# Mechanical ventilation

# Measurements of airflow

Air flow can be measured in terms of:
1. Volumetric flow
2. Mass airflow

Units of volumetric airflow rates are generally expressed in:

1. Litres per second (l/s)
2. Cubic metres per second ($m^3$/s)
3. Litres per hour (l/h)
4. Cubic metres per hour ($m^3$/h).

*NOTE: 1000 litres = 1m$^3$.*

Mass airflow rates are frequently used for air conditioning calculations, but are rarely used for general mechanical ventilation calculations.

Mass airflow rates are generally expressed in kilograms per second (kg/s).

Where a mass airflow rate is used, it can be converted to volumetric airflow by multiplying the value, by the density of the air being considered. For general ventilation calculations, the density of air can be assumed at 1,2m$^3$/kg.

The airflow rate into a room space, for general mechanical supply and extract systems, is usually expressed in:

1. Air changes per hour
2. An airflow rate per person
3. An airflow rate per unit floor area.

Air changes per hour (Ach/h) is the most frequently used basis for calculating the required airflow. Air changes per hour are the number of times in one hour an equivalent room volume of air will be introduced into, or extracted from, the room space.

Airflow rates per person are generally expressed as litres per person (l/p), and are usually used only where fresh air ventilation is required within occupied spaces.

Airflow rates per unit floor area are similar in effect to air changes per hour except that the height of the room is not taken into consideration.

## Calculating room ventilation rates using air changes per hour

**Formula 1**

The formula for calculating the ventilation rate is:

Q = v x n

Where Q is the ventilation rate in m$^3$/h
v is the volume of the room in m$^3$
n is the air change rate (Ach/h)

**Example 1**

Calculate the ventilation rate within a toilet 2.2m long x 1.9m wide x 2.7m high, to provide 6 air changes per hour:

Q = v x n
= 2.2 x 1.9 x 2.7 x 6
= 67.72m$^3$/h

## Calculating room ventilation rates using airflow per person

**Formula 2**

Q = q x n

Where Q is the ventilation rate in l/s
q is the ventilation rate per person in l/s per person
n is the number of people in the room.

**Example 2**

Calculate the fresh air ventilation rate required within an office which is designed to accommodate 5 people. The fresh airflow rate required is 12 litres per second, per person.

Q = q x n
= 5x12
= 60 l/s

## Calculating room ventilation rates using airflow per unit floor area

**Formula 3**

The formula for calculating the ventilation rate is:

Q = q x a

Where Q is the ventilation rate in m$^3$/h
q is the airflow rate in m$^3$/h per m$^2$ of floor area
a is the room floor area in m$^2$

**Example 3**

Calculate the ventilation rate within a restaurant 8.5m long x 5.7m wide to provide an airflow of 40m$^3$/h per m$^2$ of floor area.

Q = q x a
= 40 x 8.5 x 5.7
= 1938m$^3$/h

# Extract air ventilation rates – domestic properties

## Building Regulations requirements

| Room Air | Change Rate |
|---|---|
| Toilets | 3Ach/h* |
| Bathrooms/ shower rooms | 15 l/s** |
| Kitchens | 60 l/s or 30 l/s if incorporated within a cooker hood** *Plus* System capable of operating continuously at 1 Ach/h*** |

* *Fan may operate intermittently but must have a 15 minute over-run. Mechanical ventilation is not required if natural ventilation is provided having at least 1/20th of the floor area, some of which must be at least 1.75 metres above floor level.*

** *May be operated intermittently.*

*** *Requirements for continuous operation unnecessary if controllable and secure ventilation openings are provided having a free area not less than 4000mm$^2$, located so as to avoid draughts, e.g.: a trickle ventilator.*

# Extract air ventilation rates – commercial properties

| Room Air | Change Rate |
|---|---|
| Toilets | 3 – 6Ach/h minimum or 6 l/s per WC pan |
| Public toilets | 6 – 12Ach/h |
| Kitchens | 20 – 30Ach/h, 0.35m/s minimum air velocity over hood openings. |
| Offices | 4 – 6Ach/h |
| Meeting rooms/ board rooms | 6 – 10Ach/h |
| Restaurants | 8 – 15Ach/h |
| Laundries | 10 – 15Ach/h |
| Bathrooms/ shower rooms | 3 – 8Ach/h |
| Classrooms | 3 – 6Ach/h |
| Shops | 8 – 12Ach/h |

# Fresh air ventilation rates

| | |
|---|---|
| No smoking areas | 8 l/s per person |
| Some smoking | 16 l/s per person |
| Heavy smoking | 24 l/s per person |
| Very heavy smoking | 32 l/s per person |

# Mechanical ventilation heating load

Mechanical ventilation systems, whether they be supply or extract systems, will have an effect on the heating load within the building they serve.

For example, with small supply air systems, the fresh air drawn from outside can be discharged into the unheated room space. The air introduced will then form part of the room heat load in a similar way to fresh air infiltration. Care must be exercised to ensure that the cold air being discharged will not cause draughts or discomfort.

For larger systems, it is necessary to pre-heat the air before it is discharged into the room space. This is to avoid draughts and also the possibility of condensation as the warm moist air within the room comes into contact with the cold surfaces of the supply air system.

It is normal practice to pre-heat the air to at least room temperature. When heated to room temperature, the supply air will not have any effect on the room heat load. If the supply air is heated to a temperature in excess of the room design temperature, the result will be that useful heat is introduced into the space which can be deducted from the room heating requirements.

The supply air discharge temperature can be generally selected to provide all heating requirements within the room space if required. It must be ensured that the temperature and location of the air being discharged will not cause discomfort. Generally the supply air discharge temperature should not be more than 10°C above room temperature, where the ceiling height is less than 2.7m.

If the supply air is below room temperature, then the balance to heat the air to room temperature must be added to the room heat load.

Pre-heating of the air can be achieved by a water heating coil, an electric heater battery or a heat reclaim device.

Various proprietary heat reclaim devices are available which transfer heat from the warm air within extract systems to heat the incoming cold supply air.

The following should be considered when selecting a heat reclaim device:

1. The device will not be 100% efficient and therefore the supply air will always be at a lower temperature than the extract.
2. Heat reclaim is not required during warm weather. It is therefore necessary to switch off the supply or extract fan, arrange for both fans to be operated in the supply or extract mode simultaneously, or to provide an air by-pass arrangement around the heat reclaim device.

**Formula 4**

The formula for the heat load of air passing through a ventilation system or being introduced into or extracted from a room space is:

$Q = mCp\Delta t$

Where $Q$ = The heating load in kW

$m$ = The mass airflow rate of air in kg/s

$Cp$ = The specific heat capacity of air in kJ/kg°C

$\Delta t$ = The temperature differential of the air under consideration (°C)

The volume airflow rate is converted to mass airflow by multiplying by the density of the air. For general applications this can be taken as $1.2m^3/kg$.

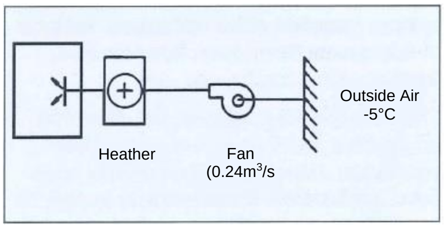
The specific heat capacity of air can be taken as 1.01 kJ/kg°C.

Therefore the formula can be written as:

$Q = q \times 1.2 \times 1.01 \times \Delta t$

Where $q$ = The airflow rate in $m^3/s$.

**Example 4**




*Design fabric and infiltration heat loss from room = 3.7k W. Room temperature 21°C.*

A fresh air mechanical ventilation system supplies $0.24m^3/s$ into a small office. Calculate:

1. The ventilation system heat load to be added to the room heat loss, if a heater is not provided in the system.
2. The required discharge air temperature into the room and the duct heater output if the system is to provide for the design room heat loss of 3.7kW.

***Ventilation system heat load***

From Formula 4

$Q = q \times 1.2 \times 1.01 \times \Delta t$

$= 0.24 \times 1.2 \times 1.01 \times [21 - (-5)]$

$= 7.563kW$

*Table 1 Maximum air velocity and pressure drops in ductwork*

| | Recommended velocity (m/s) | Maximum velocity (m/s) | Recommended pressure drop (Pa/m) | Maximum pressure drop (Pa/m) |
|---|---|---|---|---|
| Main ducts | 5 | 7 | 1.2 | 2 |
| Branch ducts | 3.5 | 5 | 1.0 | 1.5 |
| Louvres | 2.5 | 3 | – | – |
| | (through free airways) | (through free airways) | | |

If the supply air is not pre-heated an additional 7.563kW will need to be added to the room heat loss of 3.7kW.

***Discharge air temperature ($T_s$) to provide 3.7kW heating within room***

$$Q = q \times 1.2 \times 1.01 \times \Delta t$$

$$3.7 = 0.24 \times 1.2 \times 1.01 \times (T_s - 21)$$

3.7

$$T_s = \frac{3.7}{0.24 \times 1.2 \times 1.01} + 21$$

$$= 33.72°C$$

***Duct heater output***

$$Q = 0.24 \times 1.2 \times 1.01 \times [33.72 - (-5)]$$

$$= 11.26kW$$

# Effect of mechanical ventilation on heat producing appliances

It is essential to ensure that any extract system installed does not cause spillage of flue gases from open flued boilers, fires, cooking appliances, etc.

The Building Regulations stipulate that an internal room provided with extract ventilation alone must not contain open flued appliances. If mechanical supply air ventilation is provided to such a room, it must be electrically interlocked with the extract fan(s) and provided with an interlocked supply air flow proving switch to ensure:

1. The supply air system is activated whenever the extract system is operating.
2. The extract fan(s) will not be activated until supply airflow has been proved.
3. The open flued appliances will not operate until supply airflow has been proved.
4. The extract fan(s) and open flued appliances will immediately shut down should the supply airflow proving switch determine that the supply air system has failed.

The Building Regulations also stipulate that a mechanical extract system must not be installed in any room containing a solid fuel burning appliance.

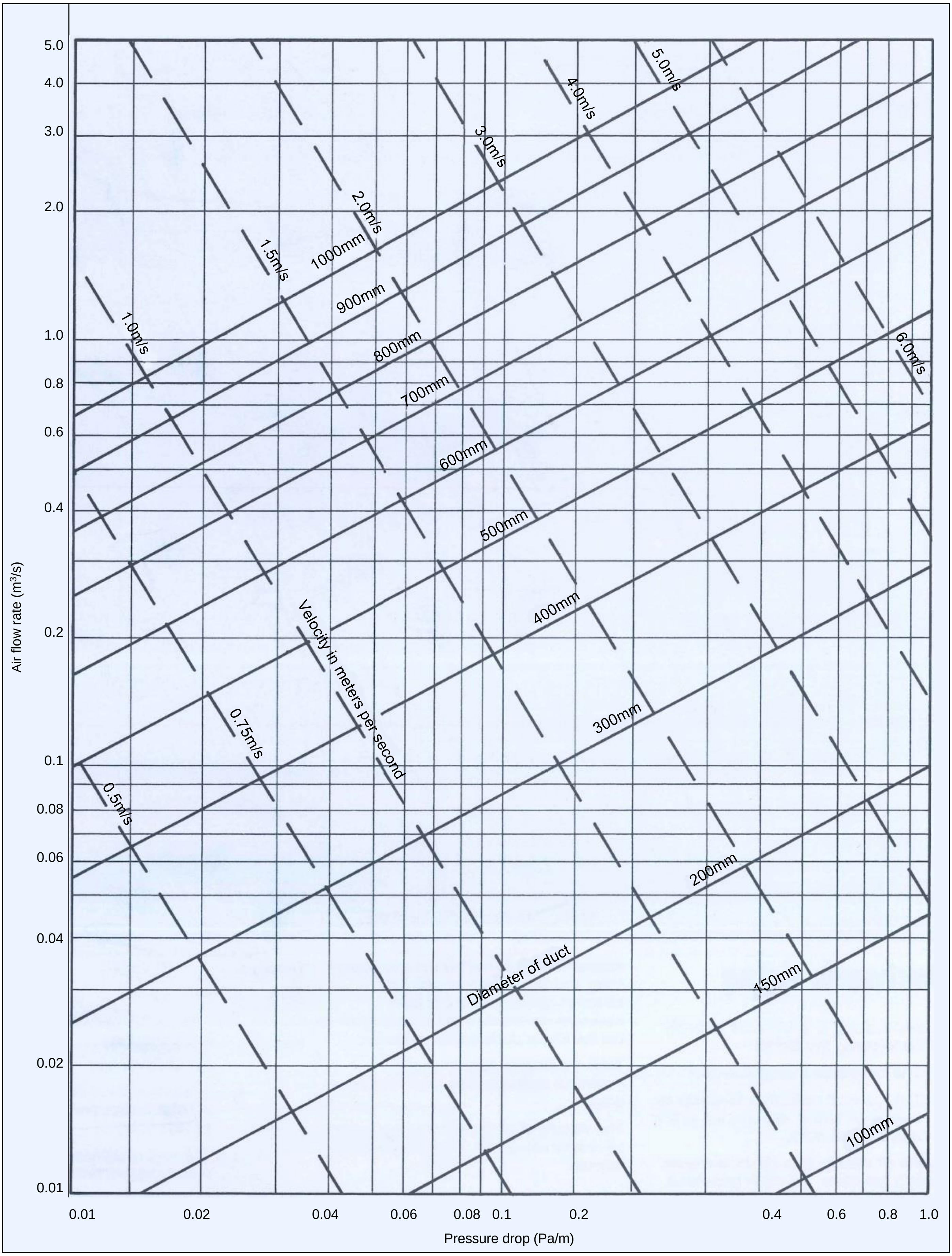
Open flued appliances must be able to operate correctly whether or not the fan(s) is (are) running where natural ventilation and mechanical extract ventilation are provided to the same room space.

**Table 2** *Ductwork fitting K factors (Velocity pressure loss factors). Percentages for branches are based on the change in air velocity of the air stream under consideration.*

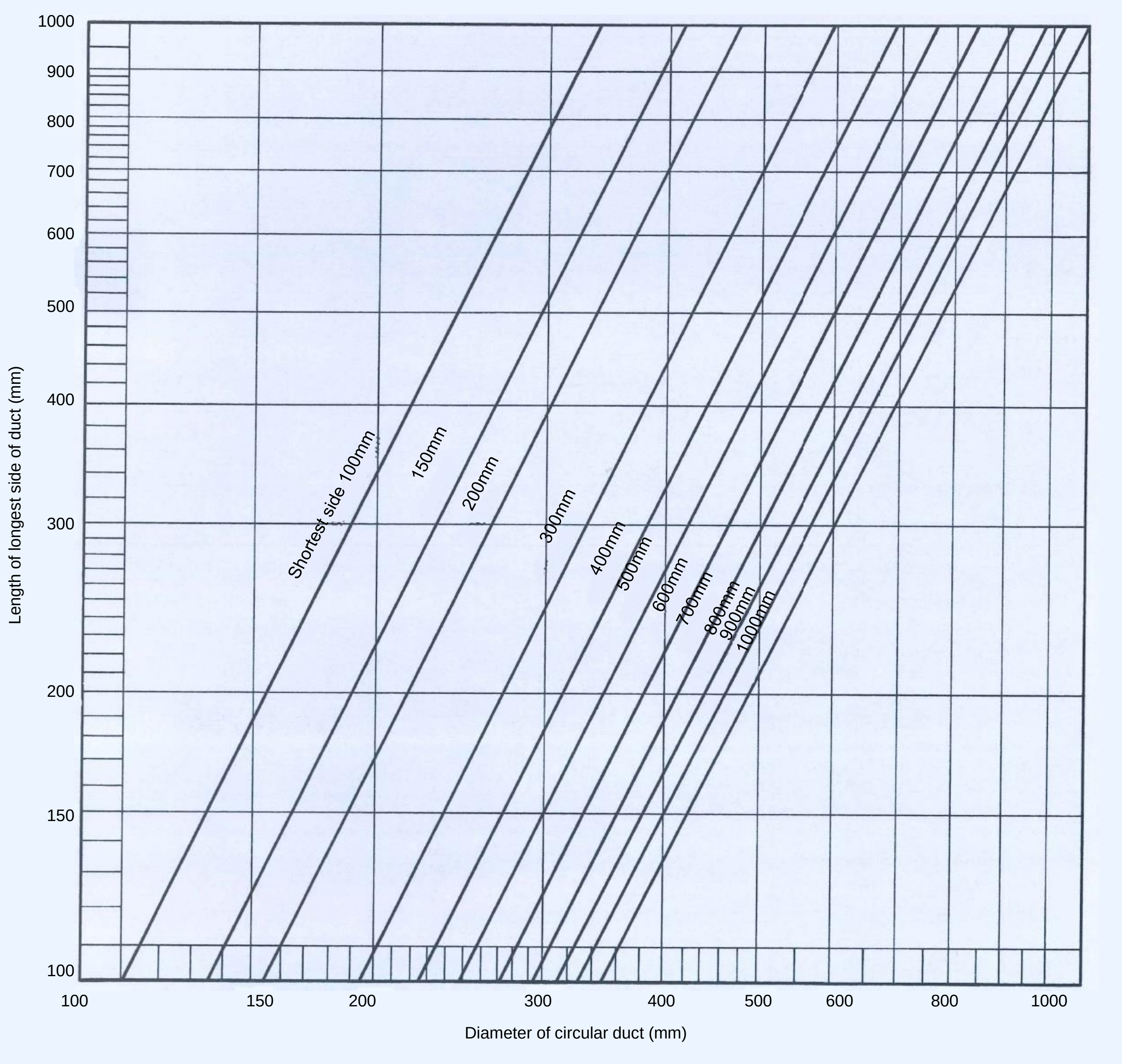
| Circular ducts | | Rectangular ducts | |
|---|---|---|---|
| **Fitting** | **K** | **Fitting** | **K** |
| *Bends 90°* (Radius to inside edge of bend) | | *Bends 90° radius ½ x D* (Radius to inside edge of bend) | |
| Radius ½ x D | 0.34 | Height/width ratio | |
| Radius 1 x D | 0.24 | 0.2 | 0.48 |
| Radius 1½ x D | 0.23 | 0.4 | 0.37 |
| | | 0.6 | 0.28 |
| | | 0.8 | 0.25 |
| | | 1.0 | 0.23 |
| Conical branch with airflow to branch ($V_C$ to $V_B$). Use velocity pressure in $V_B$ | $V_C$ $V_D$ $V_B$ | Shoe branch with airflow to branch ($V_C$ to $V_B$). Use velocity pressure in $V_B$ | $V_C$ $V_D$ $V_B$ |
| 50% | 3.0 | 50% | 2.5 |
| 60% | 2.0 | 60% | 1.7 |
| 70% | 1.5 | 70% | 1.3 |
| 80% | 1.2 | 80% | 0.8 |
| 90% | 1.0 | 90% | 0.7 |
| Conical branch with through airflow ($V_C$ to $V_D$). Use velocity pressure in $V_D$ | $V_C$ $V_D$ $V_B$ | Shoe branch with through airflow ($V_C$ to $V_D$). Use velocity pressure in $V_D$ | $V_C$ $V_D$ $V_B$ |
| 50% | 0.09 | 50% | 1.0 |
| 60% | 0.06 | 60% | 0.45 |
| 70% | 0.03 | 70% | 0.3 |
| 80% | 0 | 80% | 0.1 |
| 90% | 0 | 90% | 0.07 |
| Conical branch with airflow from branch ($V_B$ to $V_D$). Use velocity pressure in $v_D$ | $V_C$ $V_D$ $V_B$ | Shoe branch with airflow from branch ($V_B$ to $V_D$). Use velocity pressure in $v_D$ | $V_C$ $V_D$ $V_B$ |
| 50% | 0.5 | 50% | 0.5 |
| 60% | 0.52 | 60% | 0.52 |
| 70% | 0.53 | 70% | 0.53 |
| 80% | 0.54 | 80% | 0.54 |
| 90% | 0.55 | 90% | 0.55 |
| Conical branch with through airflow ($V_C$ to $V_D$). Use velocity pressure in $V_D$ | $V_C$ $V_D$ $V_B$ | Shoe branch with through airflow ($V_C$ to $V_D$). Use velocity pressure in $V_D$ | $V_C$ $V_D$ $V_B$ |
| 50% | 0.09 | 50% | 0.09 |
| 60% | 0.06 | 60% | 0.06 |
| 70% | 0.03 | 70% | 0.03 |
| 80% | 0 | 80% | 0 |
| 90% | 0 | 90% | 0 |
| *Concentric reducer* (Use velocity pressure at reduction) | | *Concentric reducer* (Use velocity pressure at reduction) | |
| 30° Angle | 0.02 | 30° Angle | 0.02 |
| 45° Angle | 0.04 | 45° Angle | 0.04 |
| *Aerofoil damper* | 0.2 | *Aerofoil damper* | 0.2 |

*Flexible spiral reinforced duct* $0.05 \times \frac{\text{Length (mm)}}{\text{Diameter (mm)}}$

***Graph 1:*** *Sizing charts for circular ducts*



***Graph 2:*** *Relationship between rectangular and circular ducts (equal volume flow rate and unit pressure drop)*




# Ductwork sizing

Ductwork sizing is determined primarily by the following two factors:

1. The air velocity through the duct
2. The resultant resistance to airflow as a result of friction between the airflow and the duct walls.

As the air velocity through the ductwork system increases, the noise generated and the frictional resistance (pressure drop) to airflow will also increase.

It is therefore necessary to limit air velocity to ensure that the system is not excessively noisy and that an economical fan selection can be made. It is also necessary to ensure that the velocity is not too low or ducts will be oversized.

Table 1 indicates recommended maximum duct velocities and pressure drops.

The velocity of air through ductwork can be readily calculated using the following formula:

**Formula 5**

$$v = \frac{q}{a}$$

Where v = The air velocity in metres per second (m/s)

q = The airflow through the duct in cubic metres per second ($m^3/s$)

a = The duct cross sectional area in square metres ($m^2$).

The unit pressure drop through a ductwork system depends to a certain

*Table 3 Velocity pressure (Pa) against velocity*

| Velocity (m/s) | 0.0 | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 0.00 | 0.01 | 0.02 | 0.05 | 0.10 | 0.15 | 0.22 | 0.29 | 0.38 | 0.49 |
| 1 | 0.60 | 0.73 | 0.86 | 1.01 | 1.18 | 1.35 | 1.54 | 1.73 | 1.94 | 2.17 |
| 2 | 2.40 | 2.65 | 2.90 | 3.17 | 3.46 | 3.75 | 4.06 | 4.37 | 4.70 | 5.05 |
| 3 | 5.40 | 5.77 | 6.14 | 6.53 | 6.94 | 7.35 | 7.78 | 8.21 | 8.66 | 9.13 |
| 4 | 9.60 | 10.09 | 10.58 | 11.09 | 11.62 | 12.15 | 12.70 | 13.25 | 13.82 | 14.41 |
| 5 | 15.00 | 15.61 | 16.22 | 16.85 | 17.50 | 18.15 | 18.82 | 19.49 | 20.18 | 20.89 |
| 6 | 21.60 | 22.33 | 23.06 | 23.81 | 24.58 | 25.35 | 26.14 | 26.93 | 27.74 | 28.57 |
| 7 | 29.40 | 30.25 | 31.10 | 31.97 | 32.86 | 33.75 | 34.66 | 35.57 | 36.50 | 37.45 |
| 8 | 38.40 | 39.37 | 40.34 | 41.33 | 42.34 | 43.35 | 44.38 | 45.41 | 46.46 | 47.53 |
| 9 | 48.60 | 49.69 | 50.78 | 51.89 | 53.02 | 54.15 | 55.30 | 56.45 | 57.62 | 58.81 |
| 10 | 60.00 | 61.21 | 62.42 | 63.65 | 64.90 | 66.15 | 67.42 | 68.69 | 69.98 | 71.29 |
| 11 | 72.60 | 73.93 | 75.26 | 76.61 | 77.98 | 79.35 | 80.74 | 82.13 | 83.54 | 84.97 |
| 12 | 86.40 | 87.85 | 89.30 | 90.77 | 92.26 | 93.75 | 95.26 | 96.77 | 98.30 | 99.85 |
| 13 | 101.40 | 102.97 | 104.54 | 106.13 | 107.74 | 109.35 | 110.98 | 112.61 | 114.26 | 115.93 |
| 14 | 117.60 | 119.29 | 120.98 | 122.69 | 124.42 | 126.15 | 127.90 | 129.65 | 131.42 | 133.21 |
| 15 | 135.00 | 136.81 | 138.62 | 140.45 | 142.30 | 144.15 | 146.02 | 147.89 | 149.78 | 151.69 |

extent, on the type of duct material and method of construction.

Graph 1 provides the relationship between airflow, unit pressure drop and air velocity through straight circular ductwork, typical for use in small ductwork systems constructed from PVCu or steel.

The rectangular equivalent of the circular duct section used can be determined from Graph 2.

Pressure loss through ductwork fittings can be calculated by multiplying the fitting K factor listed by Table 2, with the air velocity pressure in the duct.

**Formula 6**

The air velocity pressure can be calculated using the formula:

$$Pv = 1/2 \times \varphi \times v^2$$

Where $P_v$ = The Velocity Pressure in Pascals (Pa)

$\varphi$ = The density of air in $kg/m^3$

v = The air velocity in m/s

For general ventilation systems the density of air can be taken as $1{,}2 kg/m^3$.

Therefore the formula can be simplified to:

$$P_v = 0.6 \times v2$$

The velocity pressure against velocity is tabulated in Table 3.

Pressure loss through grilles, air terminals and louvres should be obtained from the manufacturer.

The individual pressure loss through each element of the ductwork system has to be added together to arrive at the required fan pressure. Where branches are provided within the system, it is necessary to determine which route would be the index run (e.g.: the run with the highest-pressure loss).

It is usual practice when selecting a fan to add a percentage (between 10 and 20%) to the required airflow for ductwork leakage.

# Noise control

Consideration must be given to ensure that vibration and airborne noise generated by a fan will not cause damage or a nuisance.

Vibration transmission can be reduced by providing a sound isolating material between the fan, support structure and connecting ductwork.

This can take the form of:

1. Proprietary anti-vibration mountings constructed from neoprene, metal or plastic springs or similar
2. Neoprene or similar inserts between the fan and fixings
3. Flexible ductwork connections to the fan unit.

It should be ensured that the fan and ductwork system are not fixed to, or in close proximity of any materials which could readily flex and vibrate.

Airborne noise can be reduced, in the first instance, by selecting a fan which has a low sound power level. Tabulated sound power levels in decibels (dB) against frequency (Hz) can be obtained from the fan manufacturer. It should be noted that the published figures can take various forms and it is therefore necessary to ensure that like for like comparisons are made.

Also note that there is a difference between 'sound power' and 'sound pressure' noise levels.

Attenuators (silencers) can be installed within the ductwork to reduce the airborne noise transmission from a fan. Advice regarding attenuator selection should be sought from the fan manufacturers or a specialist.

# Fan types

There are four main types of fans used in ventilation systems:

1. Centrifugal
2. Axial Flow
3. Mixed Flow
4. Propeller.

These four types of fans are further divided into various types, primarily associated with the type of fan blades which they contain.

Generally the type of fan selected will be governed by the fan duty required (airflow and pressure), and the type of application. The following general characteristics of each fan type should be noted:

## Centrifugal fans

a. High airflow

b. High pressure

c. Main sound power generation in lower frequencies, which are more readily controlled by attenuation

d. Un-cased units require a change in ductwork direction from intake to discharge

e. Larger fans are readily available with belt drive motors which can facilitate a future change in fan speed and thus fan duty

f. Backward curved type impellers available which have non-overloading (non-stall) characteristics.

## Axial flow fans

a. High airflow

b. High pressure

c. Fans can be installed in series to increase pressure available

d. Compact design

e. Fan installed in-line, in direction of airflow

f. Fan will stall if ductwork pressure loss is outside fan characteristic

g. Low cost.

## Mixed flow fan

a. High airflow

b. High pressure c . High efficiency

d. Low noise

e. Low energy consumption.

## Propeller fan

a. High airflow

b. Low pressure

c. Low cost

d. Low efficiency

e. Can be noisy.