Lesson
38
Design Of Air Conditioning Ducts

The specific objectives of this chapter are to:

1. Important requirements of an air conditioning duct (*Section 38.1*)

2. General rules for duct design (*Section 38.2*)

3. Classification of duct systems (*Section 38.3*)

4. Commonly used duct design methods (*Section 38.4*)

5. Principle of velocity method (*Section 38.4.1*)

6. Principle of equal friction method (*Section 38.4.2*)

7. Principle of static regain method (*Section 38.4.3*)

8. Performance of duct systems (*Section 38.5*)

9. System balancing and optimization (*Section 38.6*)

10. Introduction to fans and fan laws (*Section 38.7*)

11. Interaction between fan and duct system (*Section 38.8*)

At the end of the chapter, the student should be able to:

1. State the important requirements of an air conditioning duct and the general rules to be followed in the design of ducts
2. Classify air conditioning ducts based on air velocity and static pressure
3. Design air conditioning ducts using velocity method, equal friction method or static regain method
4. Explain typical performance characteristics of a duct system
5. Explain the importance of system balancing and optimization
6. State and explain the importance of fan laws, and use the performance of fans under off-design conditions
7. Describe interaction between fan and duct and the concept of balance point

38.1. Introduction:

The chief requirements of an air conditioning duct system are:

1. It should convey specified rates of air flow to prescribed locations

2. It should be economical in combined initial cost, fan operating cost and cost of building space

3. It should not transmit or generate objectionable noise

Generally at the time of designing an air conditioning duct system, the required airflow rates are known from load calculations. The location of fans and air outlets are fixed initially. The duct layout is then made taking into account the space available and ease of construction. In principle, required amount of air can be conveyed through the air conditioning ducts by a number of combinations. However, for a given system, only one set results in the optimum design. Hence, it is essential to identify the relevant design parameters and then optimize the design.

38.2. General rules for duct design:

1. Air should be conveyed as directly as possible to save space, power and material

2. Sudden changes in directions should be avoided. When not possible to avoid sudden changes, turning vanes should be used to reduce pressure loss

3. Diverging sections should be gradual. Angle of divergence $\leq 20^{o}$

4. Aspect ratio should be as close to 1.0 as possible. Normally, it should not exceed 4

5. Air velocities should be within permissible limits to reduce noise and vibration

6. Duct material should be as smooth as possible to reduce frictional losses

38.3. Classification of duct systems:

Ducts are classified based on the load on duct due to air pressure and turbulence. The classification varies from application to application, such as for residences, commercial systems, industrial systems etc. For example, one such classification is given below:

<u>Low pressure systems:</u> Velocity $\leq$ 10 m/s, static pressure $\leq$ 5 cm H_2O (g)
<u>Medium pressure systems:</u> Velocity $\leq$ 10 m/s, static pressure $\leq$ 15 cm H_2O (g)

<u>High pressure systems:</u> Velocity > 10 m/s, static pressure 15<p_s $\leq$ 25 cm H_2O (g)

High velocities in the ducts results in:

1. Smaller ducts and hence, lower initial cost and lower space requirement

2. Higher pressure drop and hence larger fan power consumption

3. Increased noise and hence a need for noise attenuation

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

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.

38.4. Commonly used duct design methods:

Figure 38.1 shows the schematic of a typical supply air duct layout. As shown in the figure, supply air from the fan is distributed to five outlets (1 to 5), which are located in five different conditioned zones. The letters A to I denote the portions of the duct to different outlets. Thus A-B is the duct running from the supply air fan to zone 1, A-B-C is the duct running from supply fan to conditioned zone and so on. These are known as duct runs. The run with the highest pressure drop is called as the **index run**. From load and psychrometric calculations the required supply airflow rates to each conditioned space are known. From the building layout and the location of the supply fan, the length of each duct run is known. The purpose of the duct design is to select suitable

dimensions of duct for each run and then to select a fan, which can provide the required supply airflow rate to each conditioned zone.

Due to the several issues involved, the design of an air conditioning duct system in large buildings could be a sophisticated operation requiring the use of Computer Aided Design (CAD) software. However, the following methods are most commonly used for simpler lay-outs such as the one shown in Fig.38.1.

1. Velocity method
2. Equal Friction Method
3. Static Regain method
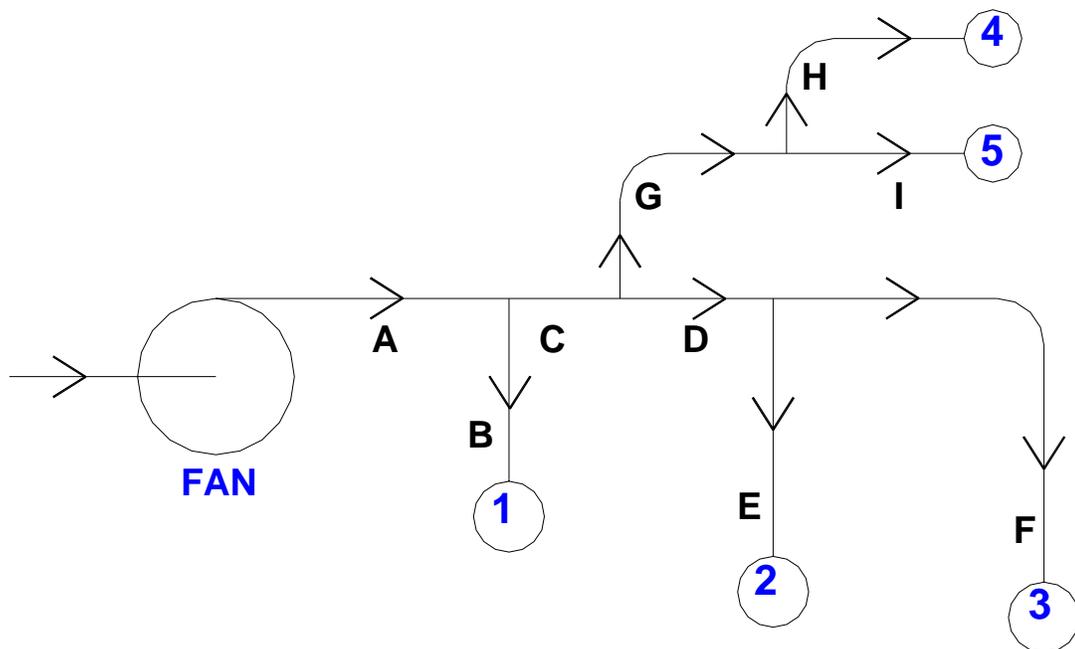
Fig.38.1: Typical air conditioning duct lay-out

38.4.1. Velocity method:

The various steps involved in this method are:

i. Select suitable velocities in the main and branch ducts

ii. 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

iii. 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.

iv. From the duct layout, dimensions and airflow rates, find the dynamic pressure losses for all the bends and fittings

v. Select a fan that can provide sufficient FTP for the index run

vi. 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.

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.

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.

38.4.2. 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.,

$$\left(\frac{\Delta p_f}{L}\right)_A = \left(\frac{\Delta p_f}{L}\right)_B = \left(\frac{\Delta p_f}{L}\right)_C = \left(\frac{\Delta p_f}{L}\right)_D = \ldots \qquad (38.1)$$

Then the stepwise procedure for designing the duct system is as follows:

i. Select a suitable frictional pressure drop per unit length ($\Delta p_f/L$) so that the combined initial and running costs are minimized.

ii. Then the equivalent diameter of the main duct (A) is obtained from the selected value of ($\Delta p_f/L$) and the airflow rate. As shown in Fig.38.1, 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.,

$$\dot{Q}_A = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_4 + \dot{Q}_5 = \sum_{i=1}^{N} \dot{Q}_i \qquad (38.2)$$

From the airflow rate and ($\Delta p_f/L$) 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.,

$$D_{eq,A} = \left(\frac{0.022243\, \dot{Q}_A{}^{1.852}}{\left(\dfrac{\Delta p_f}{L}\right)_A} \right)^{\left(1/4.973\right)} \qquad (38.3)$$

iii. 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:

$$\left(\frac{\dot{Q}^{1.852}}{D_{eq}{}^{4.973}} \right)_A = \left(\frac{\dot{Q}^{1.852}}{D_{eq}{}^{4.973}} \right)_B = \left(\frac{\dot{Q}^{1.852}}{D_{eq}{}^{4.973}} \right)_C = \ldots \qquad (38.4)$$

iv. 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.

v. 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.,

$$\Delta P_{f,A} = \left(\frac{\Delta p_f}{L}\right)_A . L_A \; ; \; \Delta P_{f,B} = \left(\frac{\Delta p_f}{L}\right)_B . L_B \; \ldots \qquad (38.5)$$

vi. Next the dynamic pressure losses in each duct run are obtained based on the type of bends or fittings used in that run.

vii. Next the total pressure drop in each duct run is obtained by summing up the frictional and dynamic losses of that run, i.e.,

$$\Delta P_A = \Delta p_{f,A} + \Delta p_{d,A} ; \; \Delta P_B = \Delta p_{f,B} + \Delta p_{d,B} \ldots \qquad (38.6)$$

viii. 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.

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.

38.4.3. 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. The procedure followed is as given below:

i. Velocity in the main duct leaving the fan is selected first.

ii. 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, Fig.38.2 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:

$$\Delta p_{f,2} + \Delta p_{d,2} = R\left(p_{v,1} - p_{v,2}\right) \qquad (38.7)$$

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.

$$p_{s,1}=p_{s,2}$$
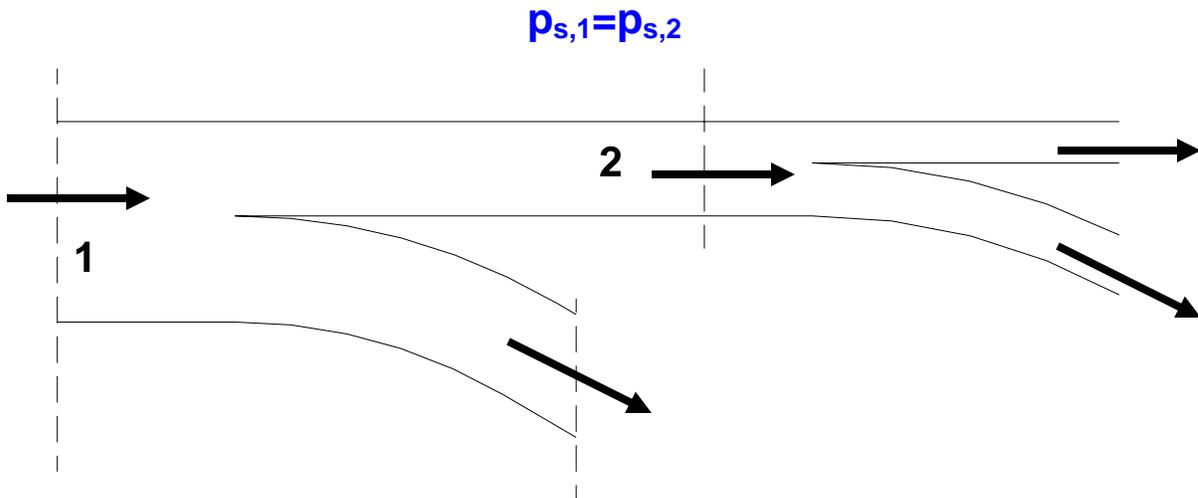
Fig.38.2: *Principle of static regain method*

iii. 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.

iv. The procedure is followed in the direction of airflow, and the dimensions of the downstream ducts are obtained.

v. As before, the total pressure drop is obtained from the pressure drop in the longest run and a fan is accordingly selected.

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.

38.5. Performance of duct systems:

For the duct system with air in turbulent flow, the total pressure loss (Δp_t) is proportional to the square of flow rate; i.e.,

$$\textbf{total pressure drop,} \Delta P_t \propto (\dot{Q})^2 \qquad\qquad (38.8)$$

$$\textbf{or, total pressure drop,} \Delta P_t = C(\dot{Q})^2 \qquad\qquad (38.9)$$

where C is the resistance offered by the duct system. Once the duct system is designed and installed, the value of C is supposed to remain constant. However, if the air filters installed in the duct become dirty and/or if the damper position is altered, then the value of C changes. Thus variation of total pressure drop with airflow rate is parabolic in nature as shown in Fig. 38.3. In this figure, the curve A refers to the performance of the duct at design conditions, while curve B refers to the performance under the conditions of a dirty filter and/or a higher damper closure and curve C refers to the performance when the damper is opened more.

From the duct characteristic curve for constant resistance, one can write

$$\frac{\Delta p_{t,1}}{\Delta p_{t,2}} = \frac{(\dot{Q}_1)^2}{(\dot{Q}_2)^2} \qquad\qquad (38.10)$$

Thus knowing the total pressure drop and airflow rate at design condition (say 1), one can obtain the total pressure drop at an off-design condition 2, using the above equation.
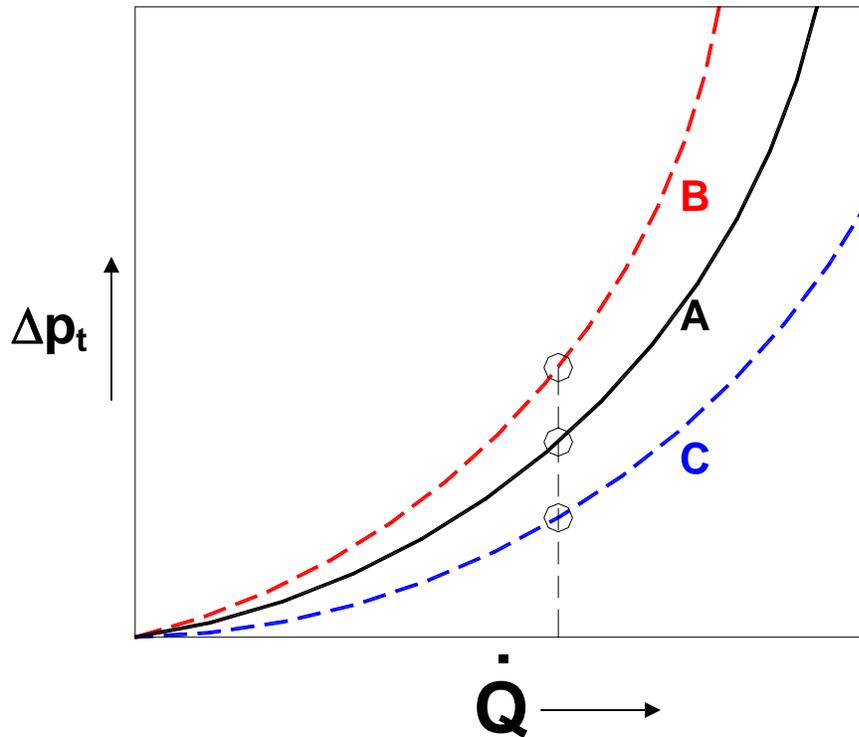
Fig.38.3: *Variation of total pressure drop with flow rate for a given duct system*

38.6. System balancing and optimization:

In large buildings, after the Air Handling Unit is installed, it has to be balanced for satisfactory performance. System balancing requires as a first step, measurements of actual airflow rates at all supply air outlets and return air inlets. Then the dampers are adjusted so that the actual measured flow rate corresponds to the specified flow rates. System balancing may also require adjusting the fan speed to get required temperature drop across the cooling or heating coils and required airflow rates in the conditioned zone. Balancing a large air conditioning system can be a very expensive and time consuming method and may require very accurate instruments for measuring air flow rates and temperatures. However, system balancing is always recommended to get the full benefit from the total cost incurred on air conditioning system.

Large air conditioning systems require optimization of the duct design so as to minimize the total cost, which includes the initial cost of the system and the lifetime operating cost. At present very sophisticated commercial computer software are available for optimizing the duct design. One such method is called as T-Method. The reader should refer to advanced textbooks or ASHRAE handbooks for details on duct optimization methods.

38.7. Fans:

The fan is an essential and one of the most important components of almost all air conditioning systems. Thus a basic understanding of fan performance characteristics is essential in the design of air conditioning systems. The centrifugal fan is most commonly used in air conditioning systems as it can efficiently move large quantities of air over a large range of pressures. The operating principle of a centrifugal fan is similar to that of a centrifugal compressor discussed earlier. The centrifugal fan with forward-curved blades is widely used in low-pressure air conditioning systems. The more efficient backward-curved and airfoil type fans are used in large capacity, high-pressure systems.

38.7.1. Fan laws:

The fan laws are a group of relations that are used to predict the effect of change of operating parameters of the fan on its performance. **The fan laws are valid for fans, which are geometrically and dynamically similar**. The fan laws have great practical use, as it is not economically feasible to test fans of all sizes under all possible conditions.

The important operating parameters of a fan of **fixed diameter** are:

1. Density of air (ρ) which depends on its temperature and pressure
2. Operating speed of the fan (ω in rps), and
3. Size of the fan.

Here the fan laws related to the density of air and the rotative speed of the fan are considered. The effect of the size of the fan is important at the time of designing the fan. For a given air conditioning system with fixed dimensions, fittings etc. it can be easily shown that:

$$\textbf{airflow rate,} \, \dot{\textbf{Q}} \propto \omega \qquad\qquad (38.11)$$

$$\textbf{static pressure rise,} \, \Delta\textbf{p}_\textbf{s} \propto \frac{\rho \textbf{V}^2}{2} \qquad\qquad (38.12)$$

$$\textbf{fan power input,} \, \dot{\textbf{W}} \propto \dot{\textbf{Q}}(\Delta\textbf{p}_\textbf{s}) + \dot{\textbf{Q}}\left(\frac{\rho \textbf{V}^2}{2}\right) \qquad (38.13)$$

From the expression for fan power input (Eqn.(38.13)), it can be seen that the 1st term on the RHS accounts for power input required for increasing the static pressure of air and the 2nd term on RHS accounts for the power input required to impart kinetic energy to air as it flows through the fan. Using the above relations, the following fan laws can be obtained.

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

$$\dot{Q} \propto \omega; \quad \Delta p_s \propto \omega^2 \quad \text{and} \quad \dot{W} \propto \omega^3 \tag{38.14}$$

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

$$\dot{Q} = \text{constant}; \quad \Delta p_s \propto \rho \quad \text{and} \quad \dot{W} \propto \rho \tag{38.15}$$

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

$$\dot{Q} \propto \frac{1}{\sqrt{\rho}}; \quad \Delta p_s = \text{constant}, \quad \omega \propto \frac{1}{\sqrt{\rho}} \quad \text{and} \quad \dot{W} \propto \frac{1}{\sqrt{\rho}} \tag{38.16}$$

38.8. Interaction between fan and duct system:

Figure 38.4 shows the variation of FTP of a centrifugal fan (fan performance curve) and variation of total pressure loss of a duct system (duct performance curve) as functions of the airflow rate. As shown in the figure, the point of intersection of the fan performance curve and the duct performance curve yield the **balance point** for the combined performance of fan and duct system. Point 1 gives a balance point between the fan and duct system when the rotative speed of fan is ω_1. At this condition the airflow rate is Q_1 and the total
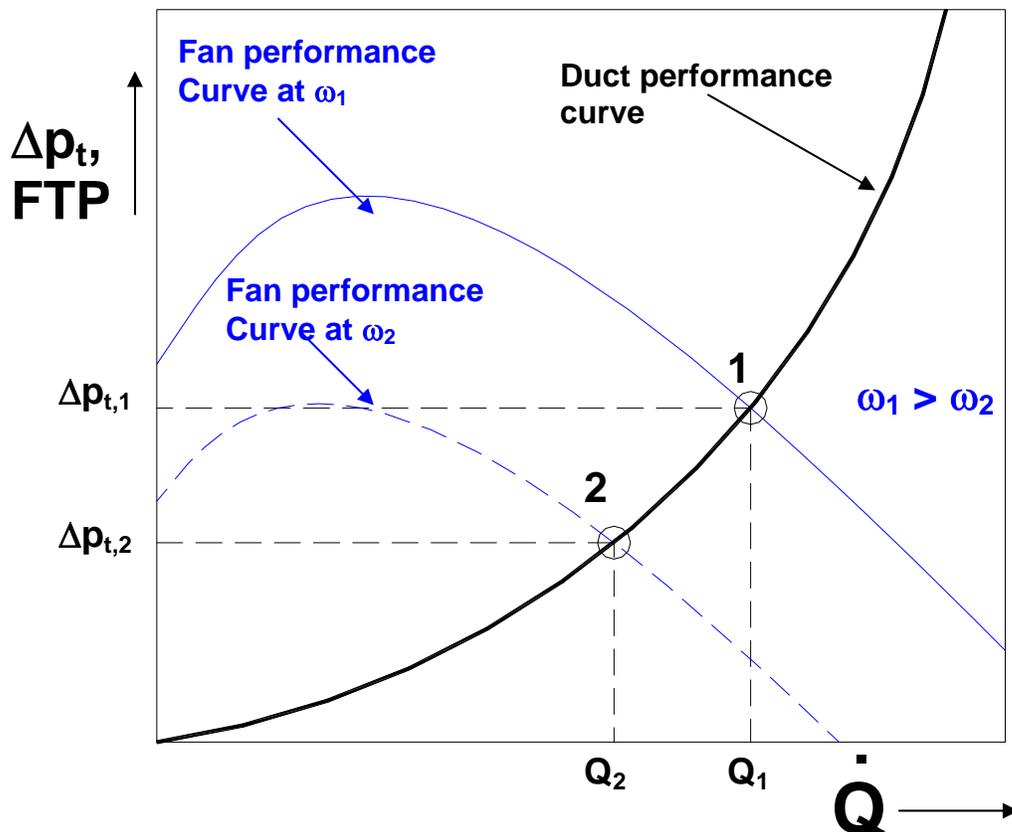
Fig.38.4: *Fan and duct performance curves and balance points*

pressure loss which is equal to the FTP is $\Delta p_{t,1}$. Now if the flow rate is reduced to Q_2, then the total pressure loss reduces to $\Delta p_{t,2}$. To match the reduced flow rate and the reduced pressure loss, the speed of the fan has to be reduced to ω_2 or the position of the inlet guide vanes of the centrifugal fan have to be adjusted to reduce the flow rate. This will give rise to a new balance point at 2. Thus the fan and duct system have to be matched when there is a change in the operating conditions.

Questions and answers:

1. State which of the following statements are TRUE?

a) The air conditioning duct should have high aspect ratio for good performance
b) If the air conditioning duct is diverging, then the angle of divergence should be as small as possible to reduce pressure loss
c) To minimize noise and vibration, air should flow with a low velocity
d) All of the above

Ans.: b) and c)

2. State which of the following statements are TRUE?

a) High air velocity in ducts results in lower initial costs but higher operating costs
b) Higher air velocities may result in acoustic problems
c) Air velocities as high as 30 m/s are used in residential systems
d) Low air velocities are recommended for recording studios

Ans.: a), b) and d)

3. State which of the following statements are TRUE?

a) In a duct layout, the total pressure drop is maximum in the index run
b) At balanced condition, the total pressure drop is equal for all duct runs
c) Dampers are required for balancing the flow in each duct run
d) All of the above

Ans.: d)

4. State which of the following statements are TRUE?

a) If not done properly, the velocity method gives rise to large sized ducts
b) In equal friction method, dampering is not required
c) In static regain method, dampering is required
d) All of the above

Ans.: c)

5. State which of the following statements are TRUE?

a) In a given duct system, the total pressure drop varies linearly with flow rate
b) In a given duct system, the total pressure drop varies in a parabolic manner with flow rate
c) For a given flow rate, the total pressure drop of a duct increases as the dampers are opened more
d) For a given flow rate, the total pressure drop of a duct is less when the air filters are new

Ans.: b) and d)

6. State which of the following statements are TRUE?

a) Compared to forward curved blades, backward curved blades are more efficient
b) Airfoil type blades are used in small capacity systems
c) Fan laws are applicable to all types of fans
d) Fan laws are applicable to fans that are geometrically and dynamically similar

Ans.: a) and d)

7. State which of the following statements are TRUE?

a) For a given fan operating at a constant temperature, the power input to fan increases by 4 times when the fan speed becomes double
b) For a given fan operating at a constant temperature, the power input to fan increases by 8 times when the fan speed becomes double
c) For a given fan operating at a constant flow rate, the power input increases as the air temperature increases
d) For a given fan operating at a constant static pressure rise, the flow rate reduces as the air temperature increases

Ans.: b)

8. State which of the following statements are TRUE?

a) For a backward curved blade, the fan total pressure (FTP) increases as flow rate increases
b) For a backward curved blade, the fan total pressure (FTP) reaches a maximum at a particular flow rate
c) When the air filter in the air conditioning duct becomes dirty, the speed has to be increased to maintain the balance between fan and duct systems
d) When the damper installed in the duct is opened more, to maintain the balance, the speed of the fan should be increased
Ans.: b) and c)

9. Find the dimensions of a rectangular duct of aspect ratio (1:2) when 0.2 m^3/s of air flows through it. The allowable frictional pressure drop is 3 Pa/m.

Ans: For a flow rate of **0.2 m^3/s** and an allowable frictional pressure drop of 3 Pa/m, the equivalent diameter is found to be **0.2 m from** friction chart or friction equation.

Then taking an **aspect ratio of 1:2**, the dimensions of the rectangular duct are found to be :

$$a \approx 0.13 \text{ m and } b \approx 0.26 \text{ m.} \qquad \text{(Ans.)}$$

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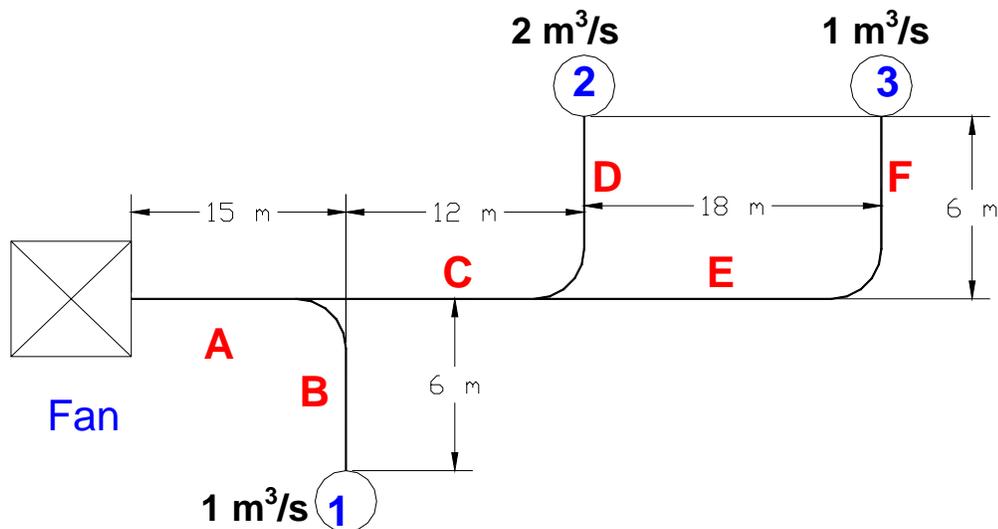
the main duct (A) as 8 m/s for both the methods. Assume a dynamic loss coefficient of 0.3 for upstream to downstream and 0.8 for upstream to branch and for the elbow. The dynamic loss coefficients for the outlets may be taken as 1.0. Find the FTP required for each case and the amount of dampering required.

Ans.:

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$^2 \Rightarrow$ **$D_{eq,A}$ = 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 $A_B = Q_B/V_B = 1/5 = 0.2\ m^2 \Rightarrow$ **$D_{eq,B}$ = 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 $A_C = Q_C/V_C = 3/5 = 0.6\ m^2 \Rightarrow$ **$D_{eq,A}$ = 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 $A_D = Q_D/V_D = 2/5 = 0.4\ m^2 \Rightarrow$ **$D_{eq,D}$ = 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 $A_{E,F} = Q_{E,F}/V_{E,F} = 1/5 = 0.2\ m^2 \Rightarrow$ **$D_{eq,A}$ = 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:

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

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

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

$$\Delta p_{u\text{-}b} = C_{u\text{-}b}\left(\frac{\rho V_d^2}{2}\right) = 0.8\left(\frac{1.2 \times 5^2}{2}\right) = \textbf{12 Pa}$$

The dynamic pressure drop at the exit is given by:

$$\Delta p_{exit,1} = C_{exit}\left(\frac{\rho V_1^2}{2}\right) = 1.0\left(\frac{1.2 \times 5^2}{2}\right) = \textbf{15 Pa}$$

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 the FTP required is:

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

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.)**

b) Equal Friction Method:

The frictional pressure drop in segment A is given by:

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

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:

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

From the above equation, we obtain:

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

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

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

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

Calculation of total pressure drop:

From fan to 1:

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

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

From fan to 2:

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

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

From fan to exit 3:

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

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

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

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

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.

11. A fan is designed to operate at a rotative speed of 20 rps. At the design conditions the airflow rate is 20 m^3/s, the static pressure rise is 30 Pa and the air temperature is 20°C. At these conditions the fan requires a power input of 1.5 kW. Keeping the speed constant at 20 rps, if the air temperature changes to 10°C, what will be the airflow rate, static pressure and power input?

Ans: At design condition 1, Rotative speed, ω_1 = 20 rps
                        Air temperature, T_1 = 20°C = 293 K
                        Airflow rate, Q_1 = 20 m^3/s
                        Static pressure rise, $\Delta p_{s,1}$ = 30 Pa
                        Power input, W_1 = 1.5 kW

At off-design condition 2, the rotative speed is same as 1, but temperature changes to 10°C (283 K), which changes the density of air. To find the other variables, the fan law 2 has to be applied as density varies; i.e.,

i) Airflow rate $\mathbf{Q_1 = Q_2 = 20\ m^3/s}$

ii) Static pressure rise at 2,

     $\Delta p_{s,2}$ $= \Delta p_{s,1}(\rho_2/\rho_1) = \Delta p_{s,1}(T_1/T_2) = 30(293/283) =$ **31.06 Pa**   **(Ans.)**

iii) Power input at 2,

     $W_2 = W_1(\rho_2/\rho_1) = W_1(T_1/T_2) = 1.5(293/283) =$ **1.553 kW**   **(Ans.)**