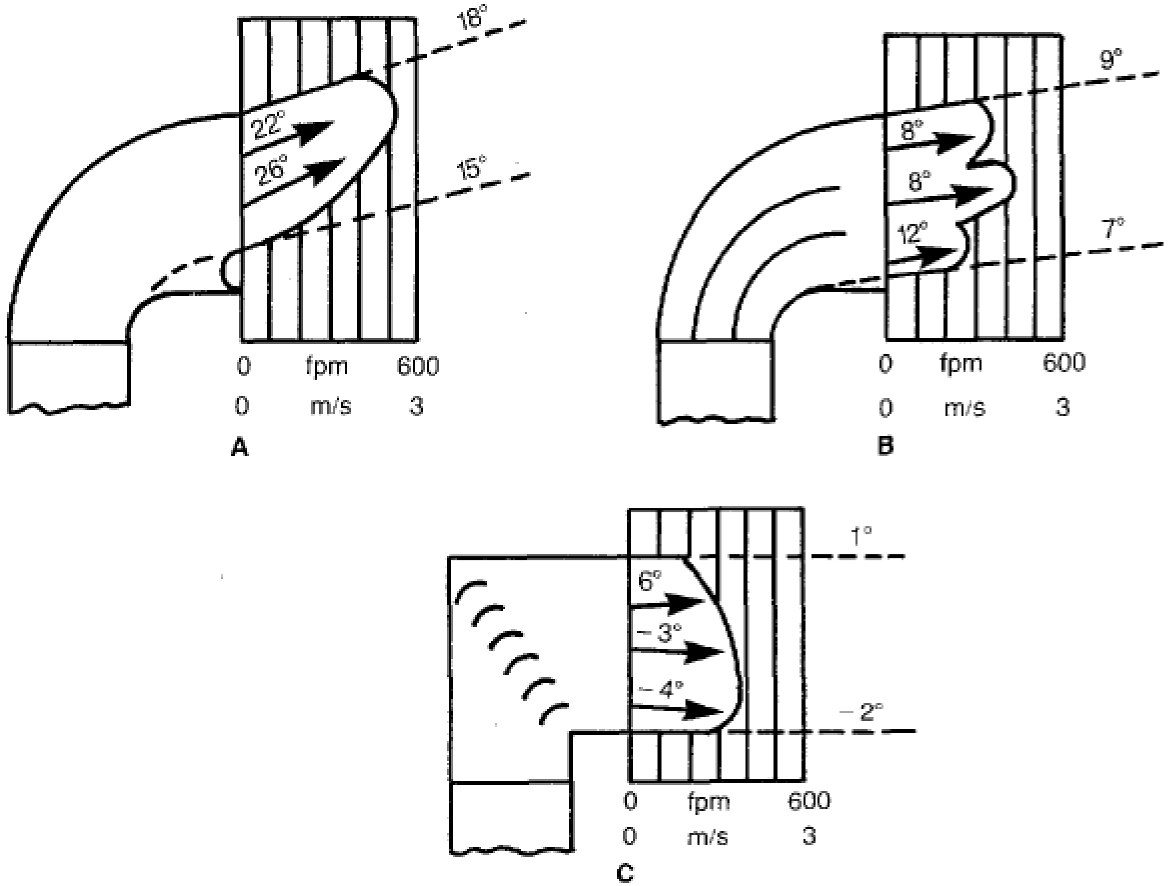


Stack, 14 · 6 in. (350 × 150 mm); Outlets, 14 · 9 in. (350 × 225 mm); Stack Velocity, 500 fpm (2.5 m/s)

A. Rounded Throat and Rounded Back.
B. Rounded Throat and Back and 2 Splitters.
C. Square Throat and Back and Turning Vanes.

Figure 3-4 OUTLET VELOCITY AND AIR DIRECTION DIAGRAMS FOR STACK HEADS WITH EXPANDING OUTLETS

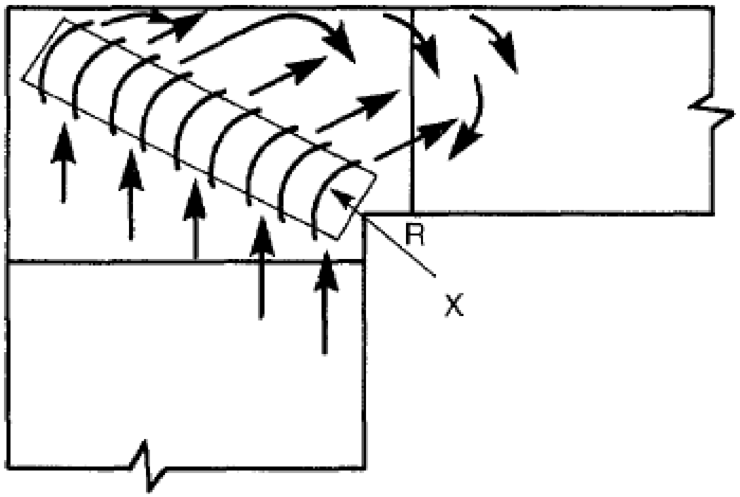


Figure 5-14 TURBULENCE CAUSED BY IMPROPER MOUNTING AND USE OF TURNING VANES

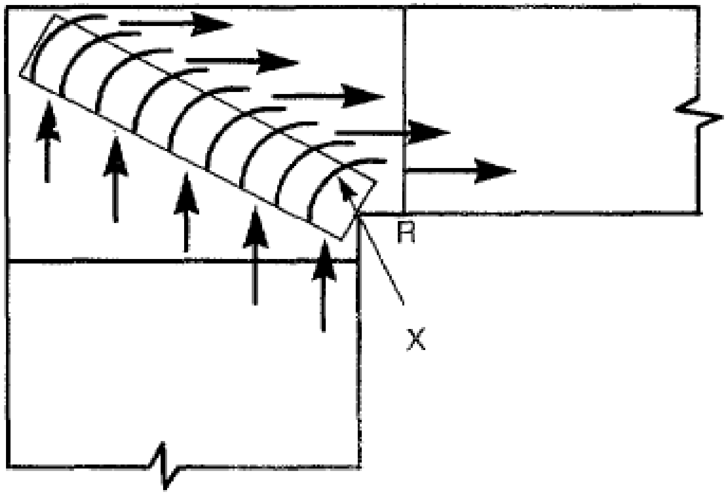


Figure 5-15 PROPER INSTALLATION OF TURNING VANES
(Vanes do not have "trailing edges," but have been moved in the vane runner to remain tangent to the airstream.)

Individual Returns

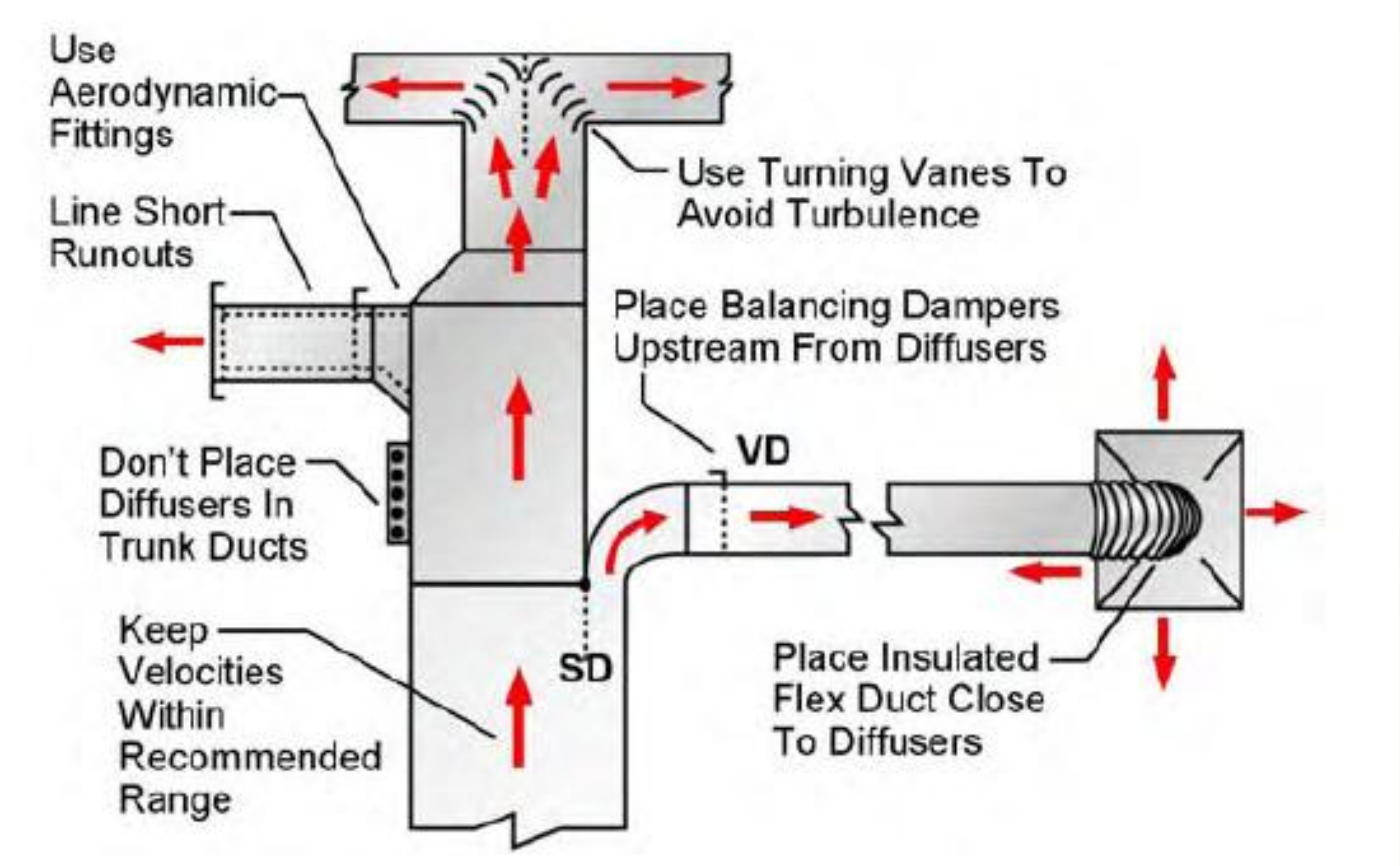
Advantages	Disadvantages
• Good sound attenuation inherent to system of branch ductwork; quieter operation • Facilitates good air flow within individual rooms, even with doors closed • Provides better privacy, especially in bedrooms, since doors need not be undercut • Small branch ductwork easily concealed within joist and stud spaces • Small grilles less conspicuous	• Requires a second duct system, usually trunk and branch similar to supply system • Installation more complex usually requiring a separate layout • Usually more costly to install

Classification of duct systems

- Low pressure systems: Velocity ≤ 10 m/s, static pressure ≤ 5 $cmH_2O(g)$
- Medium pressure systems: Velocity ≤ 10 m/s, static pressure ≤ 15 cm H_2O (g)
- High pressure systems: Velocity > 10 m/s, static pressure $15<p \leq 25$ $cmH_2O(g)$

- Recommended air velocities depend mainly on the application and the noise criteria. Typical recommended velocities are:
  - Residences: 3 m/s to 5 m/s
  - Theatres: 4 to 6.5 m/s
  - Restaurants: 7.5 m/s to 10 m/s
- High velocities in the ducts results in:
  - Smaller ducts and hence, lower initial cost and lower space requirement
  - Higher pressure drop and hence larger fan power consumption
  - Increased noise and hence a need for noise attenuation

Limit Noise Creation..



- If nothing is specified, then a velocity of **5 to 8 m/s** is used for main ducts and a velocity of **4 to 6 m/s** is used for the branches. The allowable air velocities can be as high as 30 m/s in ships and aircrafts to reduce the space requirement.

Bernoulli Eq.

$$\frac{v^2}{2} + \int \frac{dP}{\rho} + gz = const.$$ (N*m/kg)

Where υ = streamline (local) velocity , m/s
p = absolute pressure , Pa N/m²
ρ = density , kg/m³
g = acceleration caused by gravity , m/s²
z = elevation , m

- For constant density

$$\frac{v^2}{2} + \frac{P}{\rho} + gz = const.$$ (N*m/kg)

- For Actual Duct

$$\frac{\rho_1 V_1^2}{2} + P_1 + g\rho_1 z_1 = \frac{\rho_2 V_2^2}{2} + P_2 + g\rho_2 z_2 + \Delta p_{t,1-2}$$ ---- (1)

where V = average duct velocity m/s

$\Delta P_{t,1-2}$ = total pressure loss caused by friction and dynamic losses between sections 1 and 2 ,Pa

On the left side of Equation (1), add and subtract P_{z1} ; on the right side,add and subtract P_{z2} , Where P_{z1} and P_{z2} are the values of atmospheric air at heights z_1 and z_2 . Thus,

$$\frac{\rho_1 V_1^2}{2} + P_1 + (p_{z1} - p_{z1}) + g\rho_1 z_1$$
$$= \frac{\rho_2 V_2^2}{2} + P_2 + (p_{z2} - p_{z2}) + g\rho_2 z_2 + \Delta p_{t,1-2}$$

Atmospheric pressure at any elevation (p_{z1} and p_{z2}) expressed in terms of the atmospheric pressure p_a at the same datum elevation is given by

$$p_{z1} = p_a - g\rho_a z_1$$
$$p_{z2} = p_a - g\rho_a z_2$$

$$\Delta p_{t,1-2} = (p_{s,1} + \frac{\rho V_1^2}{2}) - (p_{s,2} + \frac{\rho V_2^2}{2}) + g(\rho_a - \rho)(z_2 - z_1)$$

$$\Delta p_{t,1\text{-}2} = \Delta p_t + \Delta p_{se}$$

$$\Delta p_t = \Delta p_{t,1\text{-}2} + \Delta p_{se} \qquad \text{------ (2)}$$

where $p_{s,1}$ = static pressure , gage at elevation z_1 , Pa

$p_{s,2}$ = static pressure , gage at elevation z_2 , Pa

V_1 = average velocity at section 1 , m/s

V_2 = average velocity at section 2 , m/s

ρ_a = density of ambient air , kg/m^3

ρ = density of air or gas in duct , kg/m^3

Δp_{se} = thermal gravity effect , Pa

Δp_t = total pressure change between sections 1 and 2 , Pa

$\Delta p_{t,1\text{-}2}$ = total pressure loss caused by friction and dynamic losses between sections 1 and 2 , Pa

System Analysis

The total pressure change caused by friction , fittings , equeipment, and net thermal gravity effect (stack effect) for each section of a duct system is calculated by the following eqution :

$$\Delta p_{t_i} = \Delta p_{f_i} + \sum_{j=1}^{m} \Delta p_{ij} + \sum_{k=1}^{n} \Delta p_{ik} - \sum_{r=1}^{\lambda} \Delta p_{se_{ir}}$$

for i = 1,2,……., $n_{up} + n_{dn}$

where Δp_{ti} = net total pressure change for i-section , Pa

Δp_{fi} = pressure loss due to friction for i-section , Pa

Δp_{ij} = total pressure loss due to j-fittings, including fan system effect (FSE) , for i-section , Pa

Δp_{ik} = pressure loss due to k-equipment for i-section , Pa

$\Delta p_{se_{ir}}$ = thermal gravity effect due to r-stacks for i-section , Pa

m = number of fittings within i-section

n = number of equipment within i-section

λ = number of stacks within i-section

n_{up} = number of duct sections upstream of fan (exhaust/return air subsystems)

n_{dn} = number of duct sections downstream of fan (supply air subsystems)

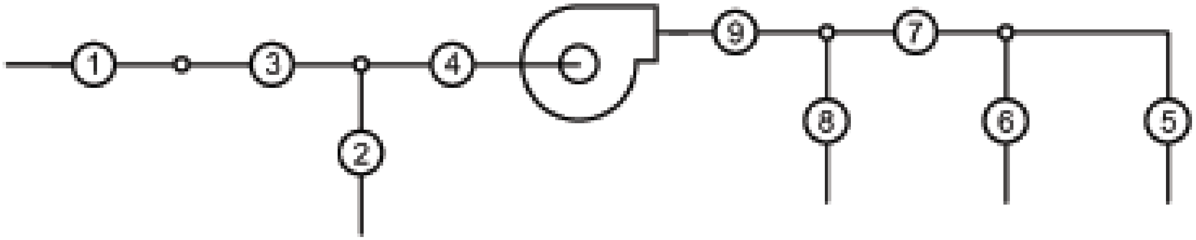


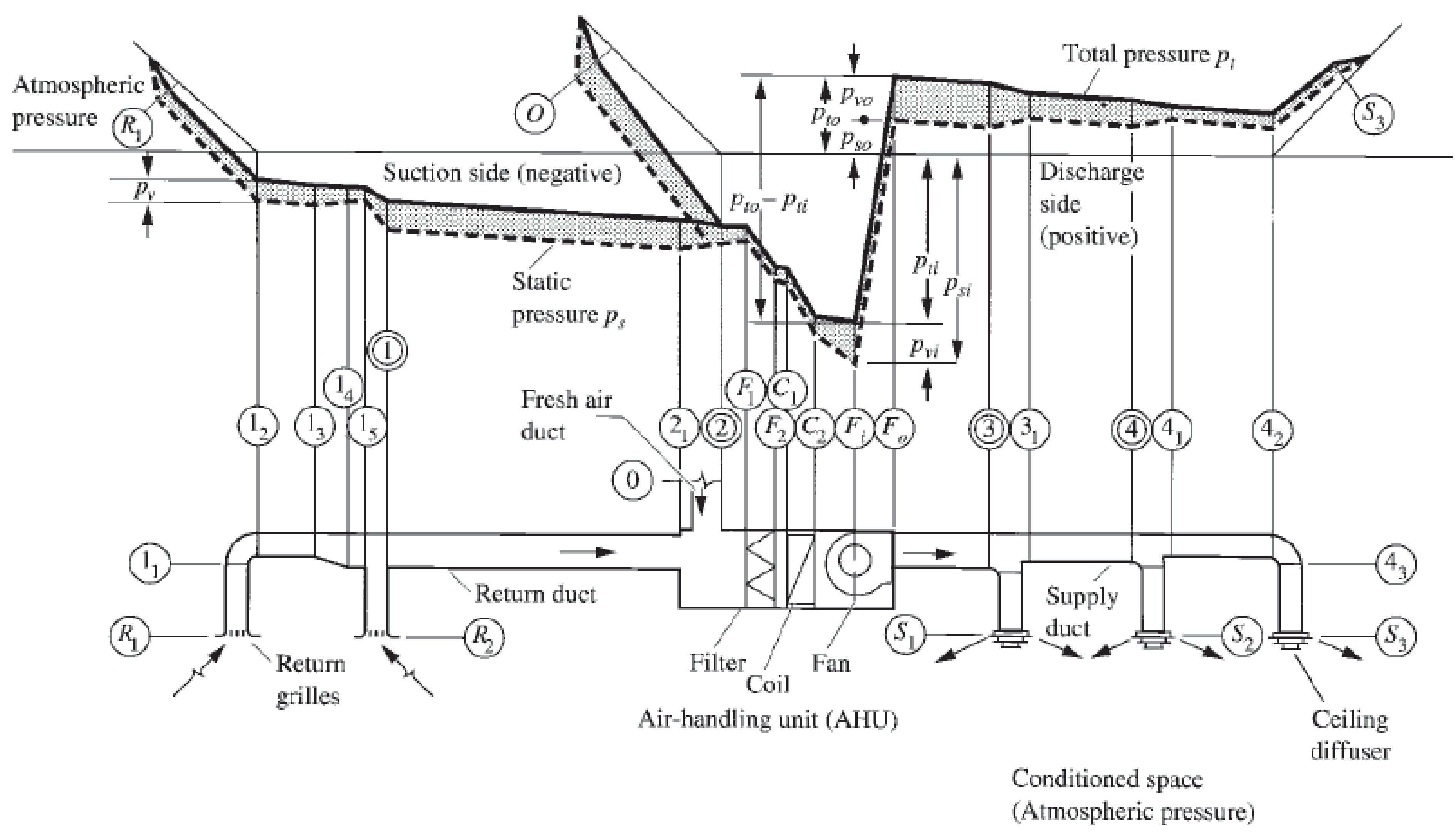
Fig. 4 Illustrative 6-Path, 9-Section System

$$\begin{cases} P_t = \Delta p_1 + \Delta p_3 + \Delta p_4 + \Delta p_9 + \Delta p_7 + \Delta p_5 \\ P_t = \Delta p_1 + \Delta p_3 + \Delta p_4 + \Delta p_9 + \Delta p_7 + \Delta p_6 \\ P_t = \Delta p_1 + \Delta p_3 + \Delta p_4 + \Delta p_9 + \Delta p_8 \\ P_t = \Delta p_2 + \Delta p_4 + \Delta p_9 + \Delta p_7 + \Delta p_5 \\ P_t = \Delta p_2 + \Delta p_4 + \Delta p_9 + \Delta p_7 + \Delta p_6 \\ P_t = \Delta p_2 + \Delta p_4 + \Delta p_9 + \Delta p_8 \end{cases}$$

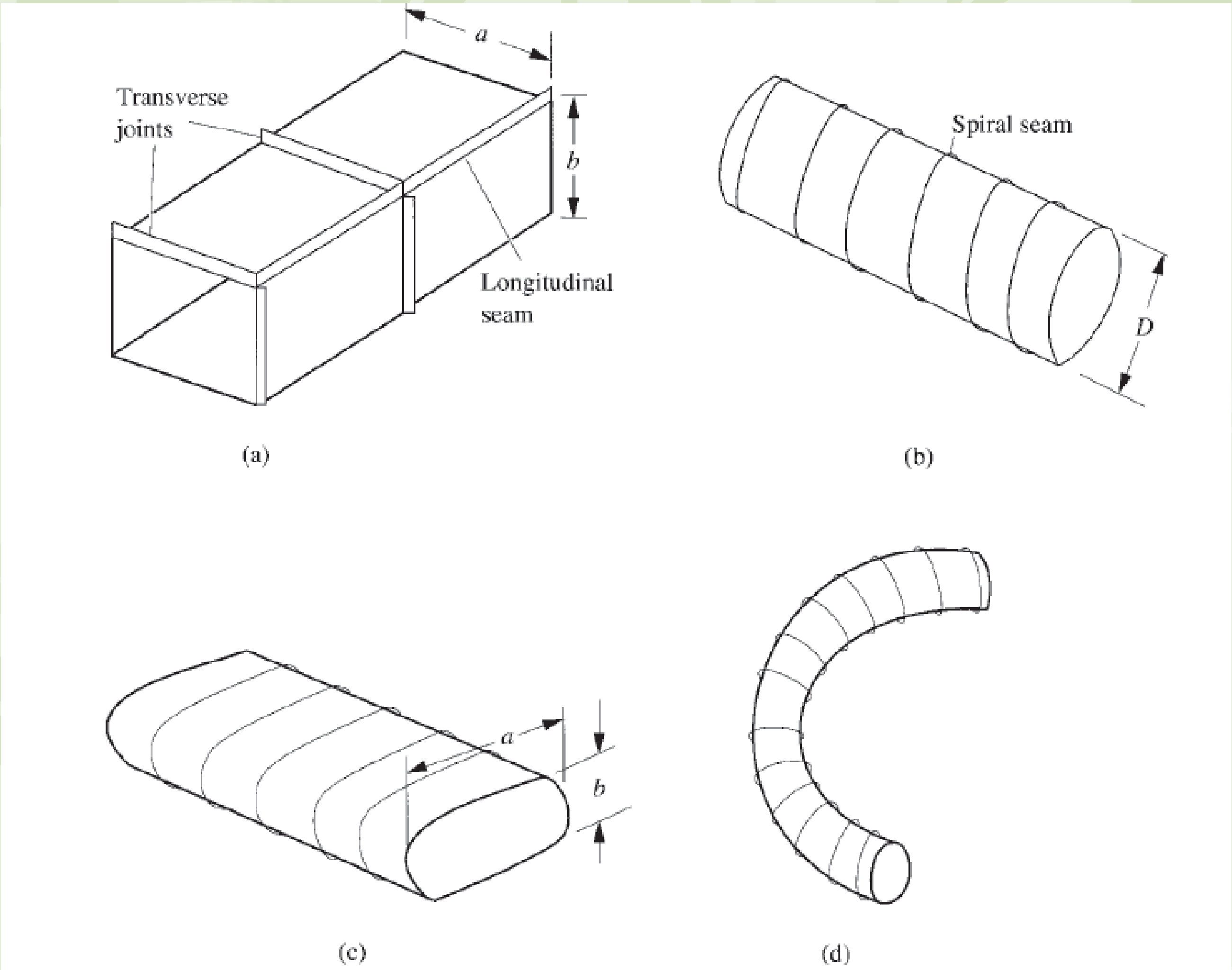
Type of air duct

- *Supply duct*. Conditioned air is supplied to the conditioned space.
- *Return duct*. Space air is returned (1) to the fan room where the air-handling unit is installed or to the packaged unit.
- *Outdoor air duct*. Outdoor air is transported to the air-handling unit, to the fan room, or to the space directly.
- *Exhaust duct*. Space air or contaminated air is exhausted from the space, equipment, fan room, or localized area.

Pressure characteristics of a fan duct system



Various types of air duct: (*a*) rectangular duct; (*b*) round duct with spiral seam; (*c*) flat oval duct; (*d*) flexible duct.



- For the space available between the structural beam and the ceiling in a building, rectangular ducts have the greatest cross-sectional area. They are less rigid than round ducts and are more easily fabricated on-site. Unsealed rectangular ducts may have an air leakage from 15 to 20 percent of the supply volume flow rate. Rectangular ducts are usually used in low-pressure systems.
- For a specified cross-sectional area and mean air velocity, a round duct has less fluid resistance against airflow than rectangular and flat oval ducts. Round ducts also have better rigidity and strength.

- Flat oval ducts have a cross-sectional shape between rectangular and round. They share the advantages of both the round and the rectangular duct with less large-scale air turbulence and a small depth of space required during installation. Flat oval ducts are quicker to install and have lower air leakage because of the factory fabrication.
- Flexible ducts are often used to connect the main duct or the diffusers to the terminal box. Their flexibility and ease of removal allow allocation and relocation of the terminal devices. Flexible ducts are usually made of multiple-ply polyester film reinforced by a helical steel wire core or corrugated aluminum spiral strips. The flexible duct should be as short as possible, and its length should be fully extended to minimize flow resistance.



Friction Loss

Noncircular Ducts

A momentum analysis can relate average wall shear stress to pressure drop per unit length for fully developed turbulent flow in a passage of arbitrary shape but uniform longitudinal cross-sectional area. This analysis leads to the definition of **hydraulic diameter :** $$D_h = 4A / P$$

where D_h = hydraulic diameter , mm

A = duct area , mm^2

P = perimeter of cross section , mm

Rectangular Ducts. Huebscher(1948) developed the relationship between rectangular and round ducts that is used to determine size equivalency based on equal flow , resistance , and length .

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

where D_e = circular equivalent of rectangular duct for equal length , fluid resistance , and airflow , mm

a = length one side of duct , mm

b = length adjacent side of duct , mm

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Pressure change in duct

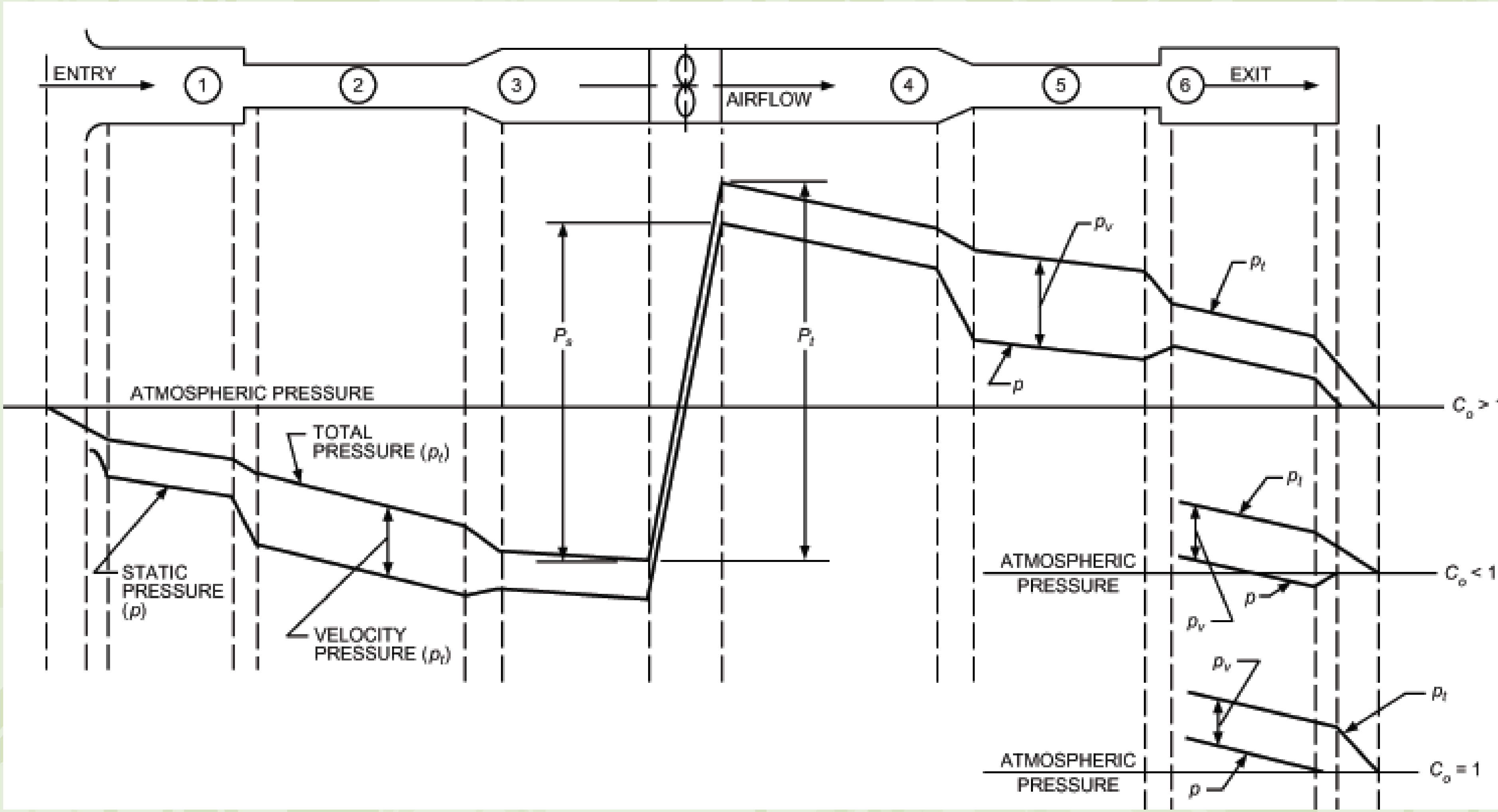

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Table 1 Duct Roughness Factors

Duct Material	Roughness Category	Absolute Roughness ε, ft
Uncoated carbon steel, clean (Moody 1944) (0.05 mm)	Smooth	0.03
PVC plastic pipe (Swim 1982) (0.01 to 0.05 mm)		
Aluminum (Hutchinson 1953) 0.04 to 0.06 mm)		
Galvanized steel, longitudinal seams, 200 mm joints (Griggs et al. 1987) (0.05 to 0.10 mm)	Medium-smooth	0.09
Galvanized steel, continuously rolled, spiral seams, 3000 mm joints (Jones 1979) (0.06 to 0.12 mm)		
Galvanized steel, spiral seam with 1, 2, and 3 ribs, 3600 mm joints (Griggs et al. 1987) (0.09 to 0.12 mm)		
Galvanized steel, longitudinal seams, 760 mm joints (Wright 1945) (0.15 mm)	Average	0.15
Galvanized steel, spiral, corrugated, 3600 mm joints (Kulkarni et al. 2009) (0.74 mm)	Medium-rough	0.9
Fibrous glass duct, rigid		
Fibrous glass duct liner, air side with facing material (Swim 1978) (1.5 mm)		
Flexible duct, fabric and wire, fully extended		
Fibrous glass duct liner, air side spray coated (Swim 1978) (4.6 mm)	Rough	3.0
Flexible duct, metallic (1.2 to 2.1 mm when fully extended)		
Concrete (Moody 1944) (1.3 to 3.0 mm)		

Flat Oval Ducts. To convert round ducts to flat oval sizes , the circular equivalent of a flat oval duct for equal airflow, resistance, and length.

$$D_e = \frac{1.55AR^{0.625}}{P^{0.25}}$$

where AR is the cross-sectional area of flat oval duct defined as

$$AR = (\pi a^2 / 4) + (A - a)$$

and the perimeter P is calculated by

$$P = \pi a + 2 (A - a)$$

where

P = perimeter of flat oval duct , mm

A = major axis flat oval duct , mm

a = minor axis of flat oval duct , mm

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Evaluation of frictional pressure drop in ducts

The Darcy-Weisbach equation is one of the most commonly used equations for estimating frictional pressure drops in internal flows. This equation is given by:

$$\Delta p_f = f\frac{L}{D}\left(\frac{\rho V^2}{2}\right)$$

where f is the dimensionless friction factor, L is the length of the duct and D is the diameter in case of a circular duct and hydraulic diameter in case of a non-circular duct. The friction factor is a function of **Reynolds number**, $Re_D = \rho$ **VD/μ** and the relative surface roughness of the pipe or duct surface in contact with the fluid. **For turbulent flow**, the friction factor can be evaluated using the empirical correlation suggested by Colebrook and White is used, the correlation is given by:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{\varepsilon}{3.7D} + \frac{2.51}{(Re_D)\sqrt{f}}\right]$$

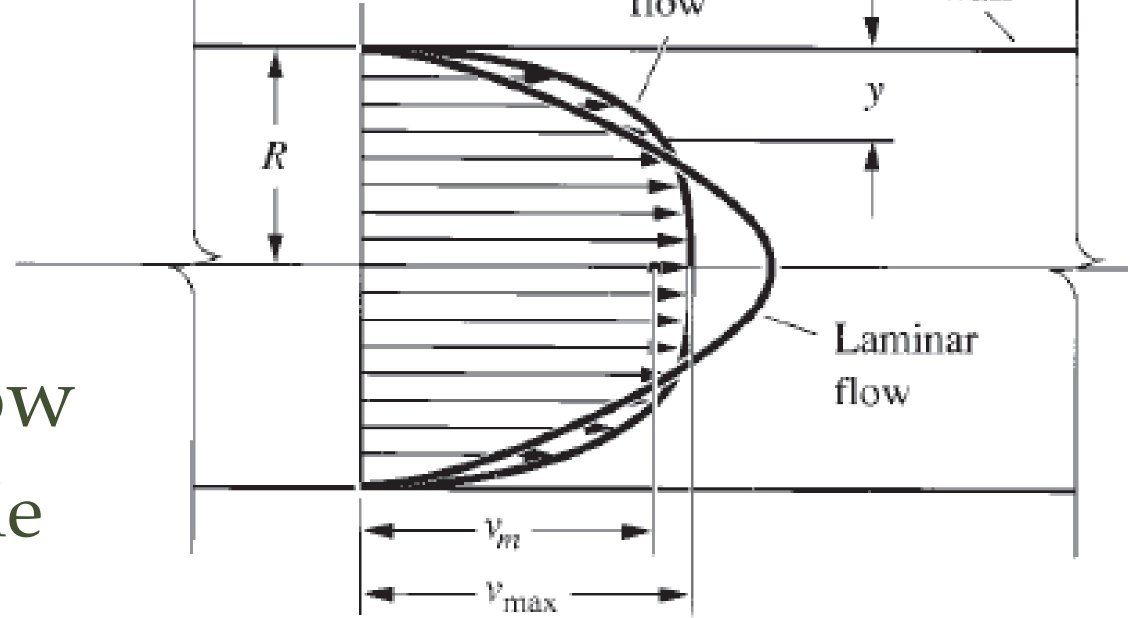
where ε is the average surface roughness of inner duct expressed in same units as the diameter D.

- Reynolds identified two types of fluid flow in 1883 by observing the behavior of a stream of dye in a water flow: laminar flow and turbulent flow. He also discovered that the ratio of inertial to viscous forces is the criterion that distinguishes these two types of fluid flow. This dimensionless parameter is now widely known as *Reynolds number* Re, or

  $\mathrm{Re} = \rho v L/\mu$

  where   density of fluid, (kg /m^3)
  - v velocity of fluid, (m/s)
  - L characteristic length, ft (m)
  - μ viscosity or absolute viscosity, lb / ft s (N s/m^2)

- For Laminar flow
  - Parabolic profile



- For turbulent flow

$$\frac{v}{v_{max}} = \left(\frac{v}{R}\right)^{1/7}$$

where v_{max} = maximum air velocity on centerline of air duct, fpm (m/s)
R = radius of duct, ft (m)

The mean air velocity v_m lies at a distance about $0.33R$ from the duct wall.

Material	Absolute roughness , ε (m)
Galvanized Iron (GI) sheet	0.00015
Concrete	0.0003 to 0.003
Riveted steel	0.0009 to 0.009
Cast Iron (CI)	0.00026
Commercial steel	0.00046

Average surface roughness of commonly used duct materials

- In general in air conditioning ducts, the fluid flow is turbulent. It is seen from the above equation that when the flow is turbulent, the friction factor is a function of Reynolds number, hydraulic diameter and inner surface roughness of the duct material. Table 2 shows absolute roughness values of some of the materials commonly used in air conditioning.
- Of the different materials, the GI sheet material is very widely used for air conditioning ducts. Taking **GI as the reference material** and properties of air at 20°C and 1 atm. pressure, the frictional pressure drop in a circular duct is given by:

$$\Delta p_f = \frac{0.022243\dot{Q}_{air}^{1.852}L}{D^{4.973}} \quad (\mathrm{N/m^2})$$

Dynamic losses in ducts

- Dynamic pressure loss takes place whenever there is a change in either the velocity or direction of airflow due to the use of a variety of bends and fittings in air conditioning ducts. Some of the commonly used fittings are: **enlargements, contractions, elbows, branches, dampers etc**.
- Normally these fittings and bends are rather short in length (< 1 m), the major pressure drop as air flows through these fittings is not because of viscous drag (friction) but due to momentum change.
- In turbulent flows, the dynamic loss is proportional to square of velocity. Hence these are expressed as: $\Delta p_d = K \frac{\rho V^2}{2}$
- where K is the dynamic loss coefficient, which is normally obtained from experiments

- Sometimes, an equivalent length L_{eq} is defined to estimate the dynamic pressure loss through bends and fittings. The dynamic pressure loss is obtained from the equivalent length and the frictional pressure drop equation or chart, i.e.,

$$\Delta p_d = K\left(\frac{\rho V^2}{2}\right) = \left(\frac{f.L_{eq}}{D_{eq}}\right)\left(\frac{\rho V^2}{2}\right)$$

- where f is the friction factor and L_{eq} is the equivalent length

Design Velocity

- In supply main ducts $v_{d,max}$ usually does not exceeds 3000 fpm (15 m/s). Airflow noise must be checked at dampers, elbows, and branch takeoffs to satisfy the indoor *NC* range.
- In buildings with more demanding noise control criteria, such as hotels, apartments, and hospital wards, in supply main ducts usually $v_{d,max}$ 2000 to 2500 fpm (10 to 12.5 m/s), in return main ducts $v_{d,max}$ 1600 fpm (8 m/s), and in branch ducts $v_{d,max}$ 1200 fpm (6 m/s).

Turns, bends or elbows:

- The most common type of bends used in air conditioning ducts are 90^o turns shown in the Fig. A. Fig. B denotes turning vane to reduce turning loss.

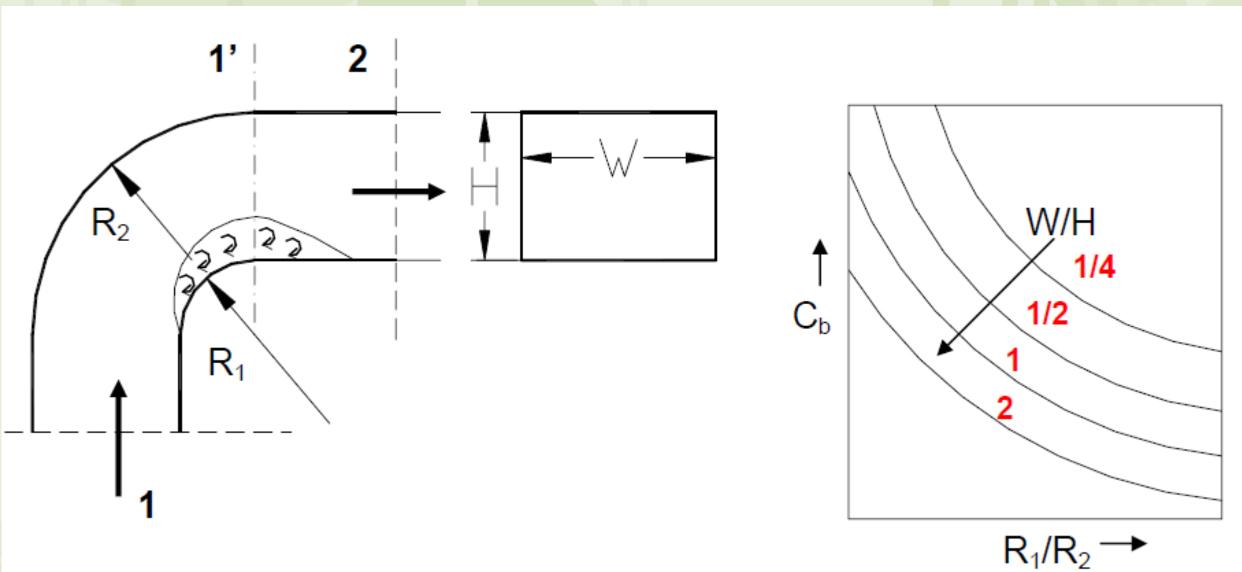


Fig. A, airflow through a 90 ^{0} bend (elbow)

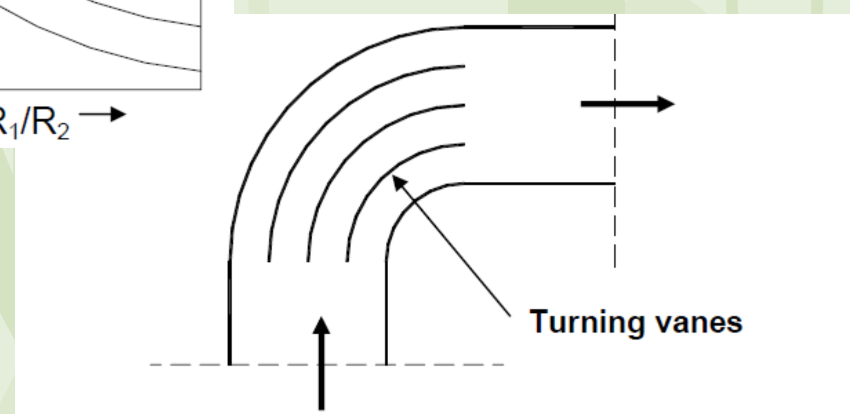


Fig. B, Use of turning vanes in a 90^0 bend (elbow)

Branch take-offs & Branch entries

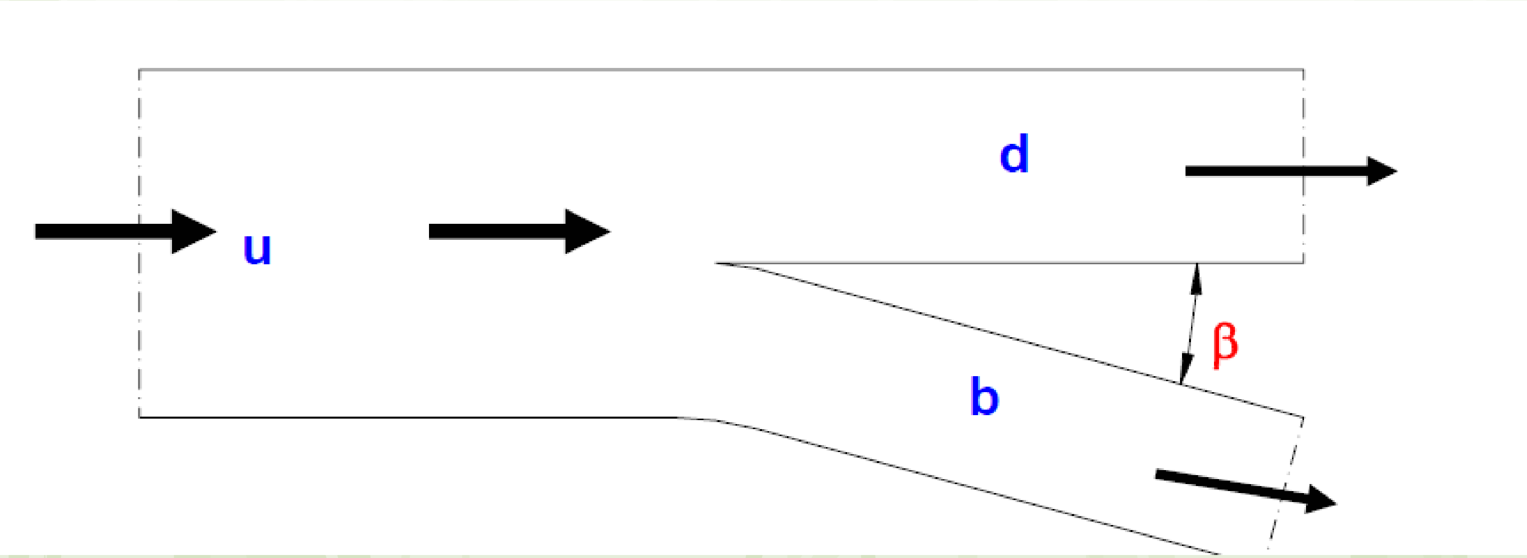


Figure - A branch take-off

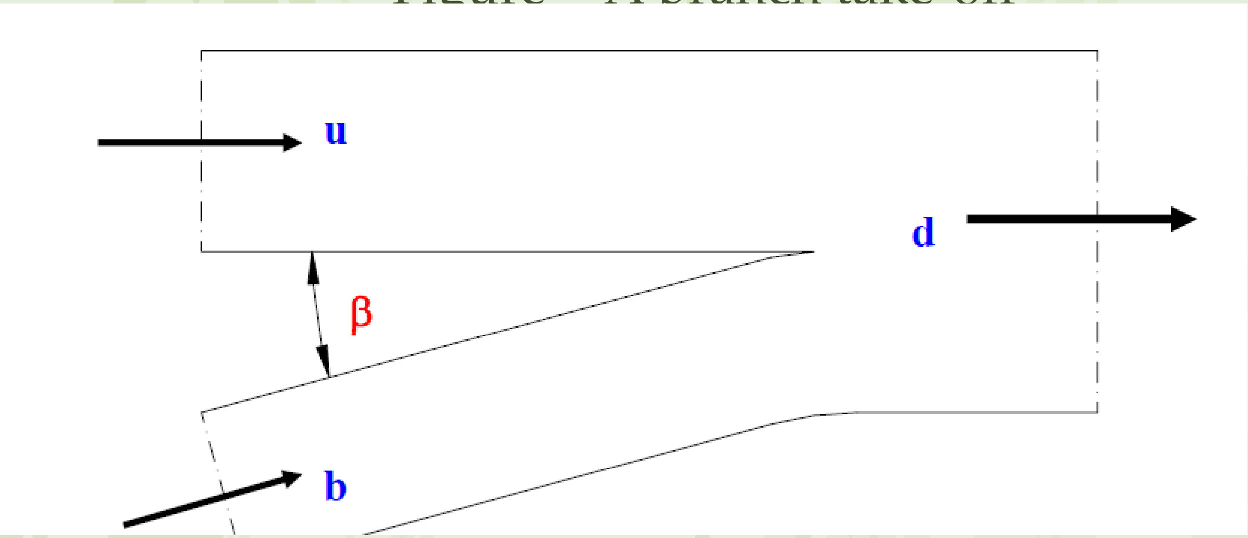


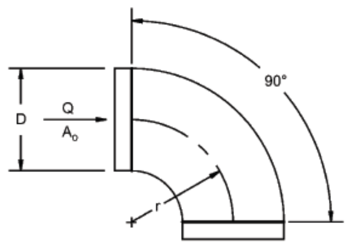
Figure - A branch entry

FITTING LOSS COEFFICIENTS

ROUND FITTINGS

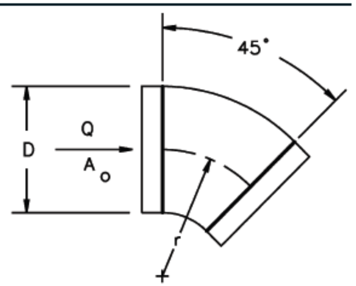
CD3-1 Elbow, Die Stamped, 90 Degree, $r/D = 1.5$

D, mm	75	100	125	150	180	200	230	250
C_o	0.30	0.21	0.16	0.14	0.12	0.11	0.11	0.11



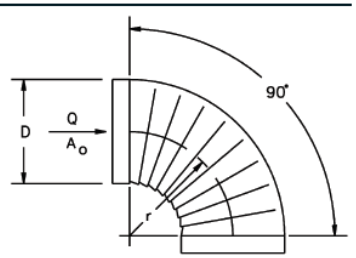
CD3-3 Elbow, Die Stamped, 45 Degree, $r/D = 1.5$

D, mm	75	100	125	150	180	200	230	250
C_o	0.18	0.13	0.10	0.08	0.07	0.07	0.07	0.07



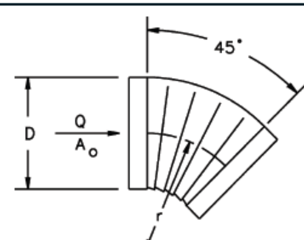
CD3-5 Elbow, Pleated, 90 Degree, $r/D = 1.5$

D, mm	100	150	200	250	300	350	400
C_o	0.57	0.43	0.34	0.28	0.26	0.25	0.25



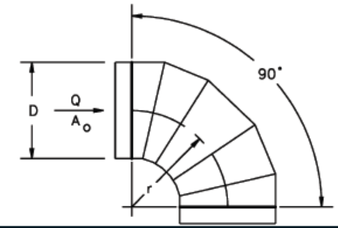
CD3-7 Elbow, Pleated, 45 Degree, $r/D = 1.5$

D, mm	100	150	200	250	300	350	400
C_o	0.34	0.26	0.21	0.17	0.16	0.15	0.15



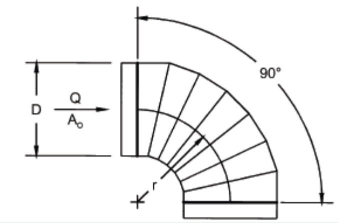
CD3-9 Elbow, 5 Gore, 90 Degree, $r/D = 1.5$

D, mm	75	150	230	300	380	450	530	600	690	750	1500
C_o	0.51	0.28	0.21	0.18	0.16	0.15	0.14	0.13	0.12	0.12	0.12



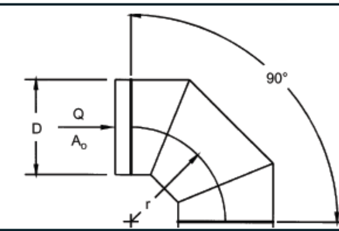
CD3-10 Elbow, 7 Gore, 90 Degree, $r/D = 2.5$

D, mm	75	150	230	300	380	450	690	1500
C_o	0.16	0.12	0.10	0.08	0.07	0.06	0.05	0.03



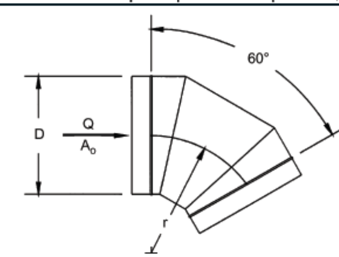
CD3-12 Elbow, 3 Gore, 90 Degree, $r/D = 0.75$ to 2.0

r/D	0.75	1.00	1.50	2.00
C_o	0.54	0.42	0.34	0.33



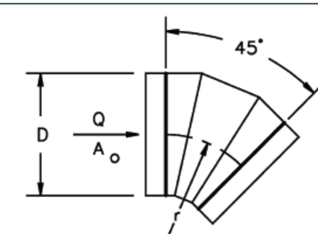
CD3-13 Elbow, 3 Gore, 60 Degree, $r/D = 1.5$

D, mm	75	150	230	300	380	450	530	600	690	750	1500
C_o	0.40	0.21	0.16	0.14	0.12	0.12	0.11	0.10	0.09	0.09	0.09

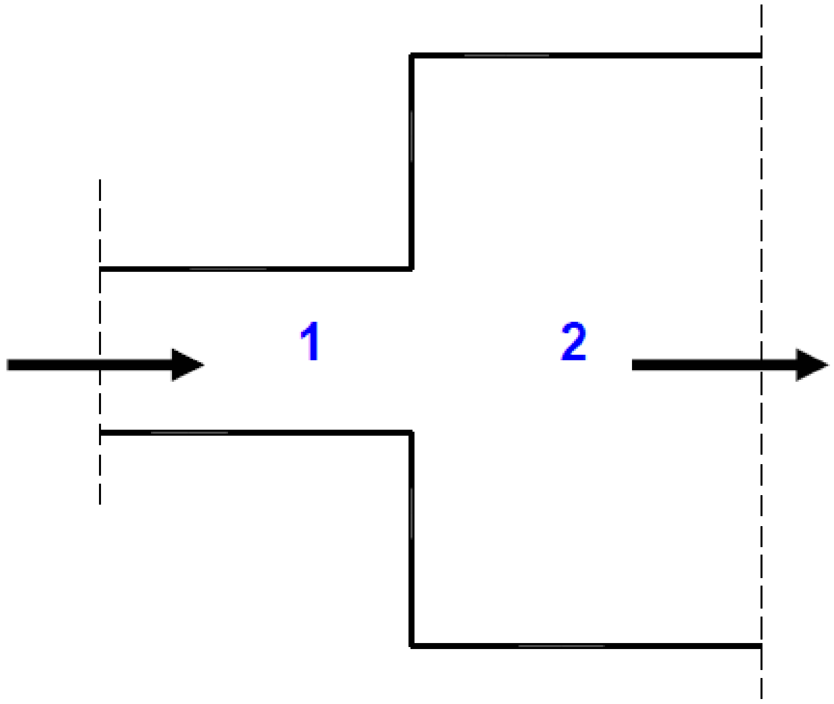


CD3-14 Elbow, 3 Gore, 45 Degree, $r/D = 1.5$

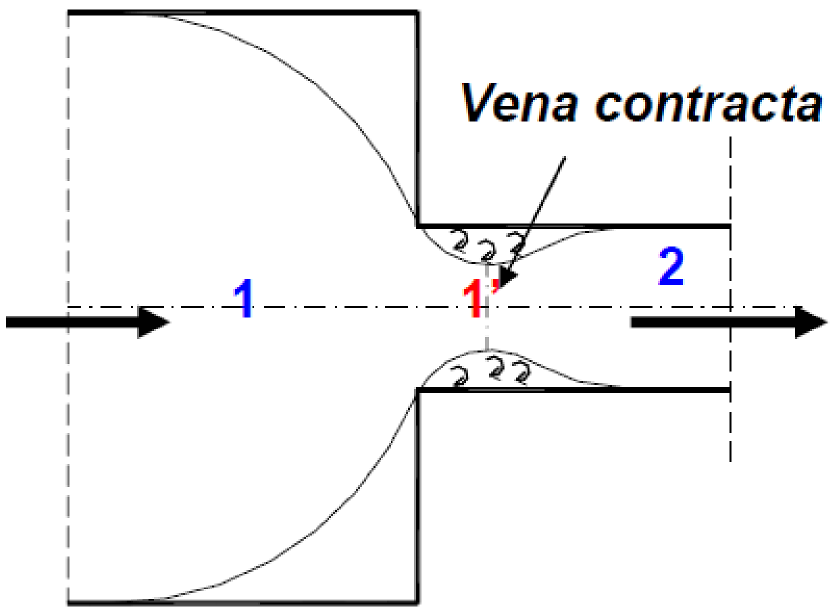
D, mm	75	150	230	300	380	450	530	600	690	750	1500
C_o	0.31	0.17	0.13	0.11	0.11	0.09	0.08	0.08	0.07	0.07	0.07



Sudden Enlargement/Contraction



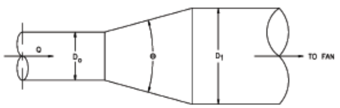
sudden enlargement



sudden contraction

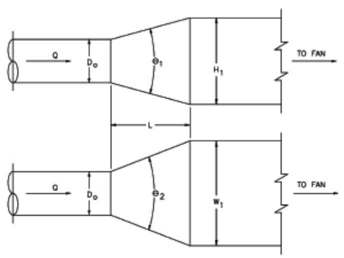
ED4-1 Transition, Round to Round, Exhaust/Return Systems

	C_o Values												
	θ												
A_o/A_1	0	3	5	10	15	20	30	45	60	90	120	150	180
0.063	0.0	0.18	0.18	0.20	0.29	0.38	0.60	0.84	0.88	0.88	0.88	0.88	0.88
0.10	0.0	0.20	0.18	0.20	0.27	0.38	0.59	0.76	0.80	0.83	0.84	0.83	0.83
0.167	0.0	0.18	0.17	0.18	0.25	0.33	0.48	0.66	0.77	0.74	0.73	0.73	0.72
0.25	0.0	0.20	0.17	0.16	0.21	0.30	0.46	0.61	0.68	0.64	0.63	0.62	0.62
0.50	0.0	0.15	0.13	0.11	0.13	0.19	0.32	0.33	0.33	0.32	0.31	0.30	0.30
1.00	0.0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2.00	0.0	0.30	0.26	0.21	0.19	0.19	0.19	0.23	0.27	0.51	0.73	0.90	0.95
4.00	0.0	1.60	1.14	0.75	0.70	0.70	0.70	0.90	1.09	2.78	4.29	5.63	6.53
6.00	0.0	3.89	3.02	1.73	1.58	1.58	1.58	2.12	2.66	6.62	10.01	13.03	15.12
10.00	0.0	11.80	9.30	5.30	5.00	5.00	5.00	6.45	7.90	19.00	28.50	36.70	42.70

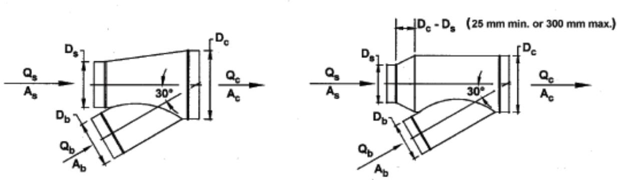


ED4-2 Transition, Round to Rectangular, Exhaust/Return Systems

	C_o Values												
	θ												
A_o/A_1	0	3	5	10	15	20	30	45	60	90	120	150	180
0.063	0.0	0.17	0.19	0.30	0.46	0.53	0.64	0.77	0.88	0.95	0.95	0.94	0.93
0.10	0.0	0.17	0.19	0.30	0.45	0.53	0.64	0.75	0.84	0.89	0.89	0.89	0.88
0.167	0.0	0.18	0.19	0.30	0.44	0.53	0.63	0.72	0.78	0.79	0.79	0.79	0.79
0.25	0.0	0.16	0.18	0.25	0.36	0.45	0.52	0.58	0.62	0.64	0.64	0.64	0.64
0.50	0.0	0.14	0.14	0.15	0.22	0.25	0.30	0.33	0.33	0.33	0.32	0.31	0.30
1.00	0.0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2.00	0.0	0.30	0.27	0.26	0.28	0.25	0.19	0.23	0.27	0.52	0.75	0.91	0.95
4.00	0.0	1.60	1.14	0.84	0.85	0.86	0.76	0.90	1.09	2.78	4.30	5.65	6.55
6.00	0.0	3.89	3.04	1.84	1.77	1.78	1.73	2.18	2.67	6.67	10.07	13.09	15.18
10.00	0.0	11.80	9.31	5.40	5.18	5.15	5.05	6.44	7.94	19.06	28.55	36.75	42.75



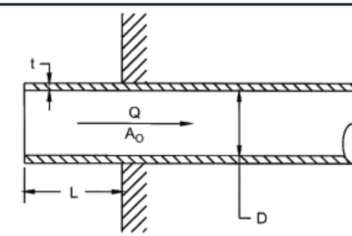
ED5-1 Wye, 30 Degree, Converging



		C_b Values								
		Q_b/Q_c								
A_s/A_c	A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	0.1	–13.25	–1.80	0.01	0.55	0.75	0.84	0.88	0.91	0.97
	0.2	–56.10	–10.12	–2.80	–0.63	0.19	0.53	0.69	0.75	0.78
	0.3	–127.28	–23.81	–7.31	–2.44	–0.59	0.19	0.52	0.66	0.70
	0.4	–226.84	–42.88	–13.55	–4.89	–1.61	–0.22	0.38	0.62	0.68
	0.5	–354.79	–67.34	–21.52	–7.98	–2.86	–0.69	0.24	0.61	0.70
	0.6	–511.13	–97.21	–31.22	–11.73	–4.35	–1.23	0.11	0.64	0.77
	0.7	–695.87	–132.47	–42.66	–16.13	–6.08	–1.84	0.00	0.71	0.89
	0.8	–909.01	–173.14	–55.83	–21.17	–8.05	–2.51	–0.12	0.82	1.05
	0.9	–1151.	–219.20	–70.73	–26.87	–10.27	–3.25	–0.22	0.97	1.26
	1.0	–1420.	–270.66	–87.36	–33.21	–12.72	–4.05	–0.31	1.15	1.51

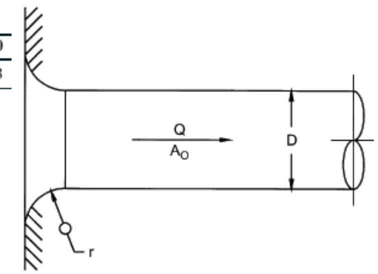
ED1-1 Duct Mounted in Wall

	C_o Values								
	L/D								
t/D	0.0	0.002	0.01	0.05	0.10	0.20	0.30	0.50	10.0
0.00	0.50	0.57	0.68	0.80	0.86	0.92	0.97	1.00	1.00
0.02	0.50	0.51	0.52	0.55	0.60	0.66	0.69	0.72	0.72
0.05	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
10.00	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50



ED1-3 Bellmouth, with Wall

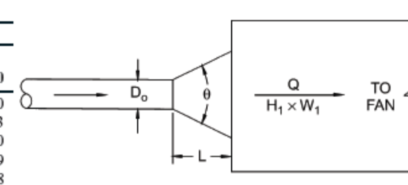
r/D	0.0	0.01	0.02	0.03	0.04	0.05	0.06	0.08	0.10	0.12	0.16	0.20	10.0
C_o	0.50	0.44	0.37	0.31	0.26	0.22	0.20	0.15	0.12	0.09	0.06	0.03	0.03



ED2-1 Conical Diffuser, Round to Plenum, Exhaust/Return Systems

	C_o Values										
	L/D_o										
A_1/A_o	0.5	1.0	2.0	3.0	4.0	5.0	6.0	8.0	10.0	12.0	14.0
1.0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
1.5	0.03	0.02	0.03	0.03	0.04	0.05	0.06	0.08	0.10	0.11	0.13
2.0	0.08	0.06	0.04	0.04	0.04	0.05	0.05	0.06	0.08	0.09	0.10
2.5	0.13	0.09	0.06	0.06	0.06	0.06	0.06	0.06	0.07	0.08	0.09
3.0	0.17	0.12	0.09	0.07	0.07	0.06	0.06	0.07	0.07	0.08	0.08
4.0	0.23	0.17	0.12	0.10	0.09	0.08	0.08	0.08	0.08	0.08	0.08
6.0	0.30	0.22	0.16	0.13	0.12	0.10	0.10	0.09	0.09	0.09	0.08
8.0	0.34	0.26	0.18	0.15	0.13	0.12	0.11	0.10	0.09	0.09	0.09
10.0	0.36	0.28	0.20	0.16	0.14	0.13	0.12	0.11	0.10	0.09	0.09
14.0	0.39	0.30	0.22	0.18	0.16	0.14	0.13	0.12	0.10	0.10	0.10
20.0	0.41	0.32	0.24	0.20	0.17	0.15	0.14	0.12	0.11	0.11	0.10
1000.0	0.41	0.32	0.24	0.20	0.17	0.15	0.14	0.12	0.11	0.11	0.10

	Optimum Angle θ, degrees										
A_1/A_o	0.5	1.0	2.0	3.0	4.0	5.0	6.0	8.0	10.0	12.0	14.0
1.0	0	0	0	0	0	0	0	0	0	0	0
1.5	17	10	6.5	4.5	3.5	2.8	2.2	1.7	1.2	1.0	0.8
2.0	21	14	8.5	6.2	5.0	4.3	3.8	3.0	2.3	2.0	1.6
2.5	25	16	10	7.4	6.0	5.4	4.8	4.0	3.5	3.0	2.5
3.0	27	17	11	8.5	7.0	6.1	5.6	4.8	4.2	3.8	3.2
4.0	29	20	13	9.8	8.0	7.2	6.6	5.8	5.2	4.8	4.4
6.0	31	21	14	11	9.4	8.2	7.4	6.2	5.6	5.2	4.7
8.0	32	22	15	12	10	8.8	8.0	6.6	5.8	5.4	5.0
33	33	23	15	12	11	9.4	8.4	7.0	6.2	5.5	5.2
14.0	33	24	16	13	11	9.6	8.7	7.3	6.3	5.6	5.4
20.0	34	24	16	13	11	9.8	9.0	7.5	6.5	6.0	5.6
1000.0	34	24	16	13	11	9.8	9.0	7.5	6.5	6.0	5.6



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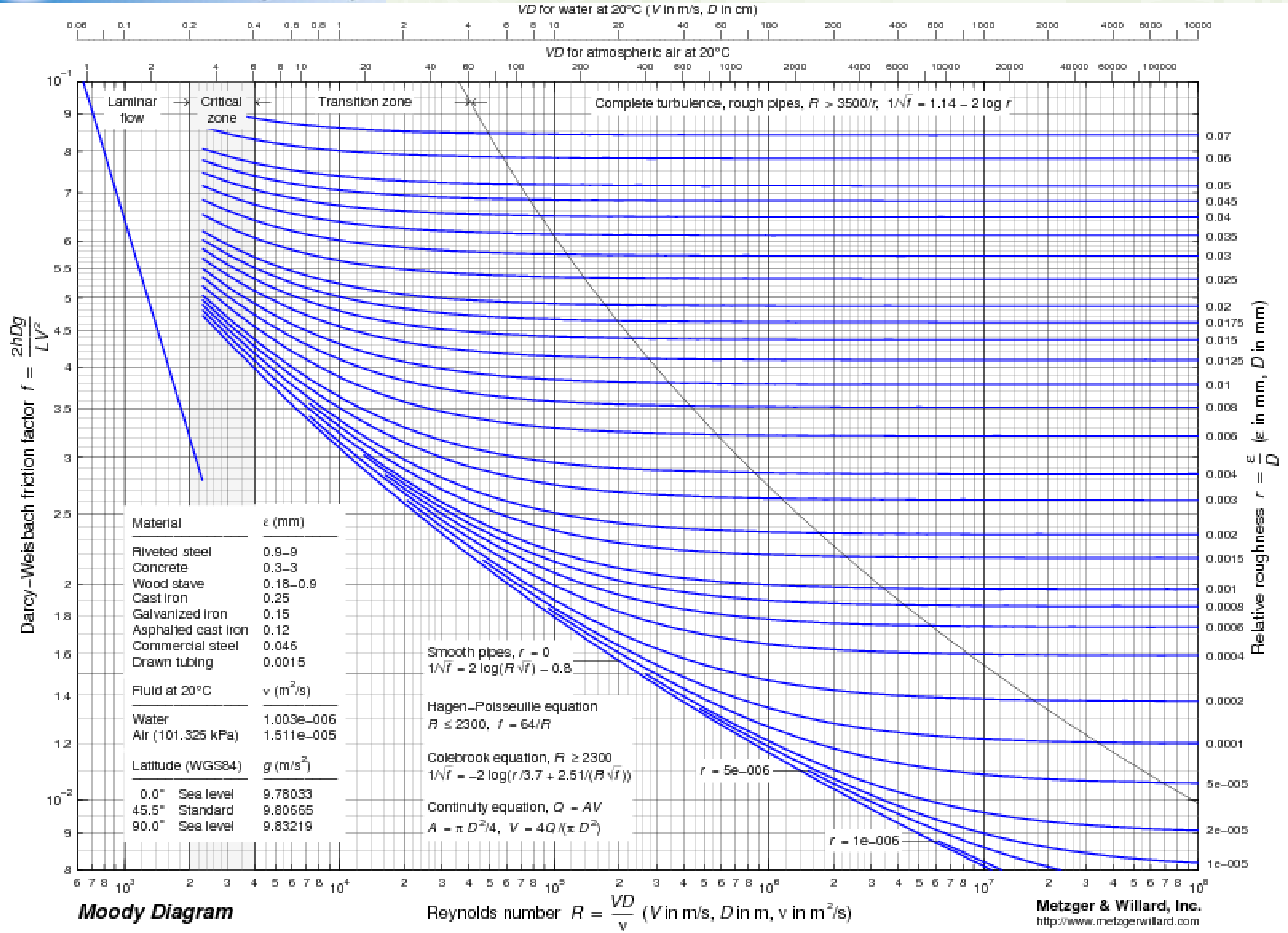
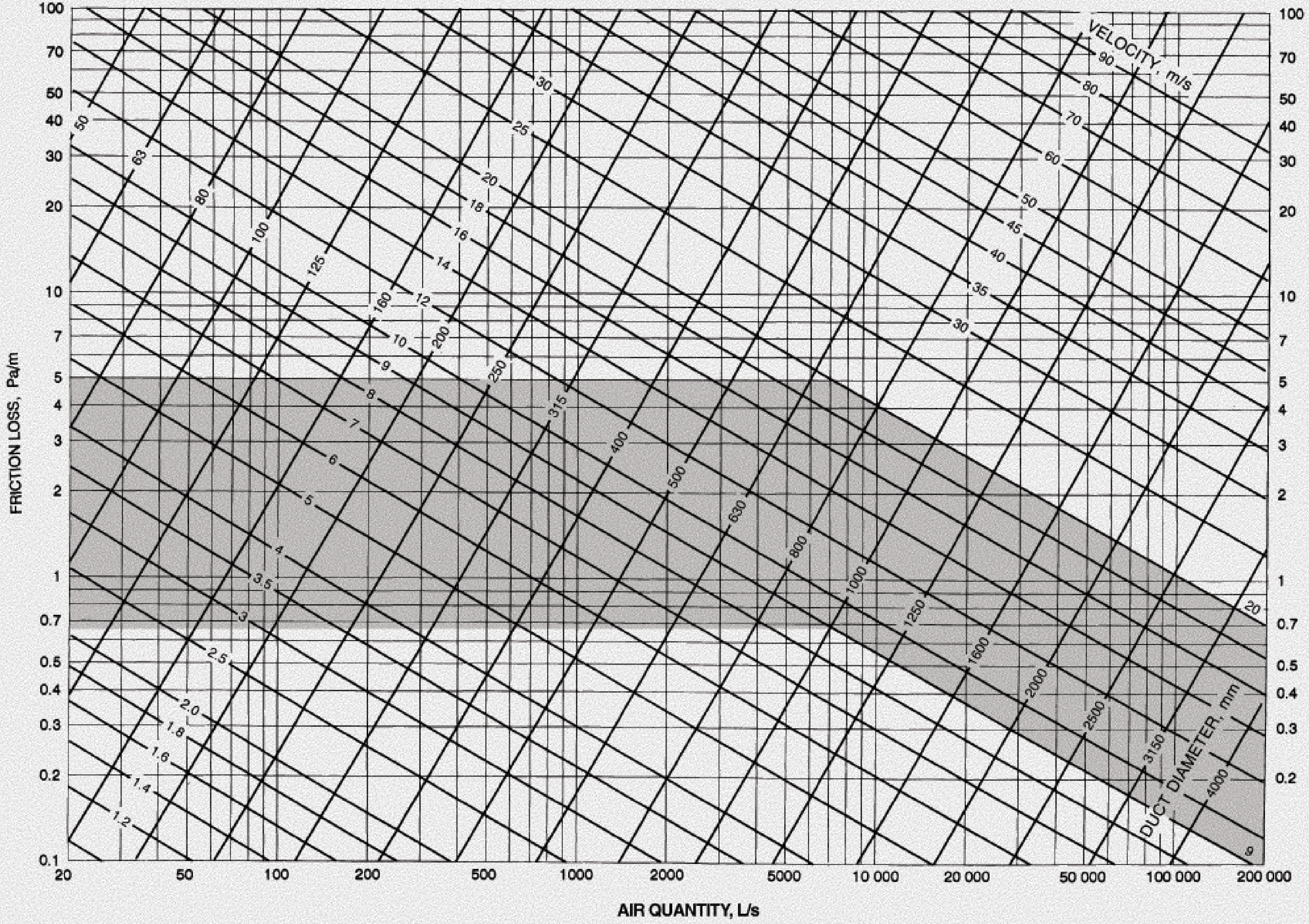
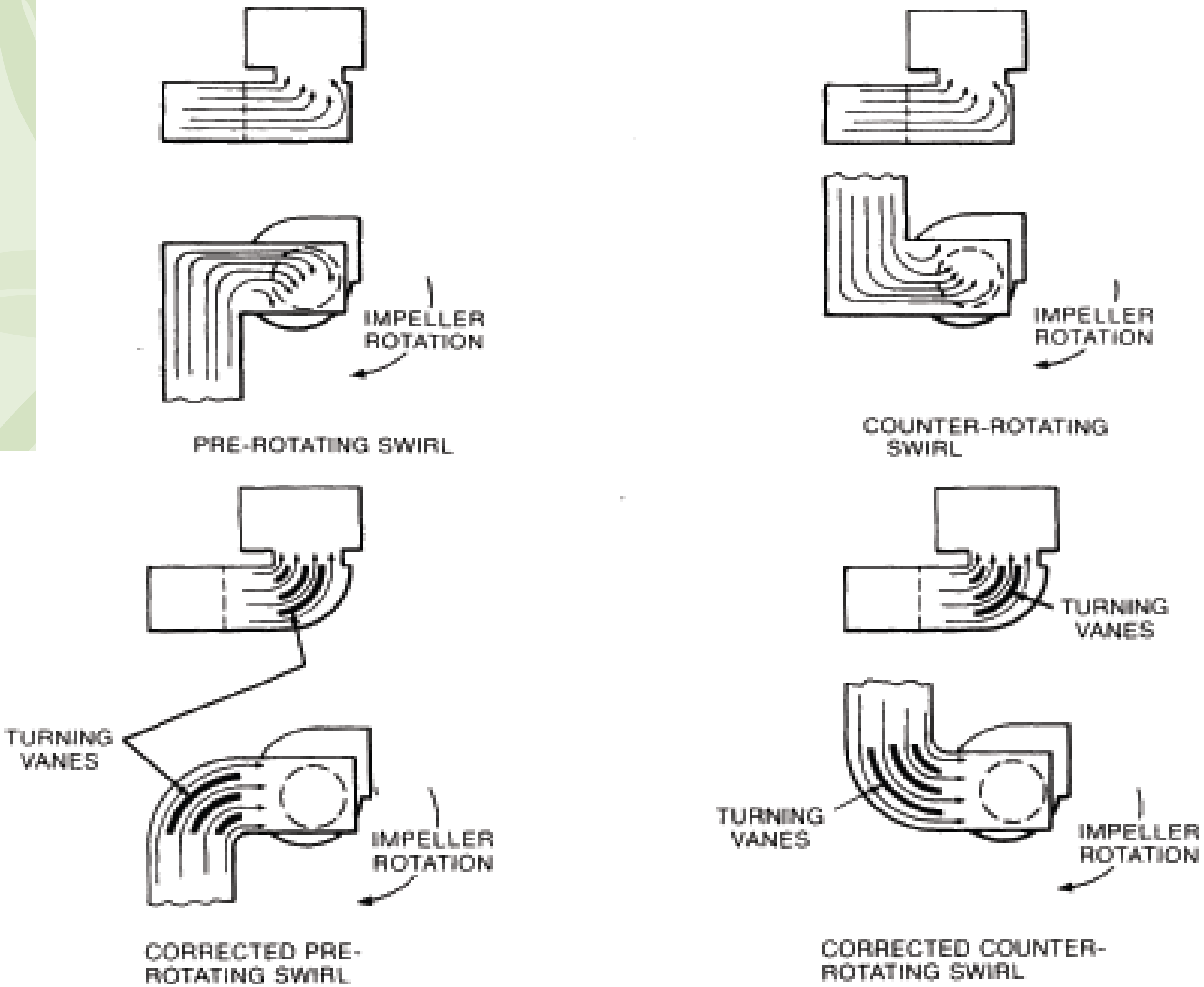
National Chiao Tung University

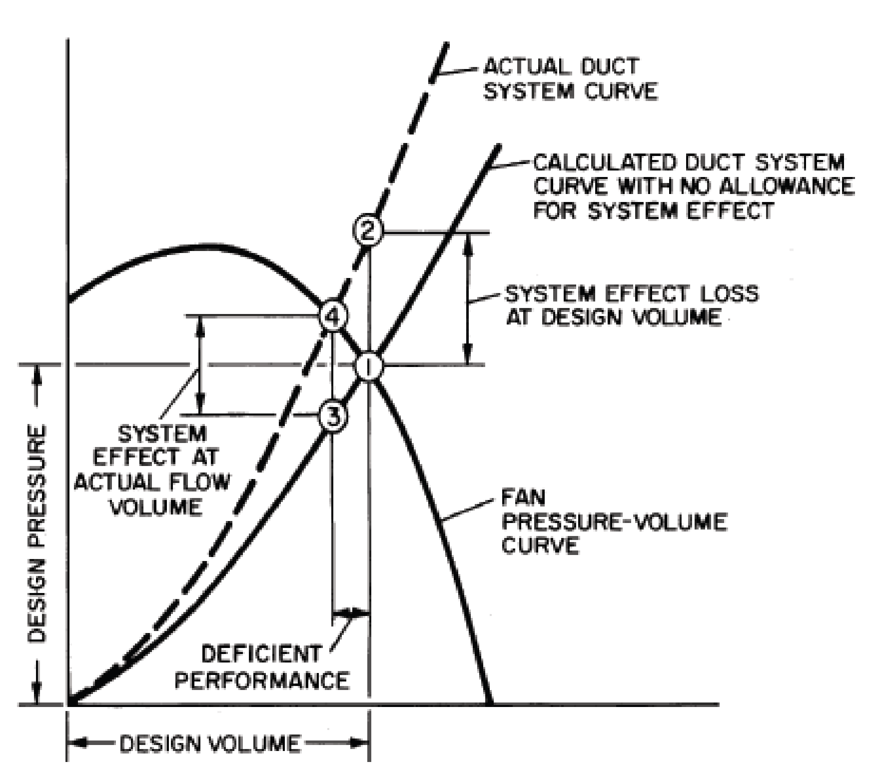


Friction Chart for Round Duct (ρ= 1.2 kg/m^3 and ε= 0.09 mm)

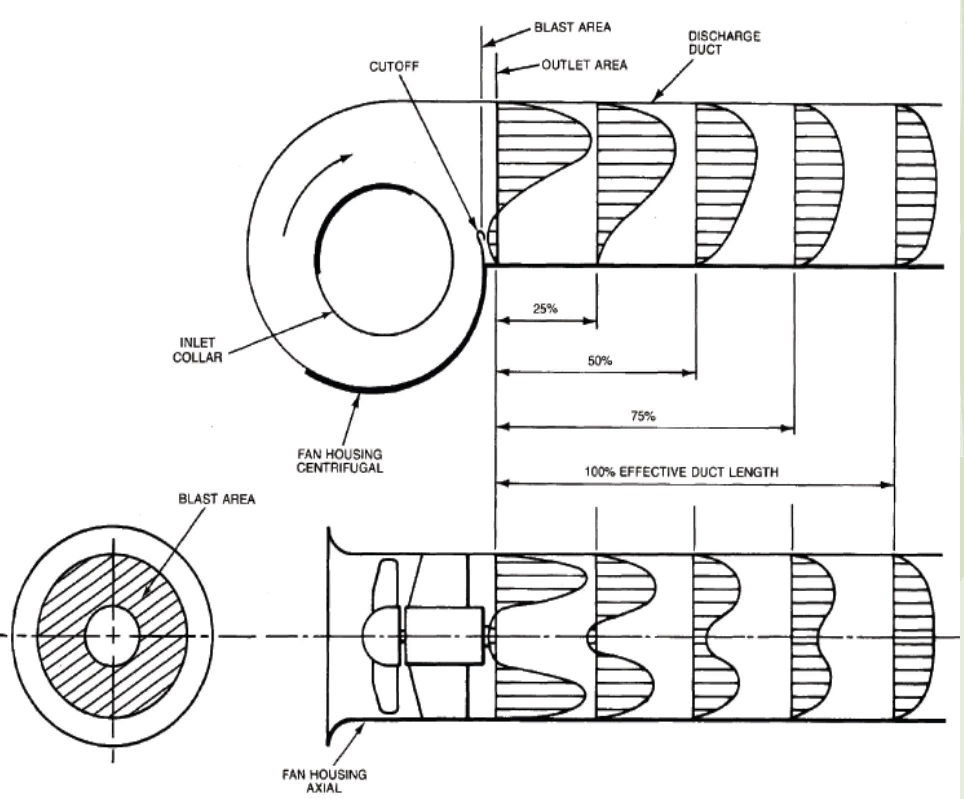




CORRECTIONS FOR INLET SPIN

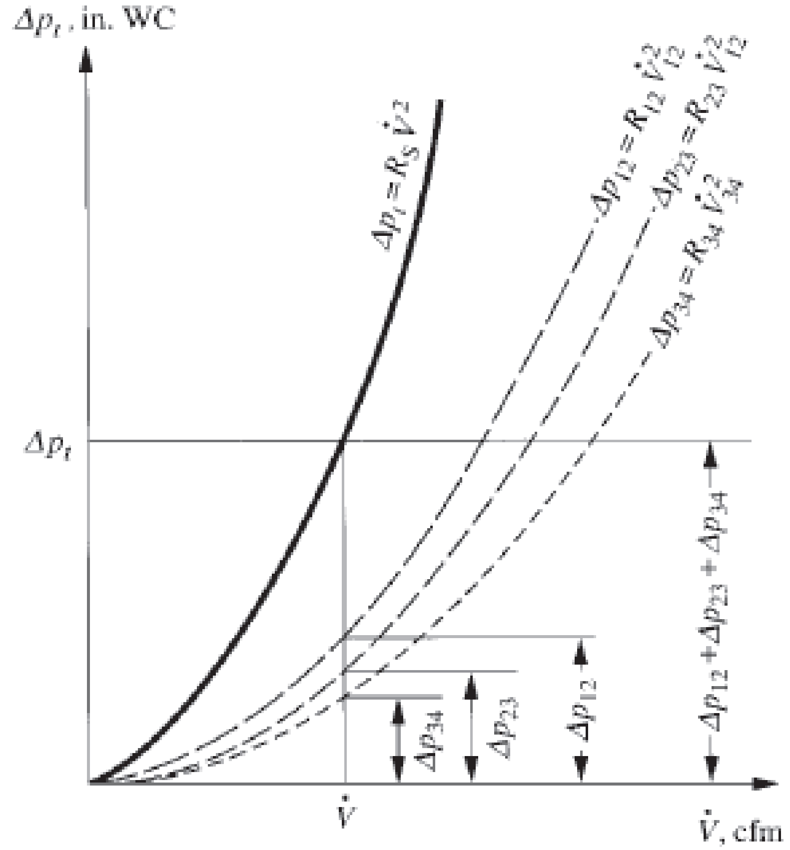


Deficient System Performance with System Effect Ignored (↑)



(↑)Establishment of Uniform Velocity Profile in Straight Fan Outlet Duct(Adapted by permission AMCA Publication201)

Flow Resistances Connected in Series



$$\Delta p_t = \Delta p_{12} + \Delta p_{23} + \cdot \cdot \cdot + \Delta p_n$$

$$= (R_{12} + R_{23} + \cdot \cdot \cdot + R_n)\dot{V}^2 = R_s \dot{V}^2 \qquad (17.59)$$

where $R_{12}, R_{23}, \ldots, R_n$ = flow resistances of duct sections 1, 2, . . . , n, in. WC/(cfm)^2 (Pa·s^2/m^6)

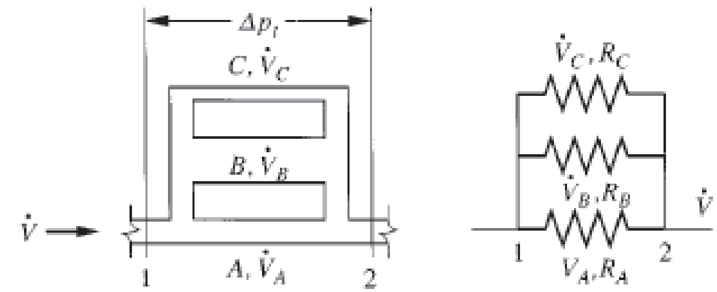
$\Delta p_{12}, \Delta p_{23}, \ldots, \Delta p_n$ = total pressure losses across flow resistances $R_{12}, R_{23}, \ldots, R_n$, in. WC (Pa)

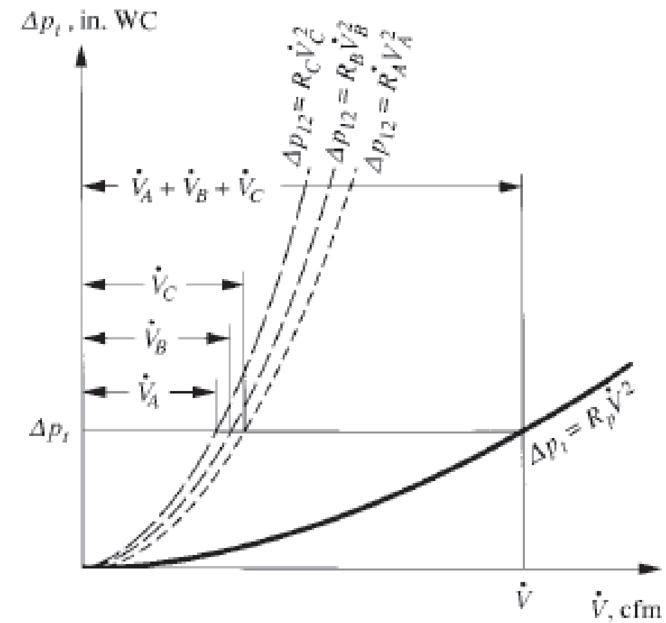
$\dot{V}_{12}, \dot{V}_{23}, \ldots, \dot{V}_n$ = volume flow rates of air flowing through resistances $R_{12}, R_{23}, \ldots, R_n$, cfm (m^3/s)

The flow resistance of the duct system R_s is equal to the sum of individual flow resistances of duct sections connected in series, or

$$R_s = R_{12} + R_{23} + \cdot \cdot \cdot + R_n \qquad (17.60)$$

Flow Resistances Connected in Parallel





$$\dot{V} = \dot{V}_A + \dot{V}_B + \cdots + \dot{V}_n$$

$$= \sqrt{\frac{\Delta p_{12}}{R_A}} + \sqrt{\frac{\Delta p_{12}}{R_B}} + \cdots + \sqrt{\frac{\Delta p_{12}}{R_n}} \tag{17.61}$$

where Δp_{12} = total pressure loss between planes 1 and 2, in. WC (Pa)
$R_A, R_B, \ldots, R_n$ = flow resistance of duct sections $A, B, \ldots, n$, in. WC/(cfm)^2

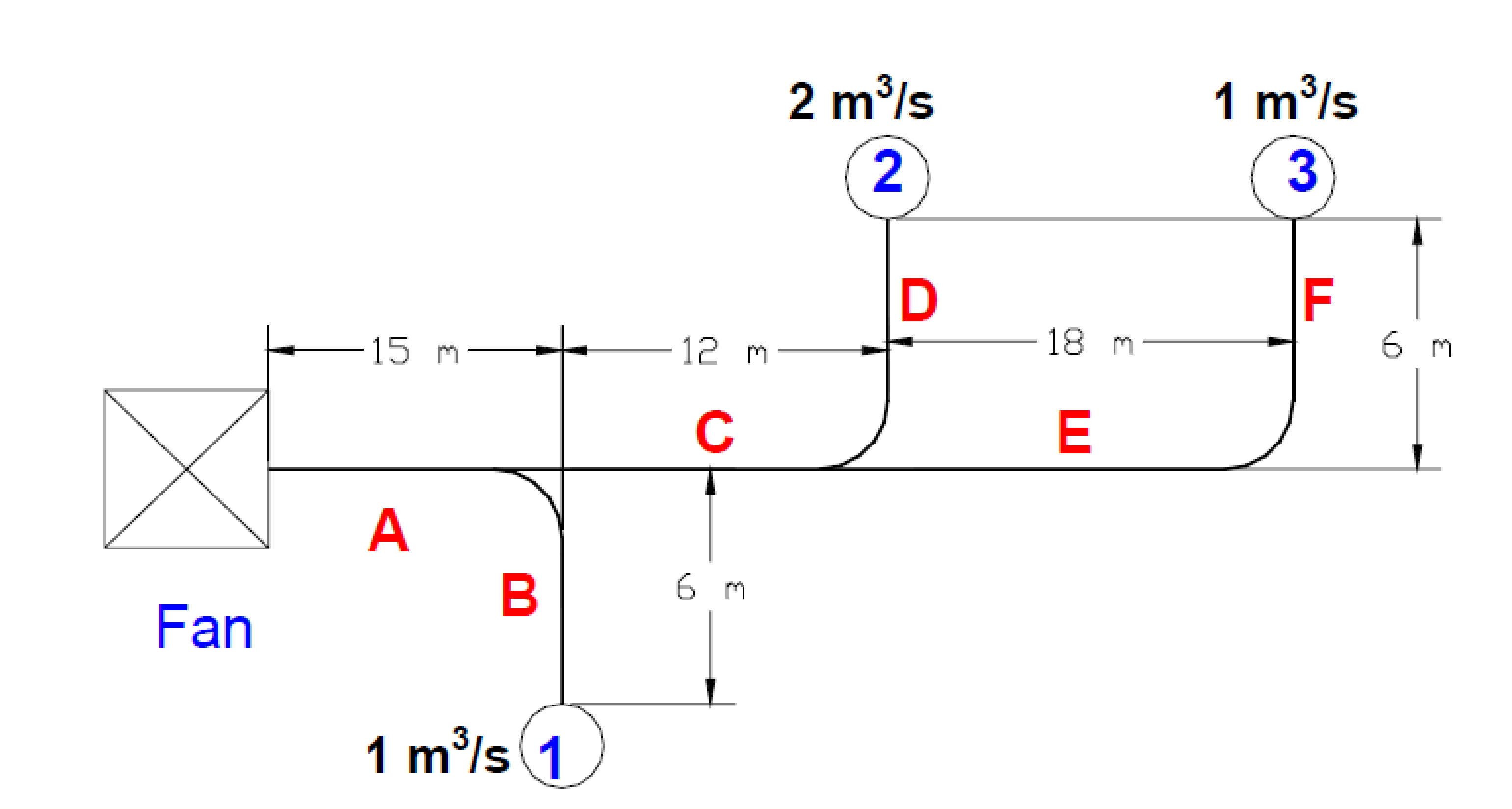
As for the entire duct, $\dot{V} = \sqrt{\Delta p_{12}/R_p}$. Here R_p represents the flow resistance of the entire duct whose sections are in parallel connection. Then

$$\frac{1}{\sqrt{R_p}} = \frac{1}{\sqrt{R_A}} + \frac{1}{\sqrt{R_B}} + \cdots + \frac{1}{\sqrt{R_n}} \tag{17.62}$$

Velocity method

- Select suitable velocities in the main and branch ducts
- Find the diameters of main and branch ducts from airflow rates and velocities for circular ducts. For rectangular ducts, find the cross-sectional area from flow rate and velocity, and then by fixing the aspect ratio, find the two sides of the rectangular duct
- From the velocities and duct dimensions obtained in the previous step, find the frictional pressure drop for main and branch ducts using friction chart or equation.
- From the duct layout, dimensions and airflow rates, find the dynamic pressure losses for all the bends and fittings
- Select a fan that can provide sufficient FTP for the index run
- Balancing dampers have to be installed in each run. The damper in the index run is left completely open, while the other dampers are throttled to reduce the flow rate to the required design values.

- Using the following figure shows a typical duct layout. Design the duct system using Velocity method,



a) Velocity method: Select a velocity of **5 m/s** for the downstream and branches. Then the dimensions of various duct runs are obtained as shown below:

Segment A: Flow rate, $Q_A = 4\ m^3/s$ and velocity, $V_A = 8$ m/s

$\Rightarrow$ cross-sectional area $A_A = Q_A/V_A = 4/8 = 0.5\ m^2 \Rightarrow$ $\mathbf{D_{eq,A} = 0.798\ m}$ **(Ans.)**

Segment B: Flow rate, $Q_B = 1\ m^3/s$ and velocity, $V_B = 5$ m/s

$\Rightarrow$ cross-sectional area $A_B = Q_B/V_B = 1/5 = 0.2\ m^2 \Rightarrow$ $\mathbf{D_{eq,B} = 0.505\ m}$ **(Ans.)**

Segment C: Flow rate, $Q_C = 3\ m^3/s$ and velocity, $V_C = 5$ m/s

$\Rightarrow$ cross-sectional area $A_C = Q_C/V_C = 3/5 = 0.6\ m^2 \Rightarrow$ $\mathbf{D_{eq,A} = 0.874\ m}$ **(Ans.)**

Segment D: Flow rate, $Q_D = 2\ m^3/s$ and velocity, $V_D = 5$ m/s

$\Rightarrow$ cross-sectional area $A_D = Q_D/V_D = 2/5 = 0.4\ m^2 \Rightarrow$ $\mathbf{D_{eq,D} = 0.714\ m}$ **(Ans.)**

Segments E&F: Flow rate, $Q_{E,F} = 1\ m^3/s$ and velocity, $V_{E,F} = 5$ m/s

$\Rightarrow$ cross-sectional area $A_{E,F} = Q_{E,F}/V_{E,F} = 1/5 = 0.2\ m^2 \Rightarrow$ $\mathbf{D_{eq,A} = 0.505\ m}$ **(Ans.)**

Calculation of pressure drop:

Section A-B:

$$\Delta P_{A\text{-}B} = \Delta P_{A,f} + \Delta P_{b,f} + \Delta P_{u\text{-}b} + \Delta P_{exit}$$

where $\Delta P_{A,f}$ and $\Delta P_{B,f}$ stand for frictional pressure drops in sections A and B, respectively, $\Delta P_{u\text{-}b}$ is the dynamic pressure drop from upstream to branch and ΔP_{exit} is the dynamic pressure loss at the exit 1.

The frictional pressure drop is calculated using the equation:

$$\Delta p_{A,f} = \frac{0.022243\,\dot{Q}_{air}^{\;1.852}\,L}{D^{4.973}} = \frac{0.022243 \text{x} 4^{1.852} \text{x} 15}{0.798^{4.973}} = 13.35 \text{ Pa}$$

$$\Delta p_{B,f} = \frac{0.022243\,\dot{Q}_{air}^{\;1.852}\,L}{D^{4.973}} = \frac{0.022243 \text{x} 1^{1.852} \text{x} 6}{0.505^{4.973}} = 3.99 \text{ Pa}$$

The dynamic pressure drop from upstream to branch is given by:

$$\Delta p_{u\text{-}b} = C_{u\text{-}b}\left(\frac{\rho V_d^{2}}{2}\right) = 0.8\left(\frac{1.2x5^{2}}{2}\right) = 12\ \text{Pa}$$

The dynamic pressure drop at the exit is given by:

$$\Delta p_{exit,1} = C_{exit}\left(\frac{\rho V_1^{2}}{2}\right) = 1.0\left(\frac{1.2x5^{2}}{2}\right) = 15\ \text{Pa}$$

Hence total pressure drop from the fan to the exit of 1 is given by:

$$\Delta P_{A\text{-}B} = \Delta P_{A,f} + \Delta P_{b,f} + \Delta P_{u\text{-}b} + \Delta P_{exit} = 13.35+3.99+12+15 = 44.34\ \text{Pa}$$

In a similar manner, the pressure drop from fan to 2 is obtained as:

$$\Delta P_{A\text{-}C\text{-}D} = \Delta P_{A,f} + \Delta P_{C,f} + \Delta P_{D,f} + \Delta P_{u\text{-}b} + \Delta P_{u\text{-}d} + \Delta P_{exit}$$

$$\Delta P_{A\text{-}C\text{-}D} = 13.35+3.99+2.57+12+4.5+15 = 51.41\ \text{Pa}$$

Pressure drop from fan to exit 3 is obtained as:

$$\Delta P_{A\text{-}C\text{-}E\text{-}F} = \Delta P_{A,f}+\Delta P_{C,f}+\Delta P_{E,f}+\Delta P_{F,f}+\Delta P_{u\text{-}d,C}+\Delta P_{u\text{-}d,E}+\Delta P_{elbow}+\Delta P_{exit}$$

$$\Delta P_{A\text{-}C\text{-}E\text{-}F} = 13.35+3.99+11.97+3.99+4.5+4.5+12+15 = 69.3\ \text{Pa}$$

Thus the run with maximum pressure drop is A-C-E-F is the index run. Hence th FTP required is:

$$\text{FTP} = \Delta P_{A\text{-}C\text{-}E\text{-}F} = 69.3\ \text{Pa} \qquad \text{(Ans.)}$$

Amount of dampering required at 1 = FTP - $\Delta P_{A\text{-}B}$ = **24.96 Pa** **(Ans.)**

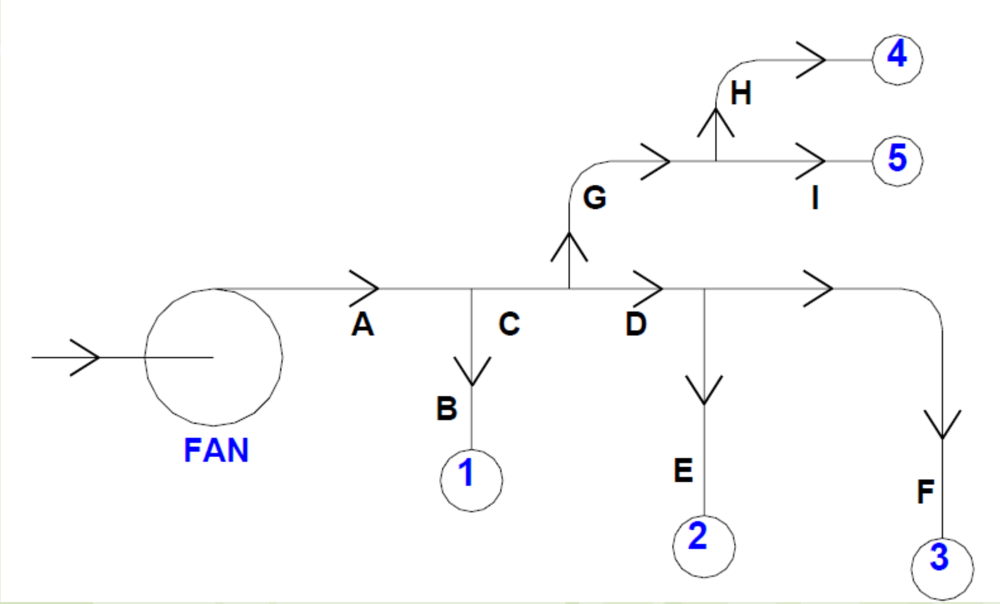
Amount of dampering required at 2 = FTP - $\Delta P_{A\text{-}C\text{-}D}$ = **17.89 Pa** **(Ans.)**

- The velocity method is one of the simplest ways of designing the duct system for both supply and return air. However, the application of this method requires selection of suitable velocities in different duct runs, which requires experience. Wrong selection of velocities can lead to very large ducts, which, occupy large building space and increases the cost, or very small ducts which lead to large pressure drop and hence necessitates the selection of a large fan leading to higher fan cost and running cost. In addition, the method is not very efficient as it requires partial closing of all the dampers except the one in the index run, so that the total pressure drop in each run will be same.

Equal friction method

- In this method the frictional pressure drop per unit length in the main and branch ducts ($\Delta p_f/L$) are kept same, i.e.,

$$\left(\frac{\Delta p_f}{L}\right)_A = \left(\frac{\Delta p_f}{L}\right)_B = \left(\frac{\Delta p_f}{L}\right)_C = \left(\frac{\Delta p_f}{L}\right)_D = \ldots$$



- Equal friction method is simple and is most widely used conventional method. This method usually yields a better design than the velocity method as most of the available pressure drop is dissipated as friction in the duct runs, rather than in the balancing dampers. This method is generally suitable when the ducts are not too long, and it can be used for both supply and return ducts. However, similar to velocity method, the equal friction method also requires partial closure of dampers in all but the index run, which may generate noise. If the ducts are too long then the total pressure drop will be high and due to dampering, ducts near the fan get over-pressurized.

- **For example,** let the duct run A-C-G-H be the index run and the total pressure drop in the index run is 100 Pa. If the pressure drop in the shortest duct run (say A-B) is 10 Pa, then the damper in this run has to be closed to provide an additional pressure drop of 90 Pa, so that the required airflow rate to the conditioned zone 1 can be maintained. Similarly the dampers in the other duct runs also have to be closed partially, so that the total pressure drop with damper partially closed in each run will be equal to the pressure drop in the index run with its damper left open fully.

- Select a suitable frictional pressure drop per unit length ($\Delta \mathbf{p_f}/\mathbf{L}$) so that the combined initial and running costs are minimized.
- Then the equivalent diameter of the main duct (A) is obtained from the selected value of ($\Delta \mathbf{p_f}/\mathbf{L}$) and the airflow rate. As shown in previous figure, airflow rate in the main duct $\dot{Q}_a$ is equal to the sum total of airflow rates to all the conditioned zones, i.e.,

$$\dot{Q}_A = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 + \dot{Q}_5 = \sum_{i=1}^{N} \dot{Q}_i$$

- From the airflow rate and $(\Delta \mathbf{p_f}/\mathbf{L})$ the equivalent diameter of the main duct $(D_{eq,A})$ can be obtained either from the friction chart or using the frictional pressure drop equation, i.e.,

$$D_{eq,A} = \left(\frac{0.022243\dot{Q}_A^{\ 1.852}}{\left(\frac{\Delta p_f}{L}\right)_A}\right)^{\frac{1}{4.973}}$$

- Since the frictional pressure drop per unit length is same for all the duct runs, the equivalent diameters of the other duct runs, B to I are obtained from the equation:

$$\left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\ 4.973}}\right)_A = \left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\ 4.973}}\right)_B = \left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\ 4.973}}\right)_C = \ldots$$

- If the ducts are rectangular, then the two sides of the rectangular duct of each run are obtained from the equivalent diameter of that run and by fixing aspect ratio as explained earlier. Thus the dimensions of the all the duct runs can be obtained. The velocity of air through each duct is obtained from the volumetric flow rate and the cross-sectional area.
- Next from the dimensions of the ducts in each run, the total frictional pressure drop of that run is obtained by multiplying the frictional pressure drop per unit length and the length, i.e.,

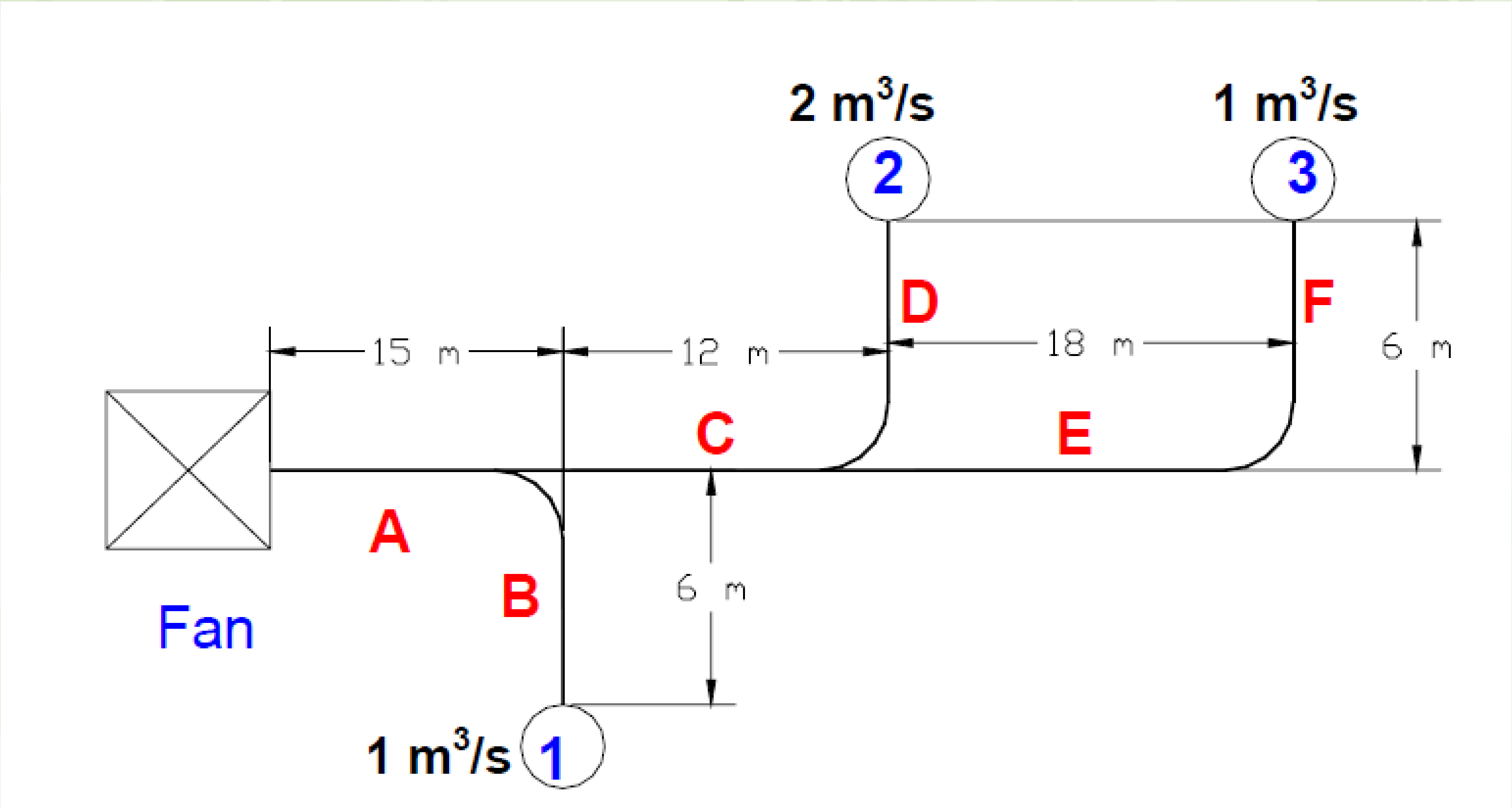
$$\Delta P_{f,A} = \left(\frac{\Delta p_f}{L}\right)_A . L_A \qquad \Delta P_{f,B} = \left(\frac{\Delta p_f}{L}\right)_B . L_B$$

- Next the dynamic pressure losses in each duct run are obtained based on the type of bends or fittings used in that run.
- Next the total pressure drop in each duct run is obtained by summing up the frictional and dynamic losses of that run, i.e.,

$$\Delta P_A = \Delta p_{f,A} + \Delta p_{d,A} \quad \Delta P_B = \Delta p_{f,B} + \Delta p_{d,B}$$

- Next the fan is selected to suit the index run with the highest pressure loss. Dampers are installed in all the duct runs to balance the total pressure loss.

Using the following figure shows a typical duct layout. Design the duct system using Equal Friction method,



b) Equal Friction Method:

The frictional pressure drop in segment A is given by:

$$\frac{\Delta p_{f,A}}{L_A} = \frac{0.022243\,\dot{Q}_{air}^{\;1.852}}{D^{4.973}} = \frac{0.022243\text{x}4^{1.852}}{0.798^{4.973}} = 0.89\ (\text{Pa}/\text{m})$$

The frictional pressure drops of B, C, D, E and F should be same as 0.89 Pa/m for Equal Friction Method. Hence, as discussed before:

$$\left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\;4.973}}\right)_A = \left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\;4.973}}\right)_B = \left(\frac{\dot{Q}^{1.852}}{D_{eq}^{\;4.973}}\right)_C = \ldots$$

$$D_{eq,B} = D_{eq,A}\left(\frac{Q_B}{Q_A}\right)^{\left(\frac{1.852}{4.973}\right)} = 0.4762\,\mathrm{m}$$

$$D_{eq,C} = D_{eq,A}\left(\frac{Q_C}{Q_A}\right)^{\left(\frac{1.852}{4.973}\right)} = 0.717\,\mathrm{m}$$

$$D_{eq,D} = D_{eq,A}\left(\frac{Q_D}{Q_A}\right)^{\left(\frac{1.852}{4.973}\right)} = 0.6164\,\mathrm{m}$$

$$D_{eq,E} = D_{eq,F} = D_{eq,A}\left(\frac{Q_E}{Q_A}\right)^{\left(\frac{1.852}{4.973}\right)} = 0.4762\,\mathrm{m}$$

<u>Calculation of total pressure drop:</u>

From fan to 1:

$$\Delta P_{A\text{-}B} = \Delta P_{A,f} + \Delta P_{B,f} + \Delta P_{u\text{-}b} + \Delta P_{exit}$$

$$\Delta P_{A\text{-}B} = 13.35 + 5.34 + 15.1 + 18.9 = 52.69 \text{ Pa}$$

From fan to 2:

$$\Delta P_{A\text{-}C\text{-}D} = \Delta P_{A,f} + \Delta P_{C,f} + \Delta P_{D,f} + \Delta P_{u\text{-}d,C} + \Delta P_{u\text{-}b} + \Delta P_{exit}$$

$$\Delta P_{A\text{-}C\text{-}D} = 13.35 + 10.68 + 5.34 + 9.94 + 21.55 + 26.9 = 87.76 \text{ Pa}$$

From fan to exit 3:

$$\Delta P_{A\text{-}C\text{-}E\text{-}F} = \Delta P_{A,f}+\Delta P_{C,f}+\Delta P_{E,f}+\Delta P_{F,f}+\Delta P_{u\text{-}d,C}+\Delta P_{u\text{-}d,E}+\Delta P_{elbow}+\Delta P_{exit}$$

$$\Delta P_{A\text{-}C\text{-}E\text{-}F} = 13.35 + 10.68 + 16.02 + 5.34 + 9.94 + 5.67 + 15.1 + 18.9 = 95 \text{ Pa}$$

As before, the Index run is from fan to exit 3. The required FTP is:

$$\text{FTP} = \Delta P_{A\text{-}C\text{-}E\text{-}F} = 95 \text{ Pa} \qquad \text{(Ans.)}$$

Amount of dampering required at 1 = FTP - $\Delta P_{A\text{-}B}$ = **42.31 Pa** **(Ans.)**

Amount of dampering required at 2 = FTP - $\Delta P_{A\text{-}C\text{-}D}$ = **7.24 Pa** **(Ans.)**

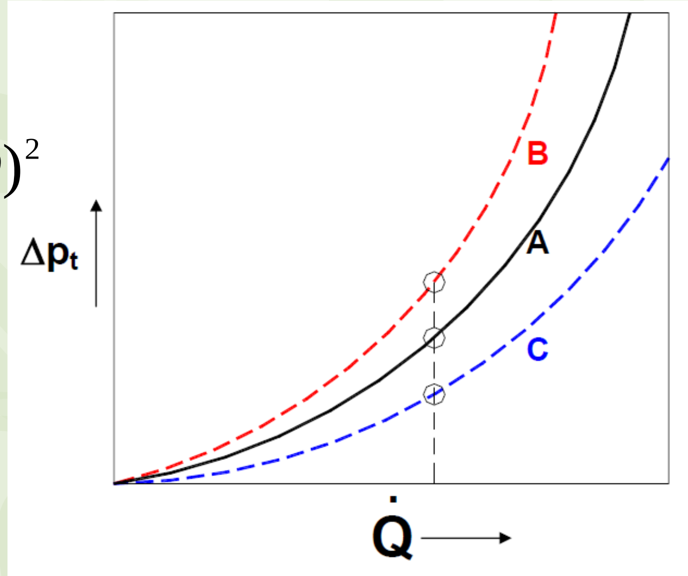
From the example, it is seen that the Velocity method results in larger duct diameters due to the velocities selected in branch and downstream. However, the required FTP is lower in case of velocity method due to larger ducts.
Equal Friction method results in smaller duct diameters, but larger FTP. Compared to velocity method, the required dampering is more at outlet 1 and less at outlet 2 in case of equal friction method.

Performance of duct systems

- For the duct system with air in turbulent flow, the total pressure loss ($\Delta \mathbf{p}_t$) is proportional to the square of flow rate; i.e.,

- Total pressure drop , $\Delta P_t \propto (\dot{Q})^2$

  or , total pressure drop , $\Delta P_t = C(\dot{Q})^2$



Variation of total pressure drop with flow rate for a given duct system

National Chiao Tung University

Fan

- To overcome the fluid friction and the resulting head, a fan is required in air conditioning systems. When a fan is introduced into the duct through which air is flowing, then the static and total pressures at the section where the fan is located. This rise is called as **Fan Total Pressure (FTP)**. Then the required power input to the fan is given by:

$$W_{fan} = \frac{\dot{Q}_{air}.FTP}{\eta_{fan}}$$

- The FTP should be such that it overcomes the pressure drop of air as it flows through the duct and the air finally enters the conditioned space with sufficient momentum so that a good air distribution can be obtained in the conditioned space. Evaluation of FTP is important in the selection of a suitable fan for a given application. It can be easily shown that when applied between any two sections 1 and 2 of the duct, in which the fan is located, the FTP is given by:

$$FTP = (p_2 - p_1) + \frac{\rho(V_2^{\,2} - V_1^{\,2})}{2g} + \rho g(z_2 - z_1) + \rho g H_1$$

Fan

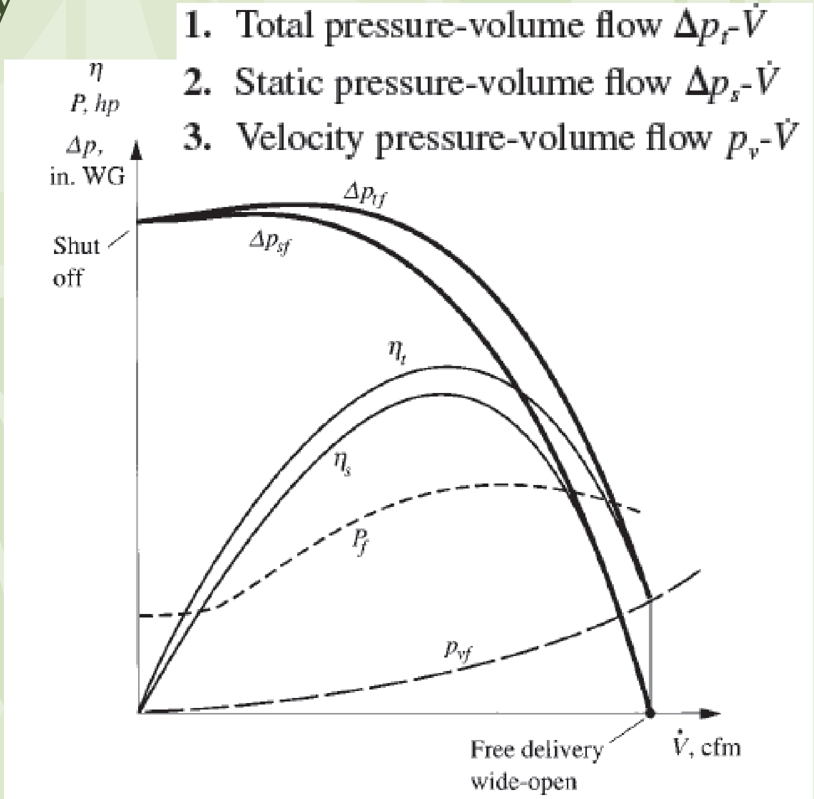
- A fan is the prime mover of an air system or ventilation system. It moves the air and provides continuous airflow so that the conditioned air, space air, exhaust air, or outdoor air can be transported from one location to another through air ducts or other air passages.
- A fan is also a turbomachine in which air is usually compressed at a compression ratio R_{com} not greater than 1.07. The *compression ratio*, dimensionless, is defined as

$$R_{\text{com}} = \frac{p_{\text{dis}}}{p_{\text{suc}}}$$

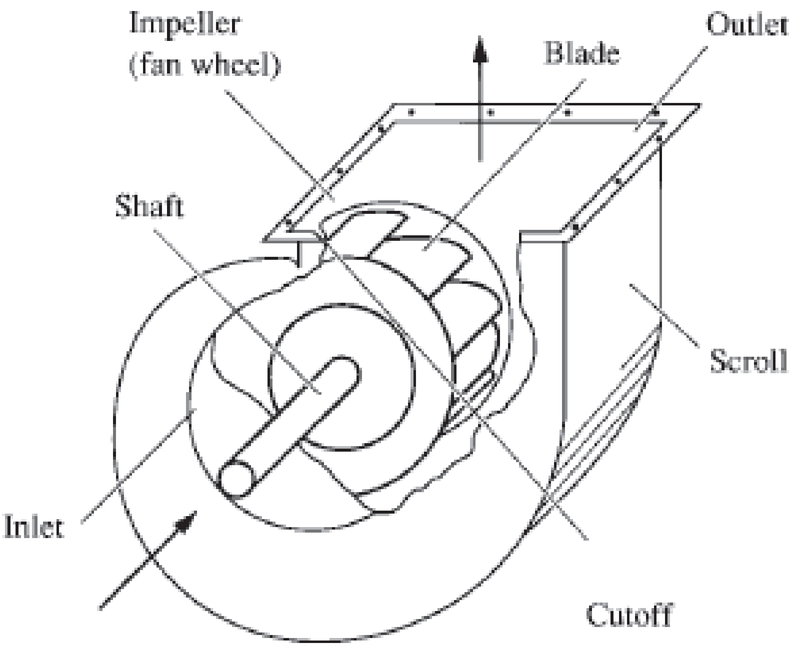
- where P_{dis} discharge pressure at outlet of compressor or fan, lbf/in.^{2} abs. or psia (kPa abs.) P_{suc} suction pressure at inlet of compressor or fan, psia (kPa abs.)

Fan

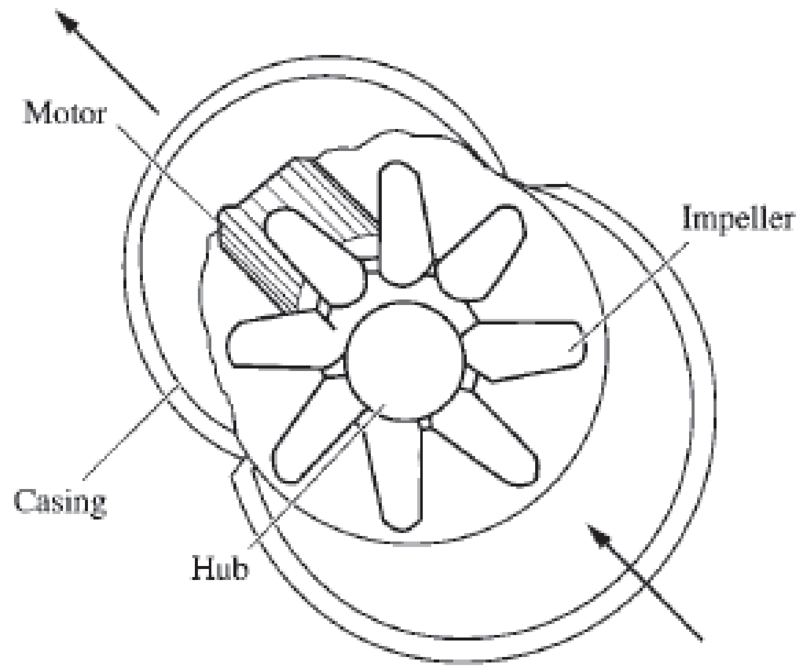
- Fan Capacity or Volume Flow Rate
- Fan Pressure
- Air Temperature Increase through Fan
- Fan Power and Fan Efficiency
- Fan Performance Curves



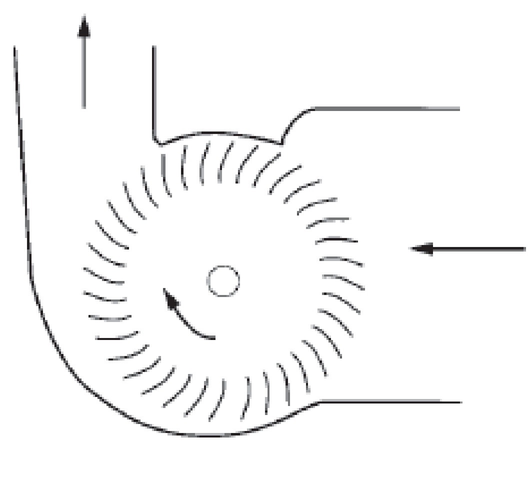
Types of fans: (*a*) centrifugal; (*b*) axial; (*c*) crossflow

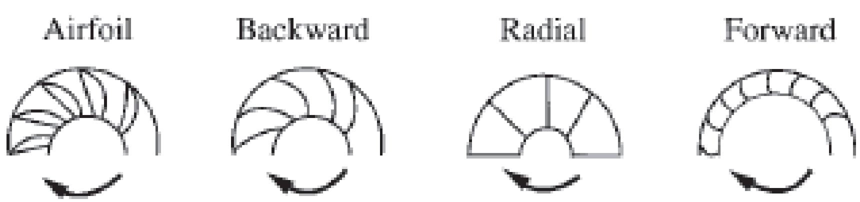


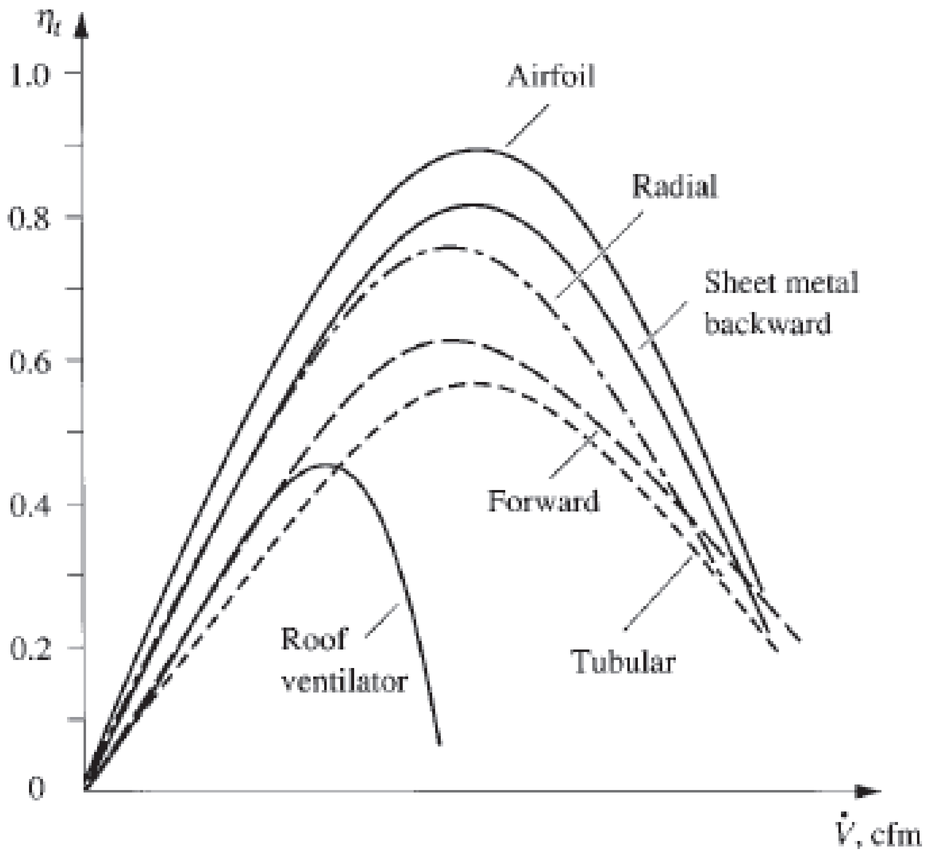
(a)

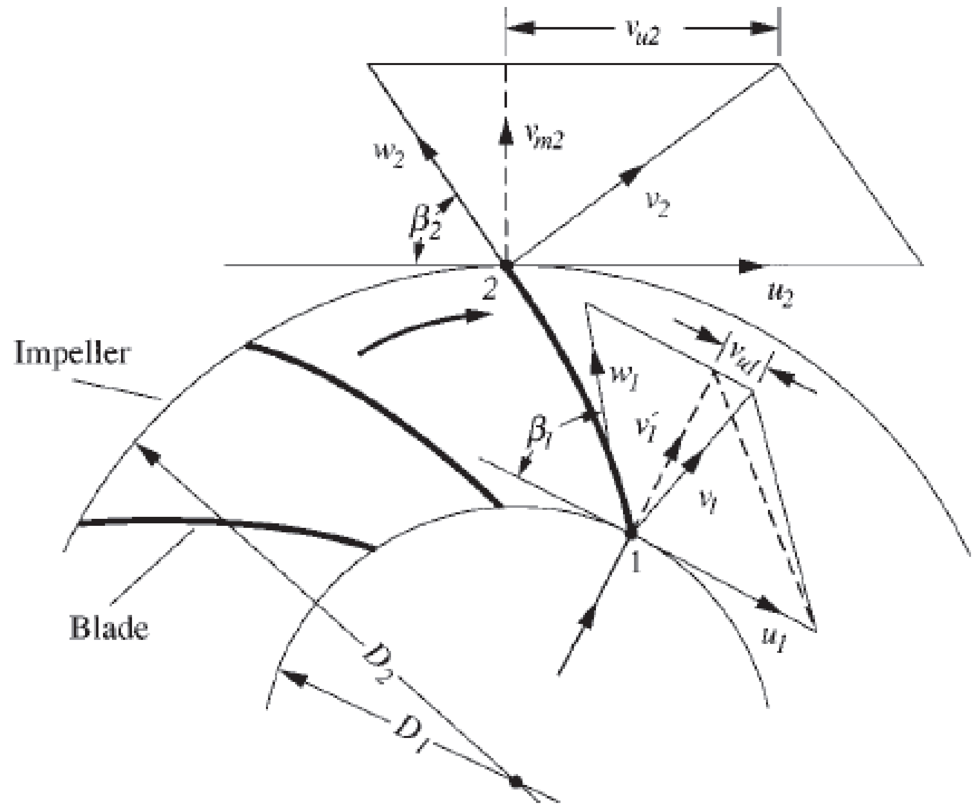


(b)




Airfoil
Backward
Radial
Forward


η_t
1.0
0.8
0.6
0.4
0.2
0
Airfoil
Radial
Sheet metal backward
Forward
Roof ventilator
Tubular
$\dot{V}$, cfm


v_{u2}
w_2
v_{m2}
v_2
β_2
2
u_2
Impeller
w_1
v_{u1}
β_1
v_1'
v_1
1
Blade
u_1
D_2
D_1

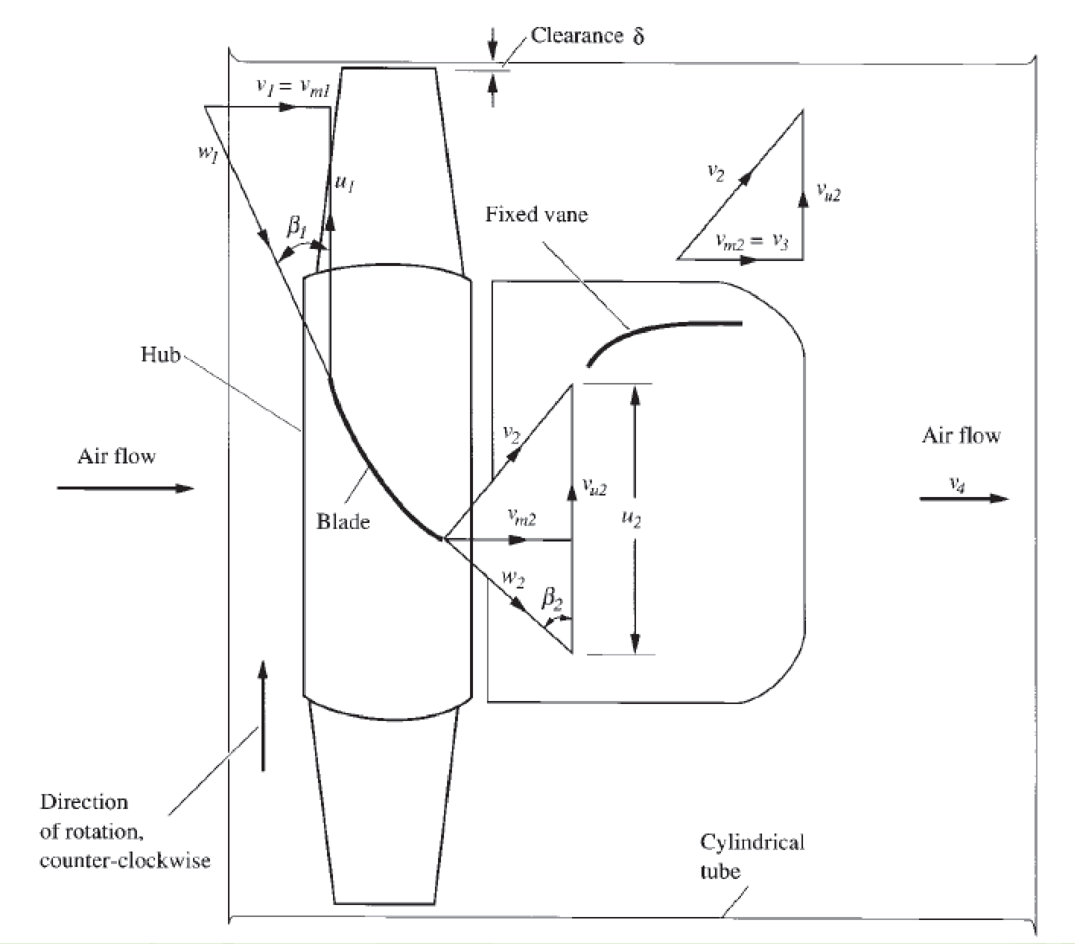
In a backward-curved or backward-inclined centrifugal fan, the blade tip inclines away from the direction of rotation of the impeller. The 2 angle of a backward-curved centrifugal fan is less than 90°. The impeller of a backward-curved centrifugal fan usually consists of 8 to 16 blades. For greater efficiency, the shape of the blades is often streamlined to provide minimum flow separation and, therefore, minimum energy losses. Backward-curved centrifugal fans with such blades are called airfoil fans, as distinguished from fans with sheet-metal blades. The blades in a backward-curved fan are always longer than those of a forward-curved fan. A volute or scroll casing is used.

Radial-Bladed Fans

The blades in a radial-bladed centrifugal fan are either straight or curved at the blade inlet. The blade tip or blade outlet is always radial; that is, 2 90°, as shown. Usually, there are 6 to 10 blades in a radial-bladed impeller. The construction of the radial blades is comparatively simple.

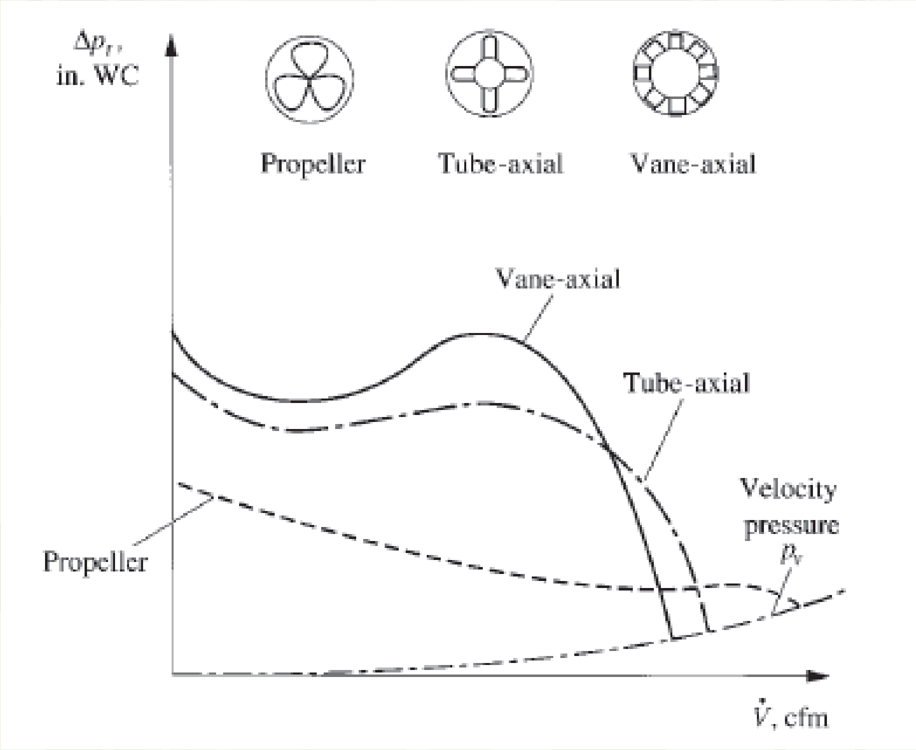
AXIAL FANS

- **Types of Axial Fans**
  - For an axial fan, a parameter called the hub ratio is closely related to its characteristics. *Hub ratio* R_{hub} is defined as the ratio of hub diameter D_{hub}, in ft (m), to the tip-to-tip blade diameter or diameter of impeller D_{bt}, in ft (m),



$$R_{hub} = \frac{D_{hub}}{D_{bt}}$$

- ***Propeller Fans.*** an impeller having 3 to 6 blades is mounted within a circular ring or an orifice plate. The blades are generally made of steel or molded plastic and sometimes may increase in width at the blade tip. If the impeller is mounted inside an orifice plate, the direction of airflow at the blade tip will not be parallel to the axle. Eddies may form at the blade tips. Propeller fans are usually operated at very low static pressure with large volume flow. They often have a hub ratio R_{hub} 0.15.



- ***Tube-Axial Fans.*** The impeller of a tube-axial fan usually has 6 to 9 blades. It is mounted within a cylindrical casing. The blades can be airfoil blades or curved sheet metal. Airfoil blades usually made of cast aluminum or aluminum alloy. The hub ratio R_{hub} is generally less than 0.3, and the clearance between the blade tip and the casing is significantly closer than in propeller fans.
- ***Vane-Axial Fans.*** The impeller of a vane-axial fan has 8 to 16 blades, usually airfoil blades. The hub ratio is generally equal to or greater than 0.3 in order to increase the fan total pressure. Another important characteristic of vane-axial fans is the installation of fixed guide vanes downstream from the impeller. These curved vanes are designed to remove swirl from the air, straighten the airflow, and convert a portion of the velocity pressure of the rotating airflow to static pressure.

Fan Laws

- Airflow rate , $\dot{Q} \propto \omega$
- Static pressure rise , $\Delta P_S \propto \dfrac{\rho V^2}{2}$
- Fan power input , $\dot{W} \propto \dot{Q}(\Delta P_S) + \dot{Q}(\dfrac{\rho V^2}{2})$

Law 1: Density of air ρ remains constant and the speed ω varies:

$\dot{Q} \propto \omega$; $\Delta P_S \propto \omega^2$ and $\dot{W} \propto \omega^3$

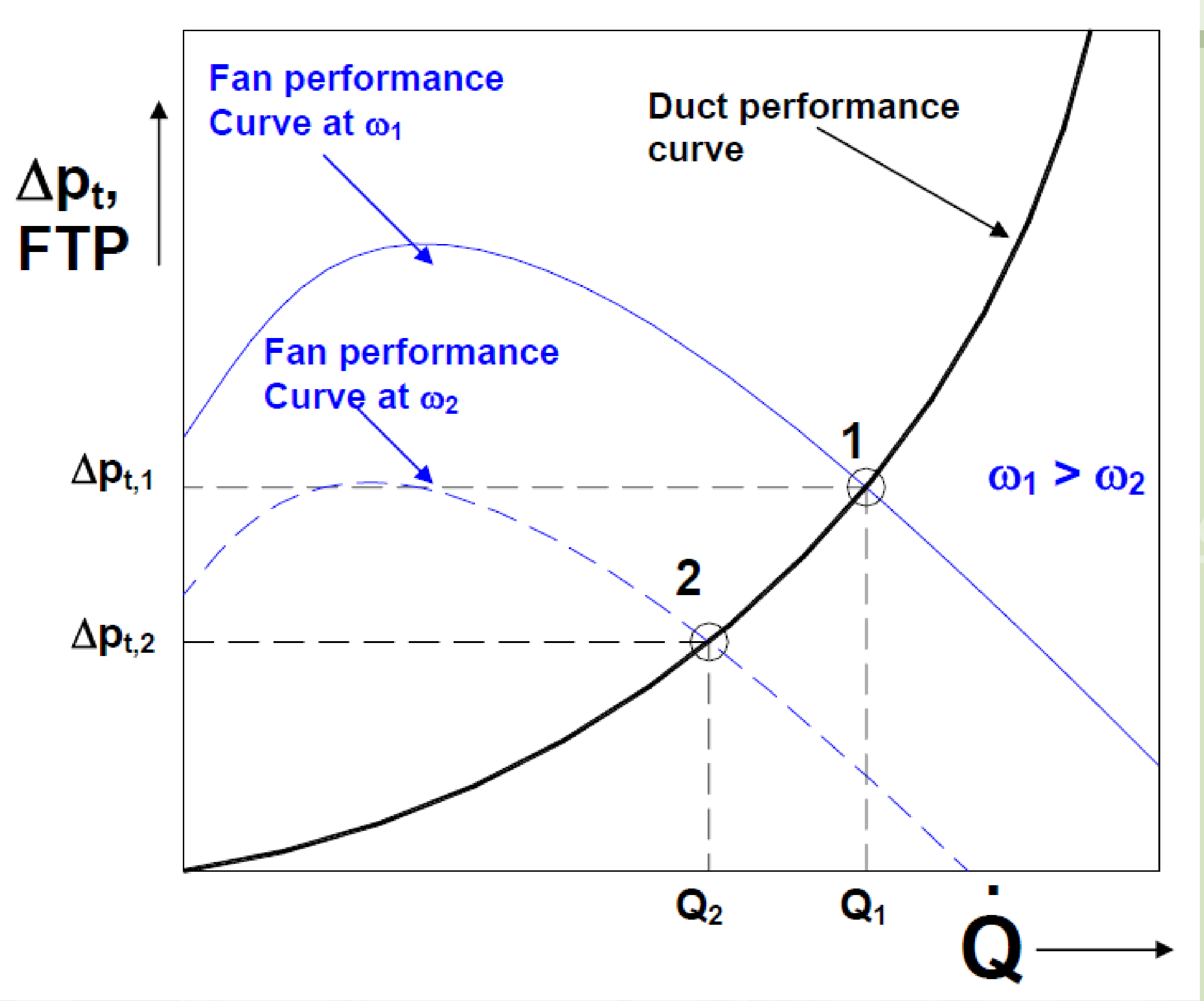
Law 2 : Airflow rate $\dot{Q}$ remains constant and the density ρ varies:

$\dot{Q} = CONST$; $\dot{W} \propto \rho$ and $\Delta P_S \propto \rho$

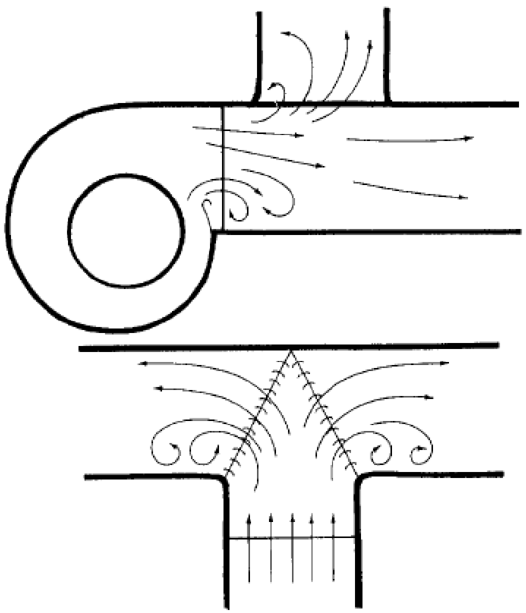
Law 3 : Static pressure rise ΔP_s remains constant and density ρ varies :

$\dot{Q} \propto \dfrac{1}{\sqrt{\rho}}$; ΔP_s=const ; $\omega \propto \dfrac{1}{\sqrt{\rho}}$ and $\dot{W} \propto \dfrac{1}{\sqrt{\rho}}$

Interaction between fan and duct system



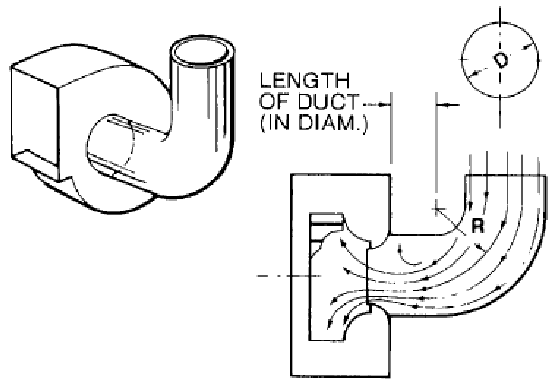
Fan and duct performance curves and balance points



AVOID LOCATION OF SPLIT OR DUCT BRANCH CLOSE TO FAN DISCHARGE. PROVIDE A STRAIGHT SECTION OF DUCT TO ALLOW FOR AIR DIFFUSION.

(See Figure 6-2 for corrective calculations)

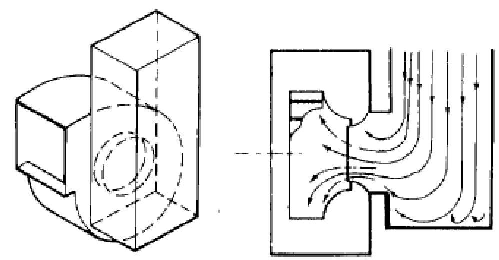
Figure 6-6 BRANCHES LOCATED TOO CLOSE TO FAN (1)



SYSTEM EFFECT CURVES

R/D	NO DUCT	2D DUCT	5D DUCT
0.75	Q-R	S	U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S-T	U	V-W

Figure 6-9 NON-UNIFORM FLOW INTO A FAN INLET INDUCED BY A 90° ROUND SECTION ELBOW—NO TURNING VANES (1)



THE REDUCTION IN CAPACITY AND PRESSURE FOR THIS TYPE OF INLET CONDITION IS IMPOSSIBLE TO TABULATE. THE MANY POSSIBLE VARIATIONS IN WIDTH AND DEPTH OF THE DUCT INFLUENCE THE REDUCTION IN PERFORMANCE TO VARYING DEGREES AND THEREFORE THIS INLET SHOULD BE AVOIDED. CAPACITY LOSSES AS HIGH AS 45 PERCENT HAVE BEEN OBSERVED. EXISTING INSTALLATIONS CAN BE IMPROVED WITH VANES OR THE CONVERSION TO SQUARE OR MITERED ELBOWS WITH VANES.

Figure 6-10 NON-UNIFORM FLOW INDUCED INTO FAN INLET BY A RECTANGULAR INLET DUCT (1)

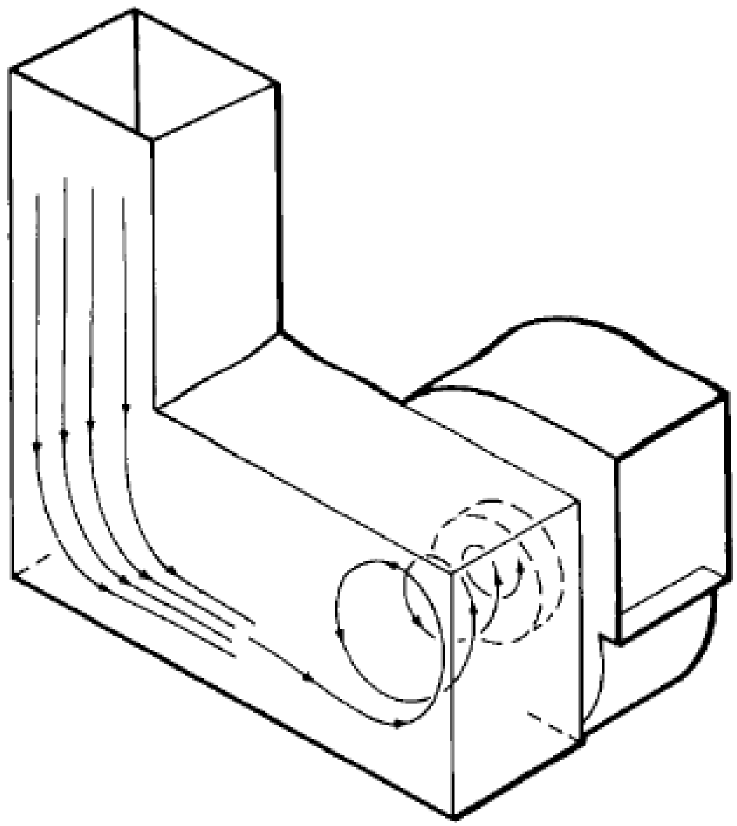
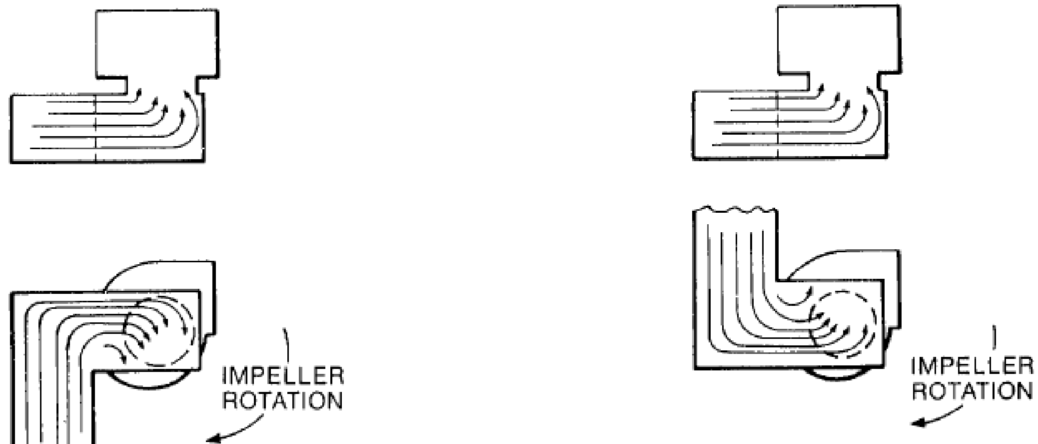


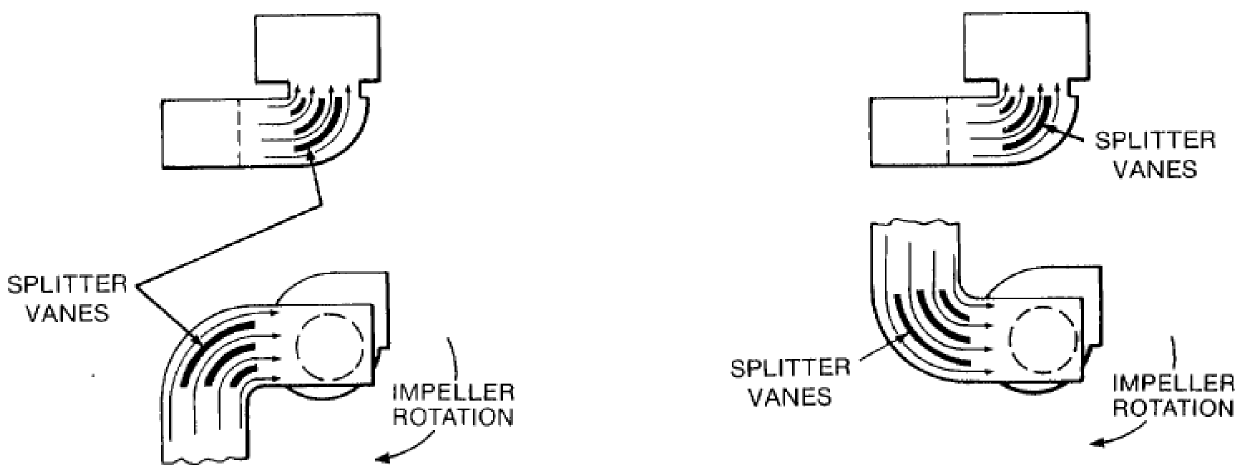
Figure 6-13 EXAMPLE OF A FORCED INLET VORTEX (SPIN-SWIRL) (1)



ROTATING SWIRL

COUNTER-ROTATING
SWIRL

Figure 6-14 INLET DUCT CONNECTIONS CAUSING INLET SPIN (1)



CORRECTED
ROTATING SWIRL

CORRECTED COUNTER-
ROTATING SWIRL

Figure 6-15 CORRECTIONS FOR INLET SPIN (1)

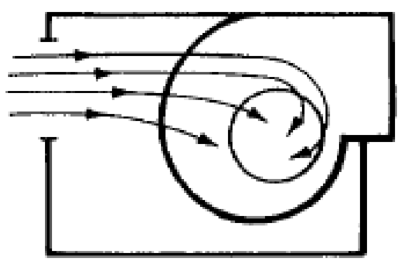


Figure 6-18 ENCLOSURE INLET NOT SYMMETRICAL WITH FAN INLET, PREROTATIONAL VORTEX INDUCED (1)

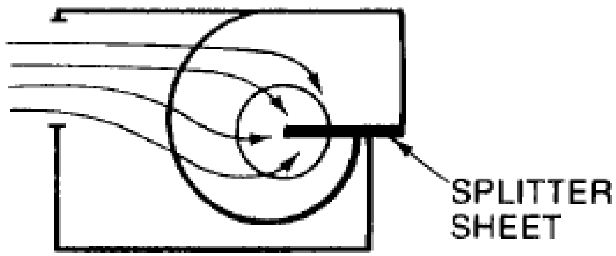


Figure 6-19 FLOW CONDITION OF FIGURE 6-18 IMPROVED WITH A SPLITTER SHEET (1)

Thank You

Stack effect

When an air duct system has an elevation difference and the air temperature inside the air duct is different from the ambient air temperature, the stack effect exists. It affects airflow at different elevations.

Form Equation (2) , the thermal gravity effect for each nonhorizontal duct with a density other than that of ambient air is determined by the following equation :

$$\Delta p_{se} = g(\rho_a - \rho)(z_2 - z_1)$$

Where Δp_{se} = thermal gravity effect , Pa

z_1 and z_2 = elevation from datum in direction airflow , m

ρ_a = density of ambient air , kg/m^3

ρ = density of air gas within duct , kg/m^3

g = 9.81 = gravitational acceleration , m/s^2

Example 1. For Figure 1, calculate the thermal gravity effect for two cases: (a) air cooled to –34°C, and (b) air heated to 540°C. Density of air at 34°C is 1.477 kg/m^3 and at 540°C is 434 kg/m^3. Density of ambient air is 1.204 kg/m^3. Stack height is 15 m.

Solution:

$$\Delta p_{se} = 9.81(\rho_a - \rho)z$$

(a) For $\rho > \rho_a$ (Figure 1A),

$$\Delta p_{se} = 9.81\,(1.204 - 1.477)15 = -40 \text{ Pa}$$

(b) For $\rho < \rho_a$ (Figure 1B),

$$\Delta p_{se} = 9.81(1.204 - 0.434)15 = +113 \text{ Pa}$$

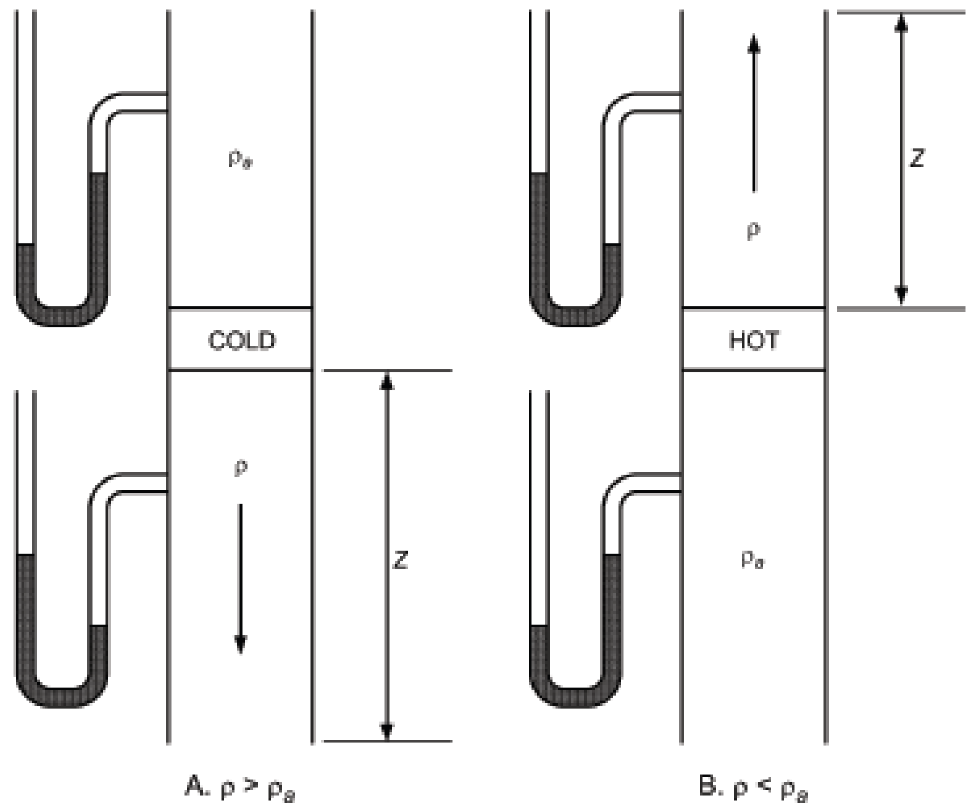


Fig. 1 Thermal Gravity Effect for Example 1

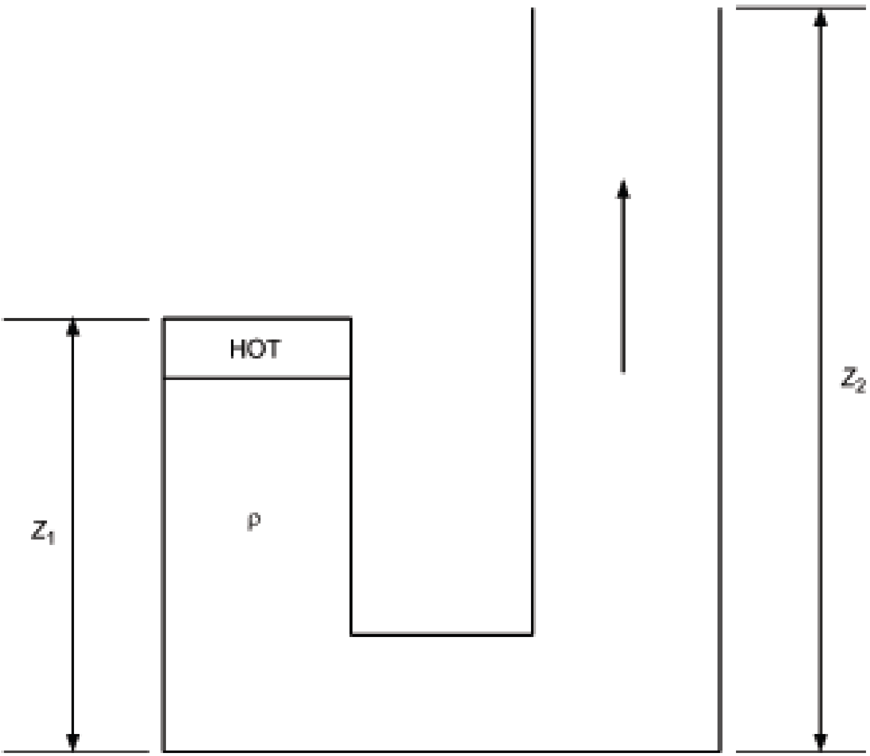


Fig. 2 Multiple Stacks for Example 2

Example 2. Calculate the thermal gravity effect for the two-stack system shown in Figure 2, where the air is 120°C and stack heights are 15 and 30 m. Density of 120°C air is 0.898 kg/m^3; ambient air is 1.204 kg/m^3.

Solution:

$$\Delta p_{se} = 9.81(\rho_a - \rho)(z_2 - z_1) = 9.81(1.204 - 0.898)(30 - 15) = 45 \text{ Pa}$$

For the system shown in Figure 3, the direction of air movement created by the thermal gravity effect depends on the initiating force (e.g., fans, wind, opening and closing doors, turning equipment on and off). If for any reason air starts to enter the left stack (Figure 3A), it creates a buoyancy effect in the right stack. On the other hand, if flow starts to enter the right stack (Figure 3B), it creates a buoyancy effect in the left stack. In both cases, the produced thermal gravity effect is stable and depends on stack height and magnitude of heating. The starting direction of flow is important when using natural convection for ventilation.

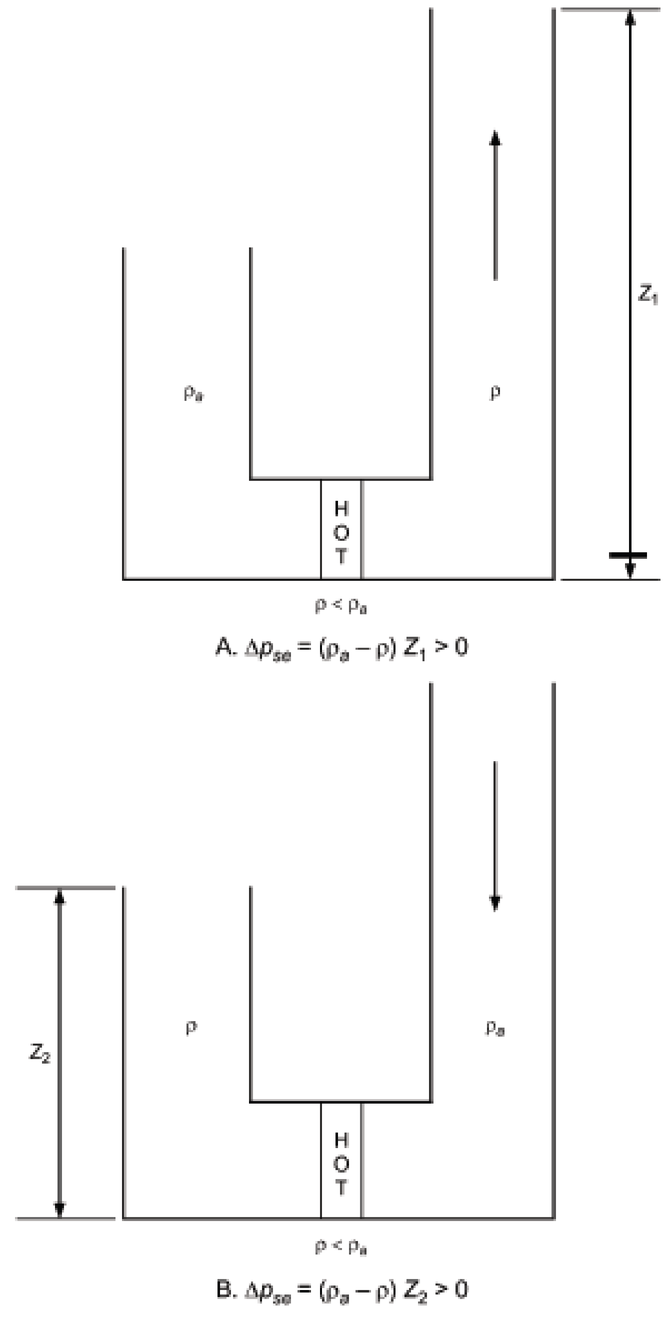


Fig. 3 Multiple Stack Analysis

Example 3. For Figures 5A and 5C, calculate the thermal gravity effect and fan total pressure required when the air is cooled to –34°C. The heat exchanger and ductwork (section 1 to 2) total pressure losses are 170 and 70 Pa respectively. The density of –34°C air is 1.477 kg/m^3; ambient air is 1.204 kg/m^3. Elevations are 21 and 3 m.

Solution:

(a) For Figure 5A (downward flow),

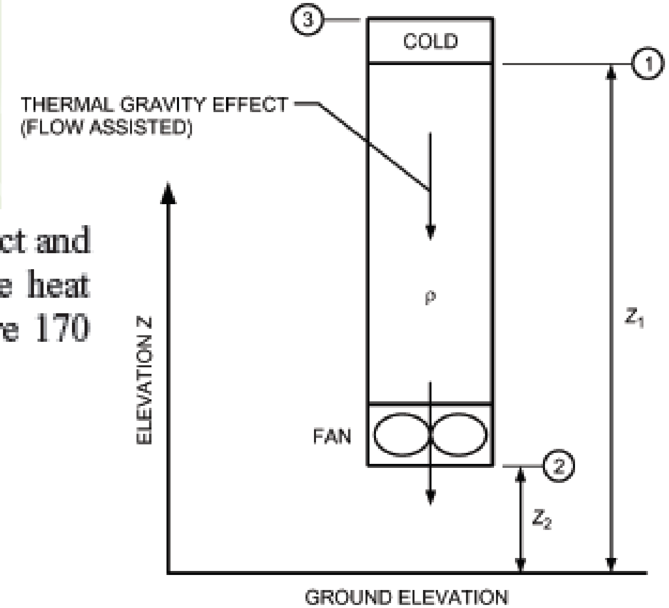
$$\begin{aligned}\Delta p_{se} &= 9.81(\rho_a - \rho)(z_2 - z_1)\\ &= 9.81(1.204 - 1.477)(3 - 21)\\ &= 48 \text{ Pa}\end{aligned}$$

$$\begin{aligned}P_t &= \Delta p_{t,3\text{-}2} - \Delta p_{se}\\ &= (170 + 70) - (48)\\ &= 192 \text{ Pa}\end{aligned}$$

(b) For Figure 5C (upward flow),

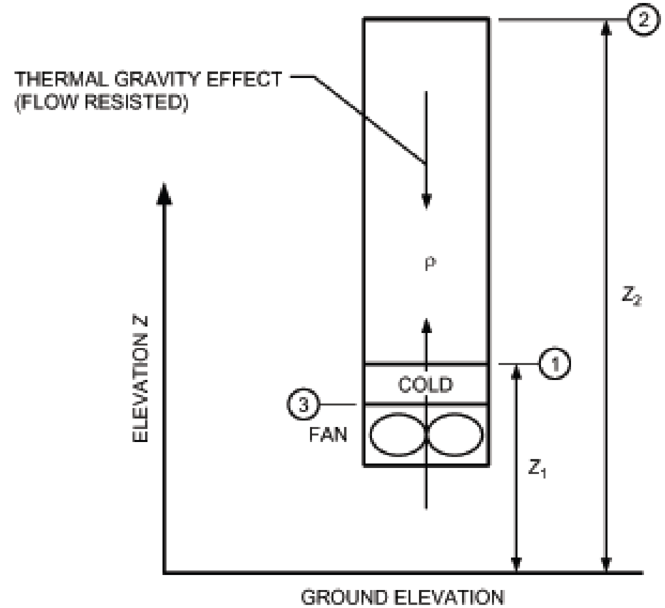
$$\begin{aligned}\Delta p_{se} &= 9.81(\rho_a - \rho)(z_2 - z_1)\\ &= 9.81(1.204 - 1.477)(21 - 3)\\ &= -48 \text{ Pa}\end{aligned}$$

$$\begin{aligned}P_t &= \Delta p_{t,3\text{-}2} - \Delta p_{se}\\ &= (170 + 70) - (-48)\\ &= 288 \text{ Pa}\end{aligned}$$



A. $\rho > \rho_a$

DOWNWAR



C. $\rho > \rho_a$

UPWARD

Example 4. For Figures 5B and 5D, calculate the thermal gravity effect and fan total pressure required when air is heated to 120°C. Heat exchanger and ductwork (section 1 to 2) total pressure losses are 170 and 70 Pa respectively. Density of 120°C air is 0.898 kg/m^3; ambient air is 1.204 kg/m^3. Elevations are 21 and 3 m.

Solution:

(a) For Figure 5B (downward flow),

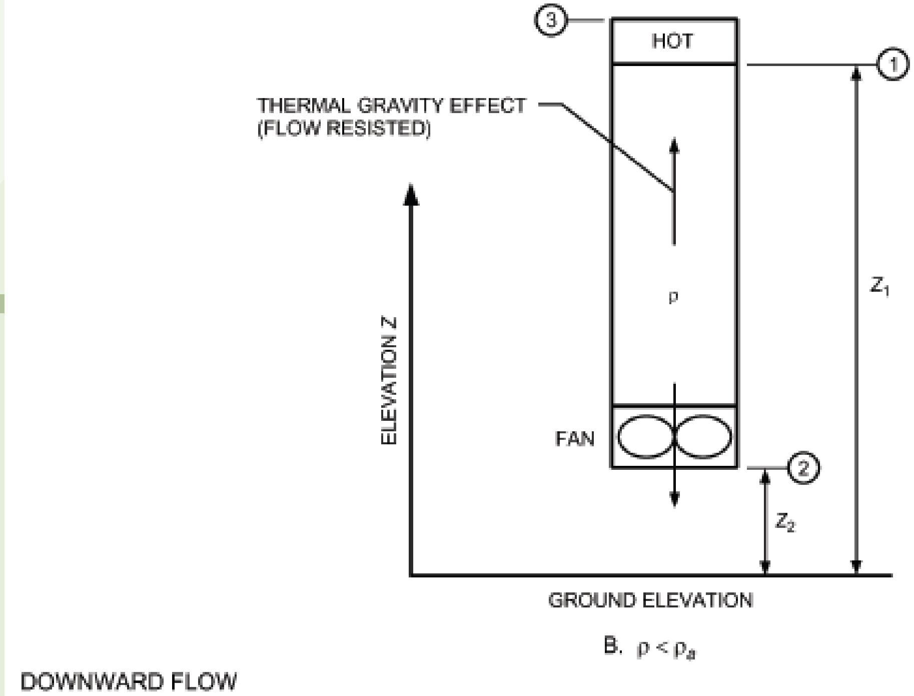
$$\begin{aligned}\Delta p_{se} &= 9.81(\rho_a - \rho)(z_2 - z_1)\\ &= 9.81(1.204 - 0.898)(3 - 21)\\ &= -54 \text{ Pa}\end{aligned}$$

$$\begin{aligned}P_t &= \Delta p_{t,3\text{-}2} - \Delta p_{se}\\ &= (170 + 70) - (-54)\\ &= 294 \text{ Pa}\end{aligned}$$

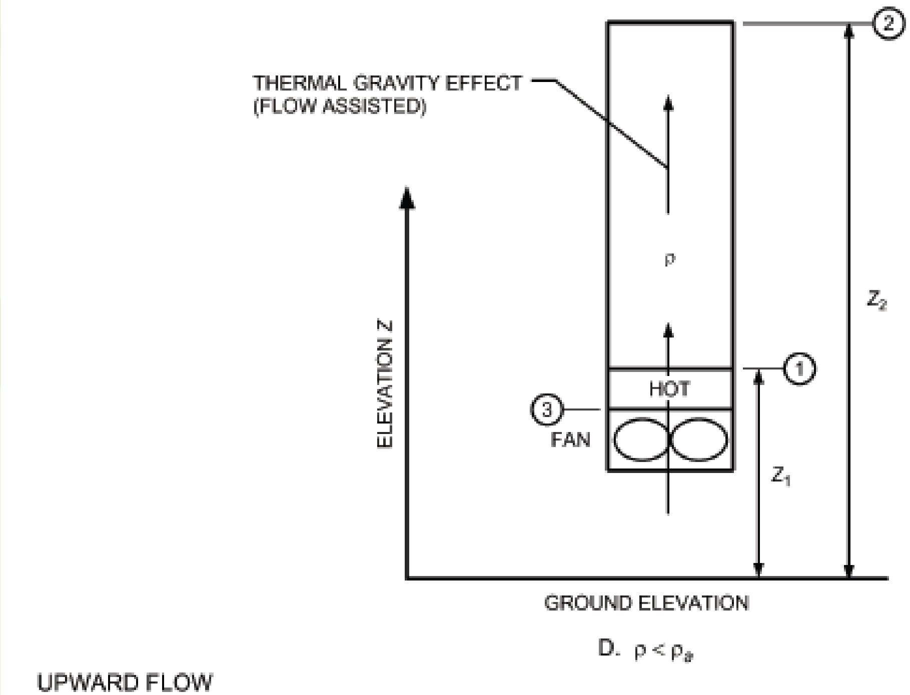
(b) For Figure 5D (upward flow),

$$\begin{aligned}\Delta p_{se} &= 9.81(\rho_a - \rho)(z_2 - z_1)\\ &= 9.81(1.204 - 0.898)(21 - 3)\\ &= 54 \text{ Pa}\end{aligned}$$

$$\begin{aligned}P_t &= \Delta p_{t,3\text{-}2} - \Delta p_{se}\\ &= (170 + 70) - (54)\\ &= 186 \text{ Pa}\end{aligned}$$



B. $\rho < \rho_a$



D. $\rho < \rho_a$

Example 5. Calculate the thermal gravity effect for each section of the system shown in Figure 6, and the systems' net thermal gravity effect. Density of ambient air is 1.204 kg/m^3, and the lengths are as follows: z_1 = 15 m, z_2 = 27 m, z_4 = 30 m, z_5 = 8 m, and z_9 = 60 m. Pressure required at section 3 is –25 Pa. Write the equation to determine the fan total pressure requirement.

Solution: The following table summarizes the thermal gravity effect for each section of the system as calculated by Equation (14). The net thermal gravity effect for the system is 118 Pa. To select a fan, use the following equation:

$$P_t = 25 + \Delta p_{t,1\text{-}7} + \Delta p_{t,8\text{-}9} - \Delta p_{se} = 25 + \Delta p_{t,1\text{-}7} + \Delta p_{t,8\text{-}9} - 118 = \Delta p_{t,1\text{-}7} + \Delta p_{t,8\text{-}9} - 93$$

Path (x–x')	Temp., °C	ρ, kg/m^3	Δz ($z_{x'} - z_x$), m	$\Delta\rho$ ($\rho_a - \rho_{x-x'}$), kg/m^3	Δp_{se}, Pa [Eq. (14)]
1-2	815	0.324	(27 – 15)	+0.880	+104
3-4	540	0.434	0	+0.770	0
4-5	540	0.434	(8 – 30)	+0.770	–166
6-7	120	0.898	0	+0.306	0
8-9	120	0.898	(60 – 0)	+0.306	+180
Net Thermal Gravity Effect					118

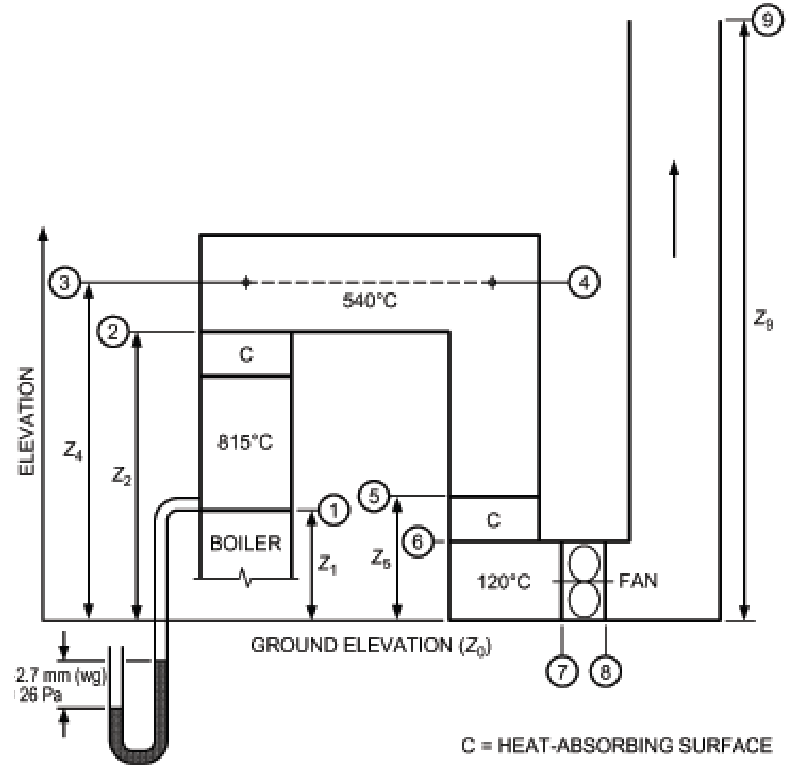


Fig. 6 Triple Stack System for Example 5

Static Regain Method

- This method is commonly used for high velocity systems with long duct runs, especially in large systems. In this method the static pressure is maintained same before each terminal or branch.
- Static Regain method yields a more balanced system and does not call for unnecessary dampering. However, as velocity reduces in the direction of airflow, the duct size may increase in the airflow direction. Also the velocity at the exit of the longer duct runs may become too small for proper air distribution in the conditioned space.

$$\Delta P_{\mathrm{stat}_{c-s}} = R\frac{\rho}{2}(V_c^2 - V_s^2) \tag{1}$$

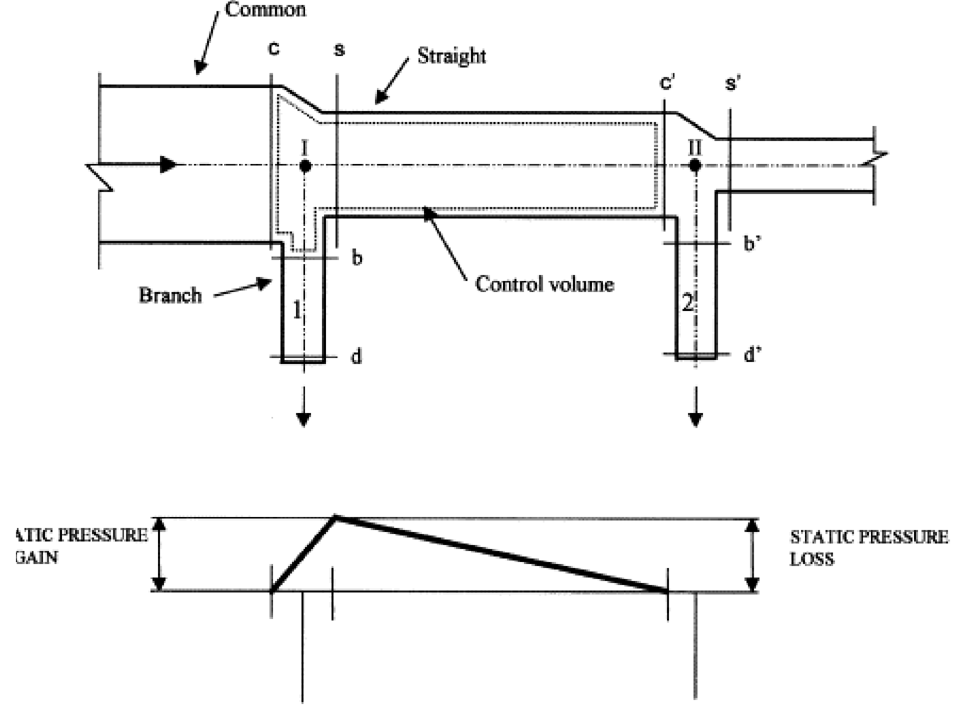
The static pressure drop between s and c′, $\Delta P_{\mathrm{stat}_{s-c'}}$, is the sum of the friction and dynamic pressure losses

$$\Delta P_{\mathrm{stat}_{s-c'}} = \frac{\rho}{2}V_s^2\left(C_{s-c'} + \left(\frac{fL}{D}\right)_{s-c'}\right) \tag{2}$$

To obtain the same static pressures at c and c′, $\Delta P_{\mathrm{stat}_{c-s}}$ and $\Delta P_{\mathrm{stat}_{s-c'}}$ are equated to give

$$V_s^2\left(C_{s-c'} + (fL)_{s-c'}\sqrt{\frac{\pi V_s}{4Q_s}} + R\right) - RV_c^2 = 0 \tag{3}$$

which can be solved for V_s. (V_c is known since the Static Regain method starts with a duct section with a chosen velocity and then works towards the air terminals.)

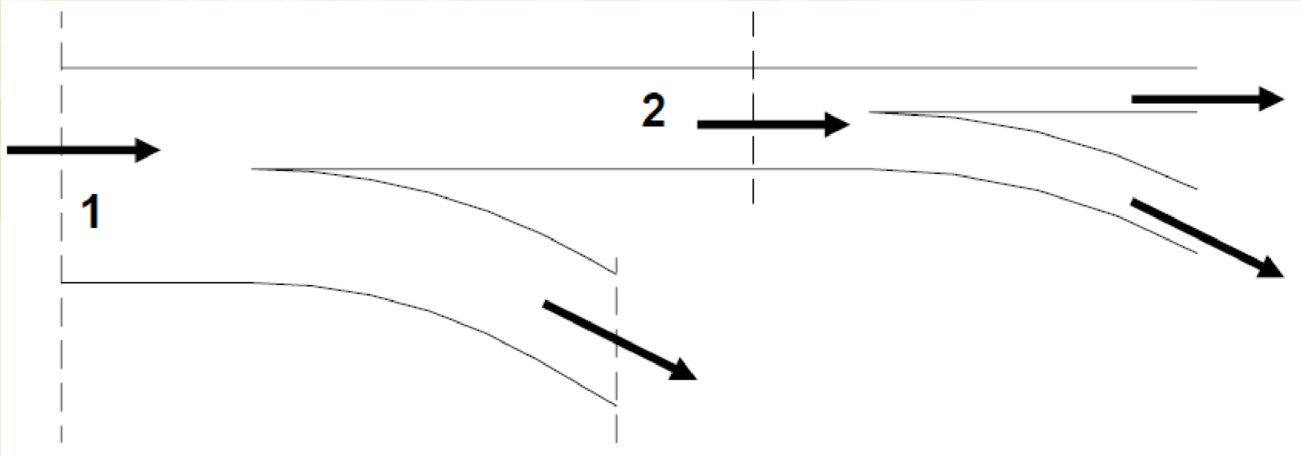


The purpose of the Static Regain method is to create equal static pressures at successive junctions which will presumably cause equal flows in branches that are identical "a branch might lead to a di}user or to an entire duct subsystem. When this principle is applied to a main duct with identical branches leading to identical diffusers equal air quantities will be delivered without the need to throttle the flow in the upstream branches The same principle is utilized to provide equal pressures at the take off points of a duct riser that serves several which means that the duct sizing of each can be considered typical.

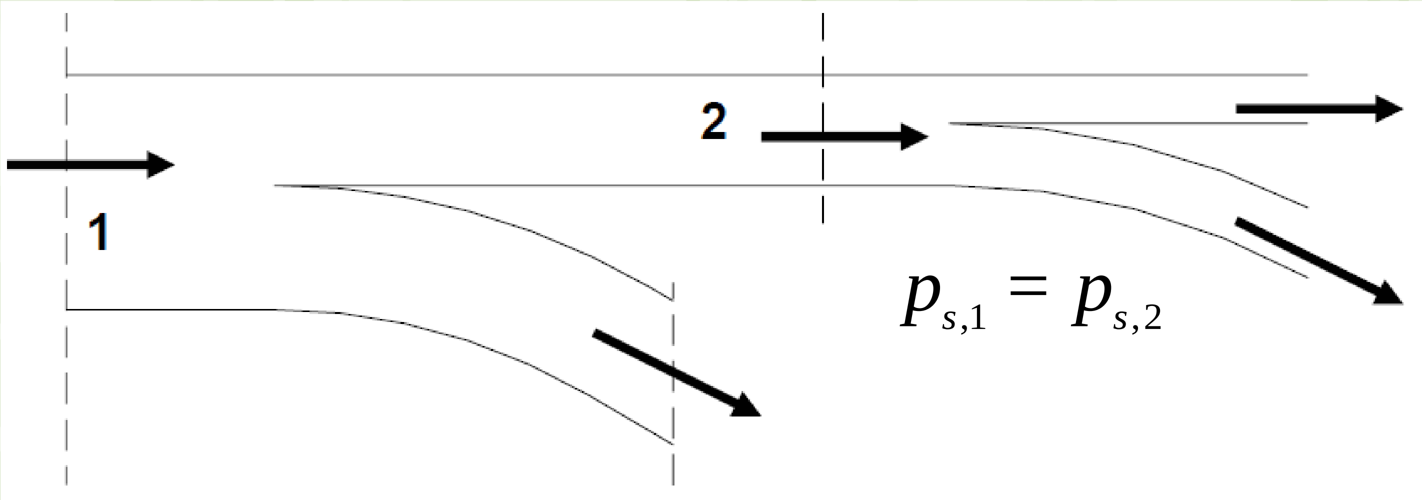
Design Procedures

- Velocity in the main duct leaving the fan is selected first.
- Velocities in each successive runs are reduced such that the gain in static pressure due to reduction in velocity pressure equals the frictional pressure drop in the next duct section. Thus the static pressure before each terminal or branch is maintained constant. For example, Figure shows a part of the duct run with two sections 1 and 2 before two branch take-offs. The velocity at 1 is greater than that at 2, such that the static pressure is same at 1 and 2. Then using the static regain factor, one can write:

$$\Delta p_{f,2} + \Delta p_{d,2} = R(p_{v,1} - p_{v,2})$$



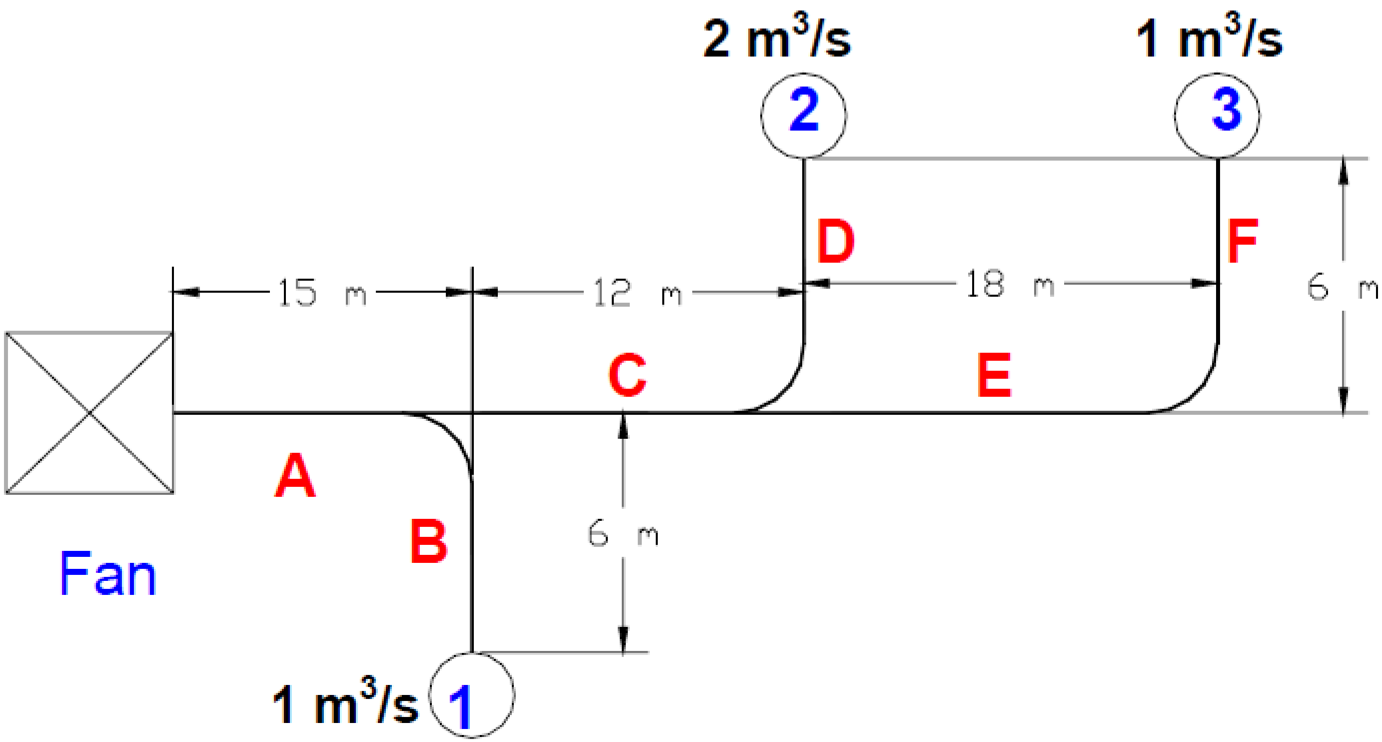
- where $\Delta p_{f,2}$ and $\Delta p_{d,2}$ are the frictional and dynamic losses between 1 and 2, and $p_{v,1}$ and $p_{v,2}$ are the velocity pressures at 1 and 2 respectively.



Principle of static regain method

- If section 1 is the outlet of the fan, then its dimensions are known from the flow rate and velocity (initially selected), however, since both the dimensions and velocity at section 2 are not known, a trial-and-error method has to be followed to solve the above equation, which gives required dimensions of the section at 2.
- The procedure is followed in the direction of airflow, and the dimensions of the downstream ducts are obtained.
- As before, the total pressure drop is obtained from the pressure drop in the longest run and a fan is accordingly selected.

10. The following figure shows a typical duct layout. Design the duct system using a) Velocity method, and b) Equal friction method. Take the velocity of air in



the main duct (A) as 8 m/s for both the methods. Assume a dynamic loss coefficient of 0.3 for upstream to downstream and 0.8 for upstream to branch and for the elbow. The dynamic loss coefficients for the outlets may be taken as 1.0. Find the FTP required for each case and the amount of dampering required.