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10.1. INTRODUCTION

A fan is a device that utilizes the mechanical energy of a rotating impeller to produce both movement of the air and an increase in its total pressure. The great majority of fans used in mining are driven by electric motors, although internal combustion engines may be employed, particularly as a standby on surface fans. Compressed air or water turbines may be used to drive small fans in abnormally gassy or hot conditions, or where an electrical power supply is unavailable.

In Chapter 4, mine fans were classified in terms of their location, main fans handling all of the air passing through the system, booster fans assisting the through-flow of air in discrete areas of the mine and auxiliary fans to overcome the resistance of ducts in blind headings. Fans may also be classified into two major types with reference to their mechanical design.

A centrifugal fan resembles a paddle wheel. Air enters near the centre of the wheel, turns through a right angle and moves radially outward by centrifugal action between the blades of the rotating impeller. Those blades may be straight or curved either backwards or forwards with respect to the direction of rotation. Each of these designs produces a distinctive performance characteristic. Inlet and/or outlet guide vanes may be fitted to vary the performance of a centrifugal fan.

An axial fan relies on the same principle as an aircraft propeller, although usually with many more blades for mine applications. Air passes through the fan along flowpaths that are essentially aligned with the axis of rotation of the impeller and without changing their macro-direction. However, later in the chapter we shall see that significant vortex action may be imparted to the air. The particular characteristics of an axial fan depend largely on the aerodynamic design and number of the impeller blades together with the angle they present to the approaching airstream. Some designs of axial impellers allow the angle of the blades to be adjusted either while stationary or in motion. This enables a single speed axial fan to be capable of a wide range of duties. Axial fan impellers rotate at a higher blade tip speed than a centrifugal fan of similar performance and, hence, tend to be noisier. They also suffer from a pronounced stall characteristic at high resistance. However, they are more compact, can easily be combined into series and parallel configurations and can produce reversal of the airflow by changing the direction of impeller rotation, although at greatly reduced performance. Both types of fan are used as main fans for mine ventilation systems while the axial type is favoured for underground locations.

In this chapter, we shall define fan pressures and examine some of the basic theory of fan design, the results of combining fans in series and parallel configurations, the theory of fan testing and booster fan installations.

10.2. FAN PRESSURES

A matter that has often led to confusion is the way in which fan pressures are defined. In Section 2.3.2. we discussed the concepts of total, static and velocity pressures as applied to a moving fluid. That section should be revised, if necessary, before reading on. While we use those concepts in the definitions of fan pressures, the relationships between the two are not immediately obvious. The following definitions should be studied with reference to Figure 10.1(a) until they are clearly understood.

- **Fan total pressure, \( FTP \),** is the increase in total pressure, \( p_t \), (measured by facing pitot tubes) across the fan,
  \[ FTP = p_{t2} - p_{t1} \]  
  (10.1)

- **Fan velocity pressure, \( FVP \),** is the average velocity pressure at the fan outlet only,
  \[ \bar{p}_{v2} = p_{i2} - p_{o2} \]
• Fan static pressure, $FSP$, is the difference between the fan total pressure and fan velocity pressure, or

$$FSP = FTP - FVP$$

$$= p_{t2} - p_{i1} - (p_{i2} - p_{s2}) = p_{s2} - p_{i1}$$

The reason for defining fan velocity pressure in this way is that the kinetic energy imparted by the fan and represented by the velocity pressure at outlet has, traditionally, been assumed to be a loss of useful energy. For a fan discharging directly to atmosphere this is, indeed, the case. As the fan total pressure, $FTP$, reflects the full increase in mechanical energy imparted by the fan, the difference between the two, i.e. fan static pressure has been regarded as representative of the useful mechanical energy applied to the system.

The interpretations of fan pressures that are most convenient for network planning are further illustrated on Figure 10.1. In the case of a fan located within an airway or ducted at both inlet and outlet (Figure 10.1(a)), the fan static pressure, $FSP$, can be measured directly between a total (facing) tube at inlet and a static (side) tapping at outlet. A study of the diagram and equation (10.3) reveals that this is, indeed, the difference between FTP and FVP.

Figure 10.1 (b) shows the situation of a forcing fan drawing air from atmosphere into the system. A question that arises is where to locate station 1, i.e. the fan inlet. It may be considered to be

(i) immediately in front of the fan
(ii) at the entrance to the inlet cone, or
(iii) in the still external atmosphere

These three positions are labelled on Figure 10.1 (b). If location(i) is chosen then the frictional and shock losses incurred as the air enters and passes through the cone must be assessed separately. At location (ii) the fan and inlet cone are considered as a unit and only the shock loss at entry requires additional treatment. However, if location (iii) is selected then the fan, inlet cone and inlet shock losses are all taken into account. It is for this reason that location (iii) is preferred for the purposes of ventilation planning. Figure 10.1 (b) shows the connection of gauges to indicate the fan pressures in this configuration.

The same arguments apply for a fan that exhausts to atmosphere (Figure 10.1 (c)). If the outlet station is taken to be in the still external atmosphere then the fan velocity pressure is zero and the fan total and fan static pressures become equal. In this configuration the fan total (or static) pressure takes into account the net effects of the fan, frictional losses in the outlet cone and the kinetic energy loss at exit.

During practical measurements, it is often found that turbulence causes excessive fluctuations on the pressure gauge when total pressures are measured directly using a facing pitot tube. In such cases, it is preferable to measure static pressure from side tappings and to add, algebraically, the velocity pressure in order to obtain the total pressure. The mean velocity can be obtained as flowrate divided by the appropriate cross-sectional area. Particular care should be taken with regard to sign. In the case of an exhausting fan, (Figure 10.1 (c)) the static and velocity pressures at the fan inlet have opposite signs.
Figure 10.1 Illustrations of fan pressures.
Fan manufacturers usually publish characteristic curves in terms of fan static pressure rather than the fan total pressure. In addition to being more useful for ventilation planning, this is understandable as manufacturers may have no control over the types of inlet and outlet duct fittings or the conditions at entry or exit to inlet/outlet cones. Where fan velocity pressures are quoted then they are normally referred to a specific outlet location, usually either at the fan hub or at the mouth of an evasee.

The question remains on which fan pressure should be used when a fan is included in the route of a pressure survey. The simple answer is that for main surface fans it is fan static pressure \( (FSP) \) that should be employed. For a main forcing fan the \( FSP \) is given by the gauge static pressure at the inby side of the fan (Figure 10.1(b)). But for a main exhausting fan the \( FSP \) is given by the gauge total pressure at the inby side of the fan (Figure 10.1(c)). This apparent anomaly arises from the way in which fan pressures are defined and is further explained in Appendix A10.1.

10.3. IMPELLER THEORY AND FAN CHARACTERISTIC CURVES

An important aspect of subsurface ventilation planning is the specification of pressure-volume duties required of proposed main or booster fans. The actual choice of particular fans is usually made through a process of perusing manufacturer's catalogues of fan characteristic curves, negotiation of prices and costing exercises (Section 9.5). The theory of impeller design that underlies the characteristic behaviour of differing fan types is seldom of direct practical consequence to the underground ventilation planner. However, a knowledge of the basics of that theory is particularly helpful in discussions with fan manufacturers and in comprehending why fans behave in the way that they do.

This section of the book requires an elementary understanding of vector diagrams. Initially, we shall assume incompressible flow but will take compressibility of the air into account in the later section on fan performance (Section 10.6.1).

10.3.1. The centrifugal impeller

Figure 10.2 illustrates a rotating backward bladed centrifugal impeller. The fluid enters at the centre of the wheel, turns through a right angle and, as it moves outwards radially, is subjected to centrifugal force resulting in an increase in its static pressure. The dotted lines represent flowpaths of the fluid relative to the moving blades. Rotational and radial components of velocity are imparted to the fluid. The corresponding outlet velocity pressure may then be partially converted into static pressure within the surrounding volute or fan casing.

At any point on a flowpath, the velocity may be represented by vector components with respect to either the moving impeller or to the fan casing. The vector diagram on Figure 10.2 is for a particle of fluid leaving the outlet tip of an impeller blade. The velocity of the fluid relative to the blade is \( W \) and has a vector direction that is tangential to the blade at its tip. The fluid velocity also has a vector component in the direction of rotation. This is equal to the tip (peripheral) velocity and is shown as \( u \). The vector addition of the two, \( C \), is the actual or absolute velocity.). The radial (or meridianal) component of velocity, \( C_m \) is also shown on the vector diagram.
10.3.1.1. Theoretical pressure developed by a centrifugal impeller.

A more detailed depiction of the inlet and outlet vector diagrams for a centrifugal impeller is given on Figure 10.3. It is suggested that the reader spend a few moments examining the key on Figure 10.3 and identifying corresponding elements on the diagram.
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Figure 10.3 Velocities and forces on a centrifugal impeller.
In order to develop an expression for the theoretical pressure developed by the impeller, we apply the principle of angular momentum to the mass of fluid moving through it.

If a mass, $m$, rotates about an axis at a radius, $r$, and at a tangential velocity, $v$, then it has an angular momentum of $mrv$. Furthermore, if the mass is a fluid that is continuously being replaced then it becomes a mass flow, $dm/dt$, and a torque, $T$, must be maintained that is equal to the corresponding continuous rate of change of momentum

$$T = \frac{dm}{dt} (rv) \quad \text{Nm or J} \quad (10.4)$$

In the case of the centrifugal impeller depicted in Figure 10.3, the peripheral component of fluid velocity is $C_u$. Hence the torque becomes

$$T = \frac{dm}{dt} (r C_u) \quad \text{Nm or J} \quad (10.5)$$

Consider the mass of fluid filling the space between two vanes and represented as $abcd$ on Figure 10.3. At a moment, $dt$, later it has moved to position $efgh$. The element $abfe$ leaving the impeller has mass $dm$ and is equal to the mass of the element $cdhg$ entering the impeller during the same time. The volume represented by $abgh$ has effectively remained in the same position and has not, therefore, changed its angular momentum. The increase in angular momentum is that due to the elements $abfe$ and $cdhg$. Then, from equation (10.5) applied across the inlet and outlet locations,

$$T = \frac{dm}{dt} [r_2 C_{u2} - r_1 C_{u1}] \quad \text{J} \quad (10.6)$$

Extending the flow to the whole impeller instead of merely between two vanes gives $dm/dt$ as the total mass flow, or

$$\frac{dm}{dt} = Q \rho \quad \frac{\text{kg}}{\text{s}}$$

where $Q$ = volume flow $(\text{m}^3/\text{s})$ and $\rho$ = fluid density $(\text{kg/m}^3)$

giving

$$T = Q \rho [r_2 C_{u2} - r_1 C_{u1}] \quad \text{J} \quad (10.7)$$

Now the power consumed by the impeller, $P_{ow}$ is equal to the rate of doing mechanical work,

$$P_{ow} = T \omega \quad \text{W} \quad (10.8)$$

where $\omega$ = speed of rotation (radians/s)

giving

$$P_{ow} = Q \rho \omega [r_2 C_{u2} - r_1 C_{u1}] \quad \text{W} \quad (10.9)$$

But $r_2 \omega = u_2$ = tangential velocity at outlet and $r_1 \omega = u_1$ = tangential velocity at inlet.

Hence

$$P_{ow} = Q \rho [u_2 C_{u2} - u_1 C_{u1}] \quad \text{W} \quad (10.10)$$
The power imparted by a fan impeller to the air was given by equation (5.56) as
\[ p_{t} Q \]
where \( p_{t} \) = rise in total pressure across the fan.

In the absence of frictional or shock losses, \( p_{t} Q \) must equal the power consumed by the impeller, \( P_{owr} \). Hence
\[ p_{t} = \rho (u_{2} C_{u2} - u_{1} C_{u1}) \quad \text{Pa} \quad (10.11) \]

This relationship gives the theoretical fan total pressure and is known as Euler's equation.

The inlet flow is often assumed to be radial for an ideal centrifugal impeller, i.e. \( C_{u1} = 0 \), giving
\[ p_{t} = \rho u_{2} C_{u2} \quad \text{Pa} \quad (10.12) \]

Euler's equation can be re-expressed in a manner that is more amenable to physical interpretation. From the outlet vector diagram

\[ W_{2}^{2} = C_{m2}^{2} + (u_{2} - C_{u2})^{2} \]
\[ = C_{m2}^{2} + u_{2}^{2} - 2u_{2} C_{u2} + C_{u2}^{2} \]

or
\[ 2u_{2} C_{u2} = u_{2}^{2} - W_{2}^{2} + (C_{m2}^{2} + C_{u2}^{2}) \]
\[ = u_{2}^{2} - W_{2}^{2} + C_{2}^{2} \quad \text{(Pythagorus)} \]

Similarly for the inlet,
\[ 2u_{1} C_{u1} = u_{1}^{2} - W_{1}^{2} + C_{1}^{2} \]

Euler's equation (10.11) then becomes
\[ p_{t} = \rho \left( \frac{u_{2}^{2} - u_{1}^{2}}{2} - \frac{W_{2}^{2} - W_{1}^{2}}{2} + \frac{C_{2}^{2} - C_{1}^{2}}{2} \right) \quad \text{Pa} \quad (10.13) \]

\[ \frac{\text{centrifugal effect}}{\text{effect of relative velocity}} + \frac{\text{change in kinetic energy}}{\text{Gain in static pressure}} + \text{Gain in velocity pressure} \]

10.3.1.2. Theoretical characteristic curves for a centrifugal impeller

Euler's equation may be employed to develop pressure-volume relationships for a centrifugal impeller. Again, we must first eliminate the \( C_{u} \) term. From the outlet vector diagram on Figure 10.3,

\[ \tan \beta_{2} = \frac{C_{m2}}{u_{2} - C_{u2}} \]
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\[ C_{u2} = u_2 - \frac{C_{m2}}{\tan \beta_2} \]

giving \( C_{u2} = u_2 - \frac{C_{m2}}{\tan \beta_2} \)

For radial inlet conditions, Euler's equation (10.12) then gives

\[ p_f = \rho u_2 C_{u2} = \rho u_2 \left( u_2 - \frac{C_{m2}}{\tan \beta_2} \right) \] Pa \hspace{1cm} (10.14)

But \( C_{m2} = \frac{Q}{a_2} = \frac{\text{volume flow rate}}{\text{flow area at impeller outlet}} \)

Equation (10.14) becomes

\[ p_f = \rho u_2^2 - \frac{\rho u_2 Q}{\tan \beta_2 a_2} \] Pa \hspace{1cm} (10.15)

For a given impeller rotating at a fixed speed and passing a fluid of known density, \( \rho, u, a \) and \( \beta \)
are all constant, giving

\[ p_f = A - BQ \] Pa \hspace{1cm} (10.16)

where constants \( A = \rho u_2^2 \)

and \( B = \frac{\rho u_2}{a_2 \tan \beta_2} \)

The flowrate, \( Q \), and, hence, the pressure developed vary with the resistance against which the
fan acts. Equation (10.16) shows that if frictional and shock losses are ignored, then fan pressure
varies linearly with respect to the airflow.

We may apply this relationship to the three types of centrifugal impeller:

**Radial bladed**

\( \beta_2 = 90^\circ \) and \( \tan \beta_2 = \infty \)

giving \( B = 0 \)

Then \( p_f = \text{constant} A = \rho u_2^2 \)

i.e. theoretically, the pressure remains constant at all flows [Figure 10.4 (a)]

**Backward bladed**

\( \beta_2 < 90^\circ, \quad \tan \beta_2 > 0 \)

\( p_f = A - BQ \)

i.e. theoretically, the pressure falls with increasing flow.

**Forward bladed**

\( \beta_2 > 90^\circ, \quad \tan \beta_2 < 0 \)

\( p_f = A + BQ \)

i.e. theoretically, pressure rises linearly with increasing flow.

The latter and rather surprising result occurs because the absolute velocity, \( C_2 \) is greater than the
impeller peripheral velocity, \( u_2 \), in a forward bladed impeller. This gives an impulse to the fluid
which increases with greater flowrates. (In an actual impeller, friction and shock losses more than
counteract the effect.).
The theoretical pressure-volume characteristic curves are shown on Figure 10.4 (a).

The theoretical relationship between impeller power and airflow may also be gained from equation (10.16).

\[ P_{ow} = \rho r Q = AQ - BQ^2 \quad W \quad (10.17) \]

The three power-volume relationships then become:

**Radial bladed**

\[ B = 0 \quad \text{and} \quad P_{ow} = AQ \quad (linear) \]

**Backward bladed**

\[ B > 0 \quad \text{and} \quad P_{ow} = AQ - BQ^2 \quad (falling parabola) \]

**Forward bladed**

\[ B < 0 \quad \text{and} \quad P_{ow} = AQ + BQ^2 \quad (rising parabola). \]

The theoretical power-volume curves are shown on Figure 10.4 (b). Forward bladed fans are capable of delivering high flowrates at fairly low running speeds. However, their high power demand leads to reduced efficiencies. Conversely, the relatively low power requirement and high efficiencies of backward bladed impellers make these the preferred type for large centrifugal fans.

### 10.3.1.3. Actual characteristic curves for a centrifugal impeller

The theoretical treatment of the preceding subsections lead to linear pressure-volume relationships for radial, backward and forward bladed centrifugal impellers. In an actual fan, there are, inevitably, losses which result in the real pressure-volume curves lying below their theoretical counterparts. In all cases, friction and shock losses produce pressure-volume curves that tend toward zero pressure when the fan runs on open circuit, that is, with no external resistance.

Figure 10.5 shows a typical pressure-volume characteristic curve for a backward bladed centrifugal fan. Frictional losses occur due to the viscous drag of the fluid on the faces of the vanes. These are denoted as \( F_f \) and \( F_b \) on Figure 10.3. A diffuser effect occurs in the diverging area available for flow as the fluid moves through the impeller. This results in a further loss of available energy. In order to transmit mechanical work, the pressure on the front face of a vane, \( p_f \), is necessarily greater than that on the back, \( p_b \). A result of this is that the fluid velocity close to the trailing face is higher than that near the front face. These effects result in an asymmetric distribution of fluid velocity between two successive vanes at any given radius and produce an eddy loss. It may also be noted that at the outlet tip, the two pressures \( p_f \) and \( p_b \) must become equal. Hence, although the tip is most important in its influence on the outlet vector diagram, it does not actually contribute to the transfer of mechanical energy. The transmission of power is not uniform along the length of the blade.

The shock (or separation) losses occur particularly at inlet and reflect the sudden turn of near 90° as the fluid enters the eye of the impeller. In practice, wall effects impart a vortex to the fluid as it approaches the inlet. By a suitable choice of inlet blade angle, \( \beta_1 \), (Figure 10.3) the shock losses may be small at the optimum design flow. An inlet cone at the eye of the impeller or fixed inlet and outlet guide vanes can be fitted to reduce shock losses.

In the development of the theoretical pressure and power characteristics, we assumed radial inlet conditions. When the fluid has some degree of pre-rotation, the flow is no longer radial at the inlet to the impeller. The second term in Euler’s equation (10.11) takes a finite value and, again, results in a reduced fan pressure at any given speed of rotation.
Figure 10.4 Theoretical characteristic curves for a centrifugal fan
The combined effect of these losses on the three types of centrifugal impeller is to produce the characteristic curves shown on Figure 10.6. The non-overloading power characteristic, together with the steepness of the pressure curve at the higher flows, are major factors in preferring the backward impeller for large installations.

Figure 10.5 Effect of losses on the pressure-volume characteristic of a backward bladed centrifugal fan.

Figure 10.6 Actual pressure and shaft power characteristics for centrifugal impellers.
10.3.2. The axial impeller

Axial fans of acceptable performance did not appear until the 1930’s. This was because of a lack of understanding of the behaviour of airflow over axial fan blades. The early axial fans had simple flat angled blades and produced very poor efficiencies. The growth of the aircraft industry led to intensive studies of aerofoils for the wings of aeroplanes. In this section, we shall discuss briefly the characteristics of aerofoils. This facilitates our comprehension of the behaviour of axial fans. The theoretical treatment of axial impellers may be undertaken from the viewpoint of a series of aerofoils or by employing either vortex or momentum theory. We shall use the latter in order to remain consistent with our earlier analysis of centrifugal impeller although aerofoil theory is normally also applied in the detailed design of axial impellers.

10.3.2.1. Aerofoils

When a flat stationary plate is immersed in a moving fluid such that it lies parallel with the direction of flow then it will be subjected to a small drag force in the direction of flow. If it is then inclined upward at a small angle, $\alpha$, to the direction of flow then that drag force will increase. However, the deflection of streamlines will cause an increase in pressure on the underside and a decrease in pressure on the top surface. This pressure differential results in a lifting force. On an aircraft wing and in an axial fan impeller, it is required to achieve a high lift, $L$, without unduly increasing the drag, $D$. The dimensionless coefficients of lift, $C_L$, and drag, $C_D$, for any given section are defined in terms of velocity heads

\[
L = \rho \frac{C^2}{2} AC_L \quad \text{N} \quad (10.18)
\]

\[
D = \rho \frac{C^2}{2} AC_D \quad \text{N} \quad (10.19)
\]

where $C = \text{fluid velocity (m/s)}$ and $A = \text{a characteristic area usually taken as the underside of the plate or aerofoil (m}^2\text{)}$

The ratio of $C_L/C_D$ is considerably enhanced if the flat plate is replaced by an aerofoil. Figure 10.7 illustrates an aerofoil section. Selective evolution in the world of nature suggests that it is no
coincidence that the aerofoil shape is decidedly fish-like. The line joining the facing and trailing edges is known as the chord while the angle of attack, \( \alpha \), is defined as that angle between the chord and the direction of the approaching fluid.

A typical behaviour of the coefficients of lift and drag for an aerofoil with respect to angle of attack is illustrated on Figure 10.8. Note that the aerofoil produces lift and a high \( C_L/C_D \) ratio at zero angle of attack. The coefficient of lift increases in a near-linear manner. However, at an angle of attack usually between 12 and 18°, breakaway of the boundary layer occurs on the upper surface. This causes a sudden loss of lift and an increase in drag. In this stall condition, the formation and propagation of turbulent vortices causes the fan to vibrate excessively and to produce additional low frequency noise. A fan should never be allowed to run continuously in this condition as it can cause failure of the blades and excessive wear in bearings and other transmission components.

![Figure 10.8 Typical behaviour of lift and drag coefficients for an aerofoil.](image)

10.3.2.2. Theoretical pressure developed by an axial fan

Consider an imaginary cylinder coaxial with the drive shaft and cutting through the impeller blades at a constant radius. Depicting an impression of the cut blades on two dimensional paper produces a drawing similar to that of Figure 10.9 (a). For simplicity each blade is shown simply as a curved vane rather than an aerofoil section.
The air approaches the moving impeller axially at a velocity, \( C_1 \). At the optimum design point, the \( C_1 \) vector combines with the blade velocity vector, \( u \), to produce a velocity relative to the blade \( W_1 \), and which is tangential to the leading edge of the blade. At the trailing edge, the air leaves at a relative velocity \( W_2 \), which also combines with the blade velocity to produce the outlet absolute velocity, \( C_2 \). This has a rotational component, \( Cu_2 \), imparted by the rotation of the impeller. The initial axial velocity, \( C_1 \), has remained unchanged as the impeller has no component of axial velocity in an axial fan.
Figure 10.9 (b) shows how the inlet and outlet velocity diagrams can be combined. As $W_1$ and $W_2$ are both related to the same common blade velocity, $u$, the vector difference between the two, $W_1 - W_2$, must be equal to the final rotational component $C_{u2}$.

As the axial velocity, $C_1$, is the same at outlet as inlet, it follows that the increase in total pressure across the impeller is equal to the rise in static pressure.

$$p_{ft} = p_{2s} - p_{1s}$$

Now let us consider again the relative velocities, $W_1$ and $W_2$. Imagine for a moment that the impeller is standing still. It would impart no energy and Bernoulli’s equation (2.16) tells us that the increase in static pressure must equal the decrease in velocity pressure (in the absence of frictional losses). Hence

$$p_{ft} = p_{2s} - p_{1s} = \rho \left[ \frac{W_1^2}{2} - \frac{W_2^2}{2} \right] \text{ Pa}$$  \hspace{1cm} (10.20)

Applying Pythagoras' Theorem to Figure 10.9 (b) gives

$$W_1^2 = C_1^2 + u^2$$

and

$$W_2^2 = C_1^2 + (u - C_{u2})^2$$

giving

$$p_{ft} = \frac{\rho}{2} \left[ 2C_{u2}u - C_{u2}^2 \right]$$

i.e.

$$p_{ft} = \rho u C_{u2} - \frac{\rho C_{u2}^2}{2}$$  \hspace{1cm} (10.21)

Comparison with equation (10.12) shows that this is the pressure given by Euler’s equation, less the velocity head due to the final rotational velocity.

The vector diagram for an axial impeller with inlet guide vanes is given on Figure 10.10. In order to retain subscripts 1 and 2 for the inlet and outlet sides of the moving impeller, subscript 0 is employed for the air entering the inlet guide vanes. At the optimum design point the absolute velocities, $C_0$ and $C_2$ should be equal and axial, i.e. there should be no rotational components of velocity at either inlet or outlet of the guide vane/impeller combination. This means that any vortex action imparted by the guide vanes must be removed by an equal but opposite vortex action imparted by the impeller.

We can follow the process on the vector diagram on Figure 10.10 using the labelled points a, b, c, d, and e.

(a) The air arrives at the entrance to the guide vanes with an axial velocity, $C_0$, and no rotational component (point a).

(b) The turn on the inlet guide vanes gives a rotational component, $C_{u1}$, to the air. The axial component remains at $C_0$. Hence, the vector addition of the two results in the absolute velocity shown as $C_1$ (point b).
Figure 10.10  Velocity diagrams for an axial impeller with inlet guide vanes.
(c) In order to determine the velocity of the air relative to the moving impeller at location (1), we must subtract the velocity vector of the impeller, \( u \). This brings us to position \( c \) on the vector diagram and an air velocity of \( W_1 \) relative to the impeller.

(d) The turn on the impeller imparts a rotational component \( C_{u_2} \) to the air. The velocity of the air relative to the impeller is thus reduced to \( W_2 \) and we arrive at point \( d \).

(e) In order to determine the final absolute velocity of the air, we must add the impeller velocity, \( u \). This will take us to point \( e \) on the vector diagram (coinciding with point \( a \)), with no remaining rotational component provided that \( C_{u_1} = C_{u_2} \).

It follows that the absolute velocities at inlet and outlet, \( C_o \) and \( C_2 \), must be both axial and equal.

To determine the total theoretical pressure developed by the system, consider first the stationary inlet guide vanes (subscript \( g \)). From Bernoulli’s equation with no potential energy term

\[
p_g = \rho \left[ C_0^2 - C_1^2 \right] / 2
\]

Applying Pythagoras' theorem to the vector diagram of Figure 10.10 gives this to be

\[
p_g = -\rho \frac{C_{u_1}^2}{2}
\]  

(10.22)

and represents a pressure loss caused by rotational acceleration across the guide vanes.

The gain in total pressure across the impeller (subscript \( i \)) is given as

\[
p_i = \frac{\rho}{2} \left( W_1^2 - W_2^2 \right)
\]  

(10.23)  

(see, also, equation (10.20))

Using Pythagorus on Figure 10.10 again, gives this to be

\[
p_i = \frac{\rho}{2} \left( C_0^2 + (u + C_{u_2})^2 - (C_0^2 + u^2) \right)
\]

\[
= \frac{\rho}{2} \left( C_{u_2}^2 + 2 C_{u_2} u \right)
\]

But, as \( C_{u_1} = C_{u_2} \), this can also be written as

\[
p_i = \frac{\rho}{2} \left( C_{u_1}^2 + 2 C_{u_2} u \right) \quad \text{Pa}
\]  

(10.24)

Now the total theoretical pressure developed by the system, \( p_t \) must be the combination \( p_g + p_i \).

Equations (10.22) and (10.24) give

\[
p_t = p_g + p_i = \rho C_{u_2} u \quad \text{Pa}
\]  

(10.25)
Once again, we have found that the theoretical pressure developed by a fan is given by Euler's equation (10.12). Furthermore, comparison with equation (10.21) shows that elimination of the residual rotational component of velocity at outlet (by balancing $C_{u1}$ and $C_{u2}$) results in an increased fan pressure when guide vanes are employed.

The reader might wish to repeat the analysis for outlet guide vanes and for a combination of inlet and outlet guide vanes. In these cases, Euler's equation is also found to remain true. Hence, wherever the guide vanes are located, the total pressure developed by an axial fan operating at its design point depends only upon the rotational component imparted by the impeller, $C_{u2}$, and the peripheral velocity of the impeller, $u$.

It is obvious that the value of $u$ will increase along the length of the blade from root to tip. In order to maintain a uniform pressure rise and to inhibit undesirable cross flows, the value of $C_{u2}$ in equation (10.25) must balance the variation in $u$. This is the reason for the "twist" that can be observed along the blades of a well-designed axial impeller.

10.3.2.3. Actual characteristic curves for an axial fan

The losses in an axial fan may be divided into recoverable and non-recoverable groups. The recoverable losses include the vortices or rotational components of velocity that exist in the airflow leaving the fan. We have seen that these losses can be recovered when operating at the design point by the use of guide vanes. However, as we depart from the design point, swirling of the outlet air will build up.

The non-recoverable losses include friction at the bearings and drag on the fan casing, the hub of the impeller, supporting beams and the fan blades themselves. These losses result in a transfer from mechanical energy to heat which is irretrievably lost in its capacity for doing useful work.

Figure 10.11 is an example of the actual characteristic curves for an axial fan. The design point, $C$, coincides with the maximum efficiency. At this point the losses are at a minimum. In practice, the region A to B on the pressure curve would be acceptable. Operating at low resistance, i.e. to the right of point B, would not draw excessive power from the motor as the shaft power curve shows a non-overloading characteristic. However, the efficiency decreases rapidly in this region.

The disadvantage of operating at too high a resistance, i.e. to the left of point A is, again, a decreasing efficiency but, more importantly, the danger of approaching the stall point, D. There is a definite discontinuity in the pressure curve at the stall point although this is often displayed as a smoothed curve. Indeed, manufacturers' catalogues usually show characteristic curves to the right of point A only. In the region E to D, the flow is severely restricted. Boundary layer breakaway takes place on the blades (Section 10.3.2.1) and centrifugal action occurs producing recirculation around the blades.

A fixed bladed axial fan of constant speed has a rather limited useful range and will maintain good efficiency only when the system resistance remains sensibly constant. This can seldom be guaranteed over the full life of a main mine fan. Fortunately, there are a number of ways in which the range of an axial fan can be extended:

(a) The angle of the blades may be varied. Many modern axial fans allow blade angles to be changed, either when the rotor is stationary or while in motion. The latter is useful if the fan is to be incorporated into an automatic ventilation control system. Figure 10.12 gives an example of the characteristic curves for an axial fan of variable blade angle. The versatility of such fans gives them considerable advantage over centrifugal fans.
(b) The **angle of the inlet and/or outlet guide vanes** may also be varied, with or without modification to the impeller blade angle. The effect is similar to that illustrated by Figure 10.12.

(c) The **pitch of the impeller** may be changed by adding or removing blades. The impeller must, of course, remain dynamically balanced. This technique can result in substantial savings in power during time periods of relatively light load.

(d) The **speed of the impeller** may be changed either by employing a variable speed motor or by changing the gearing between the motor and the fan shaft. The majority of fans are driven by A.C. induction motors at a fixed speed. Variable speed motors are more expensive although they may produce substantial savings in operating costs. Axial fans may be connected to the motor via flexible couplings which allow a limited degree of angular or linear misalignment. Speed control may be achieved by hydraulic couplings, or, in the case of smaller fans, by V-belt drives with a range of pulley sizes.

---

**Figure 10.11** Typical characteristic curves for an axial fan.
10.4 FAN LAWS

The performance of fans is normally specified as a series of pressure, efficiency and shaft power characteristic curves plotted against airflow for specified values of rotational speed, air density and fan dimensions. It is, however, convenient to be able to determine the operating characteristic of the fan at other speeds and air densities. It is also useful to be able to use test results gained from smaller prototypes to predict the performance of larger fans that are geometrically similar.

Euler's equation and other relationships introduced in Section 10.3 can be employed to establish a useful set of proportionalities known as the fan laws.

10.4.1. Derivation of the fan laws

Fan pressure
From Euler's equation (10.12 and 10.25)

$$ p_t = \rho u C_{u2} $$
However, both the peripheral speed, $u$, and the rotational component of outlet velocity, $C_{u2}$ (Figures 10.3 and 10.10) vary with the rotational speed, $n$, and the impeller diameter, $d$. Hence,

$$p_{ht} \propto \rho (nd)(nd)$$

or

$$p_{ht} \propto \rho n^2 d^2$$  \hspace{1cm} (10.26)

where $\alpha$ means 'proportional to'.

**Airflow**

For a centrifugal fan, the radial flow at the impeller outlet is given as

$$Q = \text{area of flow at impeller outlet} \times C_{m2} = \pi d \times \text{impeller width} \times C_{m2}$$

where $d = \text{diameter of impeller}$

and $C_{m2} = \text{radial velocity at outlet}$ (Figure 10.3)

However, for geometric similarity between any two fans

impeller width $\alpha d$

giving

$$Q \propto d^2 C_{m2}$$

Again, for geometric similarity, all vectors are proportional to each other. Hence

$$C_{m2} \propto u \propto nd$$

giving

$$Q \propto n d^3$$  \hspace{1cm} (10.27)

A similar argument applies to the axial impeller (Figure 10.10). At outlet,

$$Q = \frac{\pi d^2}{4} \times C_2$$

where $C_2 = \text{axial velocity}$

As before, vectors are proportional to each other for geometric similarity,

$$C_2 \propto u \propto nd$$

giving, once again,

$$Q \propto n d^3$$

**Density**

From Euler's equation (10.12 or 10.25) it is clear that fan pressure varies directly with air density

$$p_{ht} \propto \rho$$  \hspace{1cm} (10.28)

However, we normally accept volume flow, $Q$, rather than mass flow as the basis of flow measurement in fans. In other words, if the density changes we still compare operating points at corresponding values of volume flow.
Air Power

For incompressible flow, the mechanical power transmitted from the impeller to the air is given as

\[ P_{\text{ow}} = p_f t Q \]  

(see equation (5.56))

Employing proportionalities (10.26) and (10.27) gives

\[ P_{\text{ow}} \alpha \rho n^3 d^5 \]  

(10.29)

10.4.2. Summary of fan laws

In the practical utilization or design of fans, we are normally interested in varying only one of the independent variables (speed, air density, impeller diameter) at any given time while keeping the other two constant. The fan laws may then be summarized as follows:

<table>
<thead>
<tr>
<th>Variable speed ((n))</th>
<th>Variable diameter ((d))</th>
<th>Variable air density ((\rho))</th>
</tr>
</thead>
<tbody>
<tr>
<td>( p \alpha n^2 )</td>
<td>( p \alpha d^2 )</td>
<td>( p \alpha \rho )</td>
</tr>
<tr>
<td>( Q \alpha n )</td>
<td>( Q \alpha d^2 )</td>
<td>( Q ) fixed</td>
</tr>
<tr>
<td>( P_{\text{ow}} \alpha n^3 )</td>
<td>( P_{\text{ow}} \alpha d^5 )</td>
<td>( P_{\text{ow}} \alpha \rho )</td>
</tr>
</tbody>
</table>

These laws can be applied to compare the performance of a given fan at changed speeds or air densities, or to compare the performance of different sized fans provided that those two fans are geometrically similar.

If the two sets of operating conditions, or the two geometrically similar fans are identified by subscripts \(a\) and \(b\), then the fan laws may be written, more generally, as the following equations

\[ \frac{p_{\text{ft},a}}{p_{\text{ft},b}} = \frac{n_a^2 d_a^2 \rho_a}{n_b^2 d_b^2 \rho_b} \]  

(10.30)

\[ \frac{Q_a}{Q_b} = \frac{n_a d_a^3}{n_b d_b^3} \]  

(10.31)

\[ \frac{P_{\text{ow},a}}{P_{\text{ow},b}} = \frac{n_a^3 d_a^5 \rho_a}{n_b^3 d_b^5 \rho_b} \]  

(10.32)

where \( p_{\text{ft}} \) = fan pressure (applies for both total and static pressure)  
\( Q \) = airflow  
\( P_{\text{ow}} \) = airpower  
\( n \) = rotational speed  
\( d \) = impeller diameter  
\( \rho \) = air density

and
Example.
Characteristic curves are available for a fan running at 850 rpm and passing air of inlet density 1.2 kg/m³. Readings from the curves indicate that at an airflow of 150 m³/s, the fan pressure is 2.2 kPa and the shaft power is 440 kW. Assuming that the efficiency remains unchanged, calculate the corresponding points if the fan is run at 1100 rpm in air of density 1.1 kg/m³.

Solution.
As it is the same fan that is to be used in differing conditions, there is no change in impeller diameter or overall geometry.

Equation (10.30) gives the new pressure to be

\[
p_f = 2.2 \left(\frac{1100}{850}\right)^2 \frac{1.1}{1.2} = 3.377 \text{ kPa}
\]

From equation (10.31), the new volume flow is

\[
Q = 150 \times \frac{1100}{850} = 194.1 \text{ m}^3/\text{s}
\]

Equation (10.32) refers to the airpower rather than the shaft delivered to the impeller. However, if we assume that the impeller efficiency remains the same at corresponding points on the characteristic curves then we may also use that equation for shaft power.

\[
P_{ow} = 440 \times \left(\frac{1100}{850}\right)^3 \times \frac{1.1}{1.2} = 874 \text{ kW}
\]

By treating a series of points from the original curves in this way, a second set of characteristic curves can be produced that is applicable to the new conditions.

10.5. FANS IN COMBINATION

Many mines or other subsurface facilities have main and, perhaps, booster fans sited at differing locations; for example, there may be two or more upcast shafts, each with its own surface exhausting fan. With the advent of simulation programs for ventilation network analysis (Chapters 7 and 9), the relevant pressure-volume characteristic data may be entered separately for each fan unit. The system resistance offered to each of those fans becomes a function not only of the network geometry but also the location and operating characteristics of other fans in the system. That resistance is sometimes termed the effective resistance 'seen' by the fan.

There are situations in which it is advantageous to combine fans either in series or in parallel at a single location. Such combinations enable a wide spectrum of pressure-volume duties to be attained with only a limited range of fan sizes. In general, fans may be connected in series in order to pass a given airflow against an increased resistance, while a parallel combination allows the flow to be increased for any given resistance. Although ventilation network programs can allow each fan to be entered separately, it is sometimes more convenient to produce a pressure-volume characteristic curve that represents the combined unit.
10.5.1. Fans in series

Figure 10.13 shows two fans, \textit{a} and \textit{b}, located in series within a single duct or airway. The corresponding pressure-volume characteristics and the effective resistance curve are also shown. The characteristic curve for the combination is obtained simply by adding the individual fan pressures for each value of airflow. The effective operating point is located at C, where the resistance curve intersects the combined characteristic. Fans \textit{a} and \textit{b} both pass the same airflow, \textit{Q}, but develop pressures \textit{p}_a and \textit{p}_b respectively. The individual operating points are shown as A and B. For three or more fans, the process of adding fan pressures remains the same. However, if the change in density through the
combination becomes significant then the fan laws (Section 10.4.2) should be employed to correct the individual characteristic curves.

As shown in Figure 10.13, the individual fans need not have identical characteristic curves. However, if one fan is considerably more powerful than the other, or if the system resistance falls to a low level, then the impeller of the weaker unit may be driven in turbine fashion by its stronger companion. The weaker fan then becomes an additional resistance on the system. It is usual to employ identical fans in combination.

10.5.2. Fans in parallel

For fans that are combined in parallel, the airflows are added for any given fan pressure in order to obtain the combined characteristic curve. As shown in Figure 10.14, fans a and b pass airflows $Q_a$ and $Q_b$, respectively, but at the same common pressure, $p$.

![Figure 10.14 Characteristic curves for two fans connected in parallel.](image)

The operating point for the complete unit occurs at C with the individual operating points for fans a and b at A and B respectively. Here again, the fans need not necessarily be identical. However, care must be taken to ensure that the operating points A and B do not move too far up their respective curves. This is particularly important in the case of axial fans because of their pronounced stall characteristic. In practice, before this condition is reached, the fans may exhibit
a noticeable “hunting” effect. For these reasons, the maximum variation in system resistance that is likely to occur should be investigated before installing fans in parallel.

Three or more fans may be combined in parallel, adding airflows to obtain the combined characteristic curve. Although Figure 10.14 shows two fans with differing characteristics, it is prudent to employ identical fans when connected in parallel. This will reduce the tendency for one of them to approach stall conditions before the other. However, variations in the immediate surroundings of ductwork or airway geometry often results in the fans operating against slightly different effective resistances. Hence, even when identical fans are employed, it is usual for measurements to indicate that they are producing slightly different pressure-volume duties. In the case of fans located in separate ducts or airways that are connected in parallel, the resistance of those ducts or airways may be taken into account by subtracting the frictional pressure losses in each branch from the corresponding fan pressures. In these circumstances, a better approach is to consider the fans as separate units for the purposes of network analysis.

An advantage of employing fans in parallel is that if one of them fails then the remaining fan(s) continue to supply a significant proportion of the original flow. In the example shown on Figure 10.14, if fan a ceases to operate then the operating point for fan b will fall to position E, giving some 70 per cent of the original airflow. The latter value depends upon the number of fans employed, the shape of their pressure-volume characteristic curves and the provision of non-return baffles at the fan outlets.

Fans may be connected in any series/parallel configuration, adding pressures and airflows respectively to obtain the combined characteristic curve. This is particularly useful for booster fan locations. A mine may maintain an inventory of standard fans, combining them in series/parallel combinations to achieve any desired operating characteristic.

10.6. FAN PERFORMANCE

Power is delivered to the drive shaft of a fan impeller from a motor (usually electric) and via a transmission assembly. Losses occur in both the motor and transmission. For a properly maintained electrical motor and transmission, some 95 per cent of the input electrical power may be expected to appear as mechanical energy in the impeller drive shaft. The impeller, in turn, converts most of that energy into useful airpower to produce both movement of the air and an increase in pressure. The remainder is consumed by irreversible losses across the impeller and in the fan casing (Sections 10.3.1.3 and 10.3.2.3) producing an additional increase in the temperature of the air.

Impeller efficiency may be defined as

\[
\text{Impeller efficiency} = \frac{\text{Airpower}}{\text{Shaft power}}
\]  

while the overall efficiency of the complete motor/transmission/impeller unit is given as

\[
\text{Overall efficiency} = \frac{\text{Airpower}}{\text{Motor input power}}
\]

In the following subsection, we shall define other measures of fan efficiency. As there are several different forms of fan efficiency it is prudent, when perusing manufacturers’ literature, to ascertain the basis of any quoted values of efficiencies. It is also important to use the same measure of efficiency when comparing one fan with another. However, the one parameter that really matters is the input power required to achieve the specified pressure-volume duty, as this is the factor that dictates the operating cost of the fan.
10.6.1. Compressibility, fan efficiency, and fan testing

In our previous analyses, we have defined airpower as the product $pQ$ on the basis of incompressible flow. That simplification will give an error of less than one percent when testing fans that develop pressures up to 2.8 kPa. Unfortunately, fan operating costs at many large mines are such that one per cent may represent a significant expenditure. Furthermore, mine fan pressures exceeding 6 kPa are not uncommon. For these reasons, we should take air compressibility into account when measuring fan performance.

Applying the steady-flow energy equation (3.25) to a fan gives

$$\frac{u_1^2 - u_2^2}{2} + (Z_1 - Z_2)g + W = \int_{1}^{2} VdP + F_{12} = (H_2 - H_1) - q_{12} \frac{J}{kg} \quad (10.35)$$

where subscripts 1 and 2 refer to the fan inlet and outlet respectively

- $W = \text{Impeller shaft work (J/kg of air)}$
- $V = \text{Specific volume of air (m}^3/\text{kg)}$
- $P = \text{Absolute (barometric) pressure (Pa)}$
- $F_{12} = \text{Frictional losses (J/kg)}$
- $H = \text{Enthalpy (J/kg)}$
- $q_{12} = \text{Heat added through fan casing (J/kg)}$

For the purposes of this analysis we shall assume that the change in elevation through the fan is negligible, $Z_1 - Z_2 = 0$. Furthermore, we shall assume that the change in air velocity across the fan is also negligible compared to other terms, $u_1 = u_2$. This latter assumption implies that the fan total pressure, referred to here simply as $p_f$, is equal to the increase in barometric pressure across the fan, $P_2 - P_1$. Then

$$W = \int_{1}^{2} VdP + F_{12} = (H_2 - H_1) - q_{12} \frac{J}{kg} \quad (10.36)$$

It is also reasonable to assume that the heat transferred from the surroundings through the fan casing is small compared to the shaft work. The steady-flow energy equation then simplifies to the adiabatic equation

$$W = \int_{1}^{2} VdP + F_{12} = (H_2 - H_1) \frac{J}{kg} \quad (10.37)$$

In order to define a thermodynamic efficiency for the fan impeller, we must first designate a "perfect" fan against which we can compare the performance of the real fan. In the perfect fan, there are no losses, $F_{12} = 0$. As we now have a frictionless adiabatic, i.e. an isentropic compression we can write:

$$W_{isen} = \int_{1}^{2} VdP = (H_{2,isen} - H_1) \frac{J}{kg} \quad (10.38)$$

where the subscript $isen$ denotes isentropic conditions.
10.6.1.1. The pressure-volume method

There are two methods of determining the efficiency of a fan. The technique that is accepted as standard by most authorities relies upon measurement of the fan pressure, airflow and shaft power, and is known as the pressure-volume method.

The first step is to evaluate the integral \( \int_{1}^{2} VdP \). Here, we have a choice. We can use the isentropic relationship

\[
P V^{\gamma} = \text{Constant} \quad C
\]  

(10.39)

where \( \gamma \) = the isentropic index (i.e. the ratio of specific heats \( C_{P}/C_{V} = 1.4 \) for dry air). This will lead to the isentropic efficiency of the fan.

Alternatively, we could employ the actual polytrope produced by the fan

\[
P V^{n} = \text{Constant}
\]

and assume that it is a reversible (ideal) process. This will lead to the polytropic efficiency of the fan. Both measures of efficiency are acceptable provided that the choice is stated clearly. While the polytropic efficiency is closer to a true measure of output/input, and takes any heat transfer, \( q_{12} \), into account, the isentropic efficiency has the advantage that the index \( \gamma \) is defined for any given gas. For this reason, we shall continue the analysis on the basis of isentropic compression within the ideal fan. In practice, polytropic and isentropic efficiencies are near equal for most fan installations.*

Substituting \( V = (C/P)^{\gamma/\gamma} \) into equation (10.38) gives

\[
W_{isen} = \int_{1}^{2} \frac{C^{\gamma/\gamma}}{P^{\gamma/\gamma}} dP = C^{\gamma/\gamma} \left[ \frac{P_{2}^{\gamma} - P_{1}^{\gamma}}{\gamma - 1} \right]^{1/\gamma - 1}
\]

but as \( C^{\gamma/\gamma} = P^{\gamma/\gamma} V \)

\[
W_{isen} = \gamma \left[ \frac{P_{2} V_{2} - P_{1} V_{1}}{\gamma - 1} \right] = \frac{\gamma}{\gamma - 1} P_{1} V_{1} \left[ \frac{P_{2}}{P_{1}} \frac{V_{2}}{V_{1}} - 1 \right]
\]

Now \( \frac{V_{2}}{V_{1}} = \left[ \frac{P_{1}}{P_{2}} \right]^{\gamma} \) from equation (10.39)

* Fan efficiencies can be defined in terms of areas on the Ts diagram of Figure 3.7:

\[
\eta_{isen} = \frac{\text{Area ACBXY}}{\text{AreaDBXZ}} \quad \eta_{poly} = \frac{\text{Area ADBXY}}{\text{AreaDBXZ}}
\]
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\[ W_{isen} = \frac{\gamma}{\gamma - 1} P_1 V_1 \left( \frac{P_2}{P_1} \right)^{1 - \frac{1}{\gamma}} - 1 \] \quad \text{J/kg}

In order to convert the shaft work from J/kg to shaft power, \( P_{ow} \) (J/s or Watts), we must multiply by the mass flow of air

\[ M = \frac{Q_1 \rho_1}{V_1} = \frac{Q_1}{V_1} \quad \text{kg/s} \]

where \( \rho = \text{air density (kg/m}^3 \text{)} \)

Then

\[ P_{ow,isen} = \frac{\gamma}{\gamma - 1} P_1 Q_1 \left( \frac{P_2}{P_1} \right)^{1 - \frac{1}{\gamma}} - 1 \] \quad \text{J/kg W} \quad (10.40)

Now let us multiply both the numerator and denominator by fan pressure, \( p_f \).

\[ P_{ow,isen} = p_f Q_1 \frac{\gamma}{\gamma - 1} \frac{P_1}{p_f} \left( \frac{P_2}{P_1} \right)^{1 - \frac{1}{\gamma}} - 1 \] \quad \text{W}

We can rewrite this equation as

\[ P_{ow,isen} = p_f Q_1 K_p \quad (10.41) \]

where

\[ K_p = \frac{\gamma}{\gamma - 1} \frac{P_1}{p_f} \left( \frac{P_2}{P_1} \right)^{1 - \frac{1}{\gamma}} - 1 \] \quad (10.42)

and is known as the **Compressibility Coefficient**. By substituting \( p_f = (P_2 - P_1) \) we obtain

\[ K_p = \frac{\gamma}{\gamma - 1} \left( \frac{P_2 / P_1}{(P_2 / P_1)^{1/\gamma} - 1} \right) \] \quad (10.43)

We can now see that our earlier assumption of incompressible flow ( \( P_{ow} = p_f Q \) ) involved an error represented by the deviation of \( K_p \) from unity. Figure 10.15 shows the variation in compressibility coefficient with respect to the pressure ratio \( P_2 / P_1 \). For unsaturated air the value \( \gamma = 1.4 \) may be used.
The isentropic efficiency of the fan can now be defined as

\[
\eta_{\text{isen}} = \frac{P_{\text{ow,isen}}}{\text{Shaft power}} = \frac{p_f Q_1 K_p}{\text{Shaft power}}
\]  

(10.44)

Employment of this equation is a standard technique of determining fan efficiency and is known commonly as the pressure-volume method.

Figure 10.15  Variation of Compressibility Coefficient with respect to pressure ratio.  
(γ = 1.4 for dry air.)

Two other terms used frequently in the literature are static efficiency and total efficiency. These are obtained simply by using fan static pressure and fan total pressure, respectively, for \( p_f \) in equation (10.44).

All of these measures of efficiency are matters of definition rather than precision. However, care should be taken to ensure that the same measure is employed when comparing fan performances.
Equation (10.44) indicates that the pressure-volume method of fan testing requires the measurement of pressures, air volume flow and the impeller shaft power. It is frequently the case that a large fan installed at a mine site does not meet the specifications indicated by factory tests. Part of the problem may be the less than ideal inlet or outlet conditions that often exist in field installations. In particular, uneven distribution of airflow approaching the fan can produce a diminished performance and may even result in premature blade failure.

Another problem is that turbulence and an asymmetric velocity profile may make it difficult to obtain good accuracy in the measurement of airflow (Section 6.2). Water droplets in the airstream can also result in erroneous readings from both anemometers and pitot tubes.

Impeller shaft power can be obtained accurately in the laboratory or factory test rig by means of torque meters or, in the case of smaller fans, swinging carcass (dynamometer) motors. However, for in-situ tests, it may be necessary to resort to the measurement of input electrical power and to rely upon manufacturer’s data for the efficiencies of the motor and transmission.

Despite these difficulties, it is always advisable to conduct an in-situ test on a new main fan in order to verify, or modify, the fan characteristic data that are to be used for subsequent ventilation network exercises.

Example.
A fan passes an airflow of 300 m³/s at the inlet, and develops a pressure of 2.5 kPa. The barometric pressure at the fan inlet is 97 kPa. The motor consumes an electrical power of 1100 kW. Assuming a combined motor/transmission efficiency of 95 per cent, determine the isentropic efficiency of the impeller and, also, of the total unit.

Solution.
\[ P_1 = 97 \text{ kPa}, \quad P_2 = 97 + 2.5 = 99.5 \text{ kPa} \]

Using a value of \( \gamma = 1.4 \) for air, equation (10.40) gives the air power (or isentropic shaft power) as

\[
P_{ow,isen} = \frac{\gamma}{\gamma - 1} P_1 Q_1 \left( \frac{P_2}{P_1} \right)^{\frac{1}{\gamma - 1}} - 1 \] \( \frac{J}{kg} \)

\[ = 3.5 \times 97000 \times 300 \left( \frac{99.5}{97} \right)^{0.286} - 1 \]

\[ = 743.9 \times 10^3 \text{ W or } 743.9 \text{ kW} \]

Actual shaft power,
\[ P_{ow} = 1100 \times 0.95 = 1045 \text{ kW} \]

Isentropic efficiency of impeller
\[
\eta_{isen} = \frac{P_{ow,isen}}{P_{ow}} = \frac{743.9}{1045} = 0.712 \text{ or } 71.2 \text{ per cent} \]
The overall isentropic efficiency of the unit is

\[ \eta_{isen}^{(overall)} = \frac{743.9}{1100} = 0.676 \text{ or } 67.3 \text{ per cent} \]

Note that if compressibility had been ignored, the isentropic shaft power would have been

\[ P_{ow,isen} = p_f Q = 2.5 \times 300 = 750 \text{ kW} \]

involving an error of 0.8 per cent from the true value of 743.9 kW.

10.6.1.2. The thermometric method

**Unsaturated Conditions**

The problems associated with in-situ pressure-volume tests on mine fans have led researchers to seek another method that did not require the measurement of either airflow or shaft power. The earliest such work appears to have been carried out on the 1920's (Whitaker) although later work was required to make it a practical proposition (McPherson 1971, Drummond 1972).

The thermometric method of fan testing is based on the enthalpy terms in the steady flow energy equation. The shaft work for the ideal isentropic fan is given by equation (10.38)

\[ W_{isen} = (H_{2,isen} - H_1) \]

If we assume that the air contains no free water droplets and that neither evaporation nor condensation occurs within the fan then equation (3.33) gives

\[ W_{isen} = (H_{2,isen} - H_1) = C_p (T_{2,isen} - T_1) \text{ J/kg} \]

where \( C_p \) = Specific heat at constant pressure of the air (1005 J/kgK for dry air).

Similarly, for the real fan (no secondary subscript)

\[ W = C_p (T_2 - T_1) \text{ J/kg} \]

The isentropic efficiency is then given as the ratio of the shaft work for the isentropic fan to that for the actual fan

\[ \eta_{isen} = \frac{W_{isen}}{W} = \frac{(T_{2,isen} - T_1)}{(T_2 - T_1)} \]

or

\[ \eta_{isen} = \frac{\Delta T_{isen}}{\Delta T} \]

where \( \Delta T_{isen} = (T_{2,isen} - T_1) \) °C and \( \Delta T = (T_2 - T_1) \) °C

\( \Delta T_{isen} \) is, therefore, the increase in dry bulb temperature that would occur as air passes through an isentropic fan, while \( \Delta T \) is the temperature rise that actually occurs in the real fan. These two parameters are illustrated on the Ts diagram of Figure 3.7.
As $\Delta T$ can be measured, it remains to find an expression for the isentropic temperature rise. This was derived in Chapter 3 as equation (3.53)

$$\frac{T_{2,\text{isen}}}{T_1} = \left[ \frac{P_2}{P_1} \right]^{(\gamma-1)/\gamma}$$

(10.49)

where $\gamma = \text{Ratio of specific heats, } C_p/C_v$ (1.4 for dry air).

Then

$$\Delta T_{\text{isen}} = (T_{2,\text{isen}} - T_1) = T_1 \left[ \frac{P_2}{P_1} \right]^{(\gamma-1)/\gamma} - 1 \, ^\circ\text{C}$$

(10.50)

giving

$$\eta_{\text{isen}} = \frac{\Delta T_{\text{isen}}}{\Delta T} = \frac{T_1}{\Delta T} \left[ \frac{P_2}{P_1} \right]^{(\gamma-1)/\gamma} - 1$$

(10.51)

This equation may be employed directly as all of its variables are measurable. It can, however, be simplified by employing the compressibility coefficient, $K_p$, from equation (10.42)

$$\left[ \frac{P_2}{P_1} \right]^{(\gamma-1)/\gamma} - 1 = K_p \frac{P_f}{P_i} \frac{(\gamma-1)}{\gamma}$$

Substituting in equation (10.51) gives

$$\eta_{\text{isen}} = \frac{(\gamma-1)}{\gamma} \frac{P_f}{P_i} \frac{T_i}{\Delta T} K_p$$

(10.52)

Using the value of $\gamma = 1.4$ for dry air gives

$$\eta_{\text{isen}} = 0.286 \frac{P_f}{P_i} \frac{T_i}{\Delta T} K_p$$

(10.53)

Furthermore, for fan pressures not exceeding 2.8 kPa, the compressibility coefficient may be ignored for one per cent accuracy, leaving the simple equation:

$$\eta_{\text{isen}} = 0.286 \frac{P_f}{P_i} \frac{T_i}{\Delta T}$$

(10.54)

This analysis has assumed that the airflow contains no liquid droplets of water. The presence of water vapour has very little effect on the accuracy of equations (10.53 or 10.54) provided that the air remains unsaturated. (See equations A10.15 and A10.16 in Appendix A10.2 following this chapter.)
Saturated Conditions

In many deep and hot mines, the reduction in temperature as the return air ascends an upcast shaft may cause condensation. Exhaust fans operating at the surface will then pass fogged air—a mixture of air, water vapour and droplets of liquid water. Equation (10.53) no longer holds due to the cooling effect of evaporation within the fan. A more complex analysis, taking into account the air, water vapour, liquid droplets, and the phase change of evaporation produces the following differential equation which quantifies the temperature-pressure relationship for the isentropic behaviour of fogged air:

\[
\frac{dT}{dP} = \left(\frac{R + R_v X_s}{P} + \frac{L X_s}{(P-e_s)}\right) - \frac{C_p + X C_w - B X_s + \frac{L^2 P X_s}{(P-e_s) R_v T^2}}{C_p + X C_w - B X_s + \frac{L^2 P X_s}{(P-e_s) R_v T^2}} \quad ^\circ C/Pa \quad (10.55)
\]

A derivation of equation (10.55) and the full definition of the symbols are given in Appendix A10.2 at the end of this chapter.

Inserting the values of the constants gives

\[
\frac{dT}{dP} = \frac{0.286 \left(1 + 1.6078 X_s\right) T}{P} + \frac{L X_s}{287.04 (P-e_s)} - \frac{1 + 4.1662 X - 2.3741 X_s + \frac{L^2 P X_s}{463.81 \times 10^3 (P-e_s) T^2}}{1 + 4.1662 X - 2.3741 X_s + \frac{L^2 P X_s}{463.81 \times 10^3 (P-e_s) T^2}} \quad ^\circ C/Pa \quad (10.56)
\]

where
- \(T\) = Absolute temperature (K)
- \(P\) = Absolute pressure (Pa)
- \(L\) = Latent heat of evaporation (J/kg)
- \(X\) = Total moisture content (kg/kg dry air)
- \(X_s\) = Water vapour content at saturation (kg/kg dry air) and
- \(e_s\) = Saturation vapour pressure at temperature \(T\) (Pa).

This equation can be programmed into a calculator or microprocessor for the rapid evaluation of \(dT/dP\).

For the relatively small pressures and temperatures developed by a fan, we can write to a good approximation

\[
\Delta T_{\text{sen}} = \rho f \frac{dT}{dP} \quad (10.57)
\]

To improve accuracy, the values used for \(T\) and \(P\) should be the mean temperature and pressure of the air as it passes through the fan.

Although the thermometric method eliminates the need for airflow and shaft power in the determination of fan efficiency, it does introduce other practical difficulties. The temperature of the air may vary with both time and position over the measurement cross-sections due to vortex action and thermal stratification. The method of measuring the temperature rise across the fan must give an instantaneous reading of the difference between the mean temperatures at inlet and outlet measuring stations. This may be accomplished by thermocouples, connected in series, with the hot and cold junctions distributed over supporting grids at the outlet and inlet measuring stations respectively (Drummond).
Poorly designed evasees on exhausting centrifugal fans may suffer from a re-entry down-draught of external air on one side. A test should be made for this condition before positioning the exit thermocouple heads.

Although the thermometric technique does not require an airflow in order to calculate efficiency, the airflow is nevertheless still needed if that efficiency is to be compared with a manufacturer’s characteristic curve. If the site conditions are such that greater confidence can be placed in a determination of shaft power than airflow, then the latter may be computed as

\[
Q = \frac{\text{Shaft power}}{\rho_1 (H_2 - H_1)} \quad \text{m}^3 \text{s}^{-1}
\]

(10.58)

where the shaft power is in Watts,
\(\rho_1 = \text{air density at inlet} \quad (\text{kg/m}^3)\)
and
\(H = \text{enthalpy} \quad (\text{J/kg})\)

If the air is unsaturated then

\[(H_2 - H_1) = C_p (T_2 - T_1)\]

In the case of fogged air, the enthalpies must be determined from equation (A10.3) in Appendix A10.2 in order to account for water vapour and liquid droplets.

Example 1.
A temperature rise of 5.96 °C is measured across a fan developing an increase in barometric pressure of 5 kPa and passing unsaturated air. The inlet temperature and pressure are 25.20 °C and 101.2 kPa respectively. Determine the isentropic temperature rise and, hence, the isentropic efficiency of the impeller.

Solution.
From equation (10.50)

\[
\Delta T_{\text{inen}} = (273.15 + 25.2) \left\{\frac{101.2 + 5}{101.2} \right\}^{0.286} - 1 = 4.143 \quad \text{°C}
\]

Then \(\eta_{\text{inen}} = \frac{\Delta T_{\text{inen}}}{\Delta T} = \frac{4.143}{5.96} = 0.695 \quad \text{or} \quad 69.5 \quad \text{per cent}\)

Ignoring the effects of compressibility allows equation (10.54) to be applied, giving

\[
\eta_{\text{inen}} = 0.286 \times \frac{5}{101.2} \times \frac{(273.15 + 25.2)}{5.96} = 0.707 \quad \text{or} \quad 70.7 \quad \text{per cent}
\]

Hence, in this example, ignoring compressibility causes the percentage fan efficiency to be overestimated by 1.2.
Example 2.
A surface exhausting fan passes fogged air of total moisture content $X = 0.025$ kg/kg dry air. The pressure and temperature at the fan inlet are 98.606 kPa and 18.80 °C respectively. If the temperature rise across the fan is 2.84 °C when the increase in pressure is 5.8 kPa, calculate the isentropic efficiency of the impeller.

Solution.
The mean barometric pressure in the fan is

$$ P = 98.606 + \frac{5.8}{2} = 101.506 \text{ kPa} $$

At inlet, $T_1 = 273.15 + 18.8 = 291.95 \text{ K}$

and at outlet, $T_2 = 291.95 + 2.84 = 294.79 \text{ K}$

Hence, the mean temperature is

$$ T = \frac{293.37 \text{ K}}{\text{or}} \ t = 20.22 \text{ °C} $$

The psychrometric equations in Section 14.6 allow the following parameters to be calculated. For saturation conditions, the wet and dry bulb temperatures are, of course, equal.

Saturation vapour pressure:

$$ e_s = 610.6 \exp \left[ \frac{17.27 t}{237.3 + t} \right] $$

$$ e_s = 610.6 \exp \left[ \frac{17.27 \times 20.22}{237.3 + 20.22} \right] = 2369.5 \text{ Pa} $$

Saturation vapour content:

$$ X_s = 0.622 \frac{e_s}{(P - e_s)} $$

$$ X_s = 0.622 \times \frac{2369.5}{(101506 - 2369.5)} = 0.01487 \text{ kg/kg} $$

Latent heat of evaporation:

$$ L = (2502.5 - 2.386 t) \times 1000 $$

$$ L = (2502.5 - 2.386 \times 20.22) \times 1000 = 2454.26 \times 10^3 \text{ J/kg} $$

Substituting the known values into equation (10.56) gives

$$ \frac{dT}{dP} = \frac{0.286 [0.002959 + 0.001282]}{[1 + 0.10416 - 0.03530 + 2.29693]} = 3.604 \times 10^{-4} \text{ °C per Pa} $$

This example shows that the term involving the total moisture content, $X$, is relatively weak (0.10416). Hence, no stringent efforts need be made to measure this factor with high accuracy.

Then

$$ \Delta T_{isen} = \rho_r \frac{dT}{dP} \text{ (equation (10.57))} $$

$$ \Delta T_{isen} = 5800 \times 3.604 \times 10^{-4} = 2.09 \text{ °C} $$
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The isentropic efficiency is

\[ \eta_{isen} = \frac{\Delta T_{isen}}{\Delta T} = \frac{2.09}{2.84} = 0.736 \text{ or 73.6 per cent} \]

10.6.2. Purchasing a main fan.

The results of ventilation network planning exercises will produce a range of pressure-volume duties required of any new major fan that is to be installed. The process of finding and ordering the fan often commences with the ventilation engineer perusing the catalogues of fan characteristics produced by fan manufacturers. Several of those companies should be invited to submit tenders for the manufacture and, if required, installation of the fan. However, in order for those tenders to be complete, information in addition to the required pressure-volume range should be provided by the purchasing organization.

(a) The mean temperature, barometric pressure, humidity and, hence, air density at the fan inlet should be given. This allows data based on standard density to be corrected to the psychrometric conditions expected in the field.

(b) In many cases of both surface and underground fans, noise restrictions need to be applied. These restrictions should be quantified in terms of noise level and, if necessary, with respect to direction.

(c) A plan and sections of the site should be provided showing the proposed fan location and, in particular, highlighting any restrictions on space.

(d) The request for a tender must identify and, wherever possible, quantify the concentrations and types of pollutants to be handled by the fan. These include dusts, gases, water vapour and liquid water droplets. In particular, any given agents of a corrosive nature should be stressed. The purchaser should further indicate any preference for the materials to be used in the manufacture of the fan impeller and casing. Specifications on paints or other protective coatings might also be necessary.

(e) Any preference for the type of fan should be indicated. Otherwise, the manufacturer should be specifically invited to propose one or more fans that will meet the other specifications.

(f) The scope of the required tender should be clearly defined. If the contractor is to be responsible for providing, installing and commissioning the new fan, the individual items should be specified for separate costing.

(g) The motor, transmission, electrical switchgear and monitoring devices may be acquired and installed either by the provider of the fan or separately. In either case, the voltage and any restrictions on power availability should be stated.

(h) Areas of responsibility for site preparation and the provision and installation of ducting should be identified.
10.7. BOOSTER FANS

10.7.1. The application of booster fans

In deep mines or where workings have become distant from surface connections the pressures required to be developed by main fans may be very high in order to maintain acceptable face airflows. This leads to practical difficulties at airlocks and during the transportation of personnel, mined mineral and materials. More serious, however, is the fact that higher pressures at main fans inevitably cause greater leakage throughout the entire system. Any required fractional increase in face airflows will necessitate the same fractional increase in main fan volume flows for any given system resistance. Hence, as both fan operating power and costs are proportional to the product of fan pressure and airflow, those costs can rapidly become excessive as a large mine continues to expand. In such circumstances, the employment of booster fans provides an attractive alternative to the capital penalties of driving new airways, enlarging existing ones, or providing additional surface connections. Legislation should be checked for any national or state restrictions on the use of booster fans.

Unlike the main fans which, in combination, handle all of the mine air, a booster fan installation deals with the airflow for a localized area of the mine only. The primary objectives of a booster fan are

- to enhance or maintain adequate airflow in areas of the mine that are difficult or uneconomic to ventilate by main fans, and
- to redistribute the pressure pattern such that air leakage is minimized.

A modern booster fan installation, properly located, monitored and maintained creates considerable improvements in environmental conditions at the workplace, and can allow the extraction of minerals from areas that would otherwise be uneconomic to mine. It has frequently been the case that the installation of underground booster fans has resulted in improved ventilation of a mine while, at the same time, producing significant reductions in total fan operating costs. However, these benefits depend upon skilled system design and planning. An inappropriate use of booster fans can actually raise operating costs if, for example, fans act in partial opposition to each other. Furthermore, if booster fans are improperly located or sized then they may result in undesired recirculation.

This section is directed towards the planning, monitoring and control of a booster fan installation.

10.7.2. Initial planning and location

An initial step in planning the incorporation of a booster fan into an existing subsurface system is to obtain or update data by conducting ventilation surveys throughout the network (Chapter 6) and to establish a correlated basic network (Section 9.2). Two sets of network exercises should then be carried out.

First, a case must be established for adding a booster fan to the system. Network exercises should investigate thoroughly all viable alternatives. These may include

- adding or upgrading main fans
- enlarging existing airways and/or driving new ones including shafts or other surface connections
- redesigning the underground layout to reduce leakage and system resistance - for example, changing from a U-tube to a through-flow system (Section 4.3), and
- reducing face resistance by a redesign of the face ventilation system or the replacement of line brattices with auxiliary ducts and fans.
If it is decided to progress with the planning of a booster fan installation then a second series of network exercises should be carried out to study the preferred location and corresponding duty of the proposed booster fan. Conventional VNET programs can be used for this purpose. Each of the alternative feasible locations is simulated in a series of computer runs in order to compare flow patterns and operating costs. Flow patterns predicted by the simulations should be checked against the constraints of required face airflows and velocity limits as for any other network exercise (Figure 9.1). Additionally, there are several other checks that should be carried out for a booster fan investigation:

1. Optimization programs have been developed (e.g. Calizaya, 1988) that determine the optimum combination of main and booster fan duties, based on the minimization of total operating costs for each booster fan location investigated by the engineers.

Practical constraints on booster fan location
Although there may be a considerable number of branches in the network within which a booster fan may, theoretically, be sited in order to achieve the required flow enhancement, the majority of those might be eliminated by practical considerations.

Booster fans should, wherever possible, avoid locations where airlocks would interfere with the free movement of minerals, materials or personnel. Furthermore the availability of dedicated electrical power and monitoring circuitry, or the cost of providing such facilities to each potential site should be considered. If a booster fan is located remotely from frequently travelled airways, then particular vigilance must be maintained to ensure that it receives regular visual inspections in addition to continuous electronic surveillance (Section 10.7.3).

Leakage and recirculation
For any given pressure developed by a booster fan in a specified circuit, there exists a fan position where the summation of leakages inby and outby the fan is a minimum without causing undesired recirculation. In a U-tube circuit (Figure 4.4), this location occurs where the operation of the fan achieves zero pressure differential between intake and return airways at the inlet or outlet of the fan. (See, also Figure 21.7.) Conventional wisdom is to locate the booster fan at this “neutral point.” In practice, there are some difficulties. The neutral point is liable to move about quite considerably because of:

- variations in resistance due to face operations and advance/retreat of workings
- changes in the pressures developed by the booster fan or any other fan(s) in the system
- variations in the system resistance due to movement of vehicles or longer term changes in airway resistances.

It is preferable to examine the complete leakage pattern predicted by ventilation network analysis for each proposed booster fan location.

The question of recirculation must be examined most carefully. If the purpose of the booster fan is to induce a system of controlled partial recirculation (Section 4.5), then the predicted airflow pattern must conform to the design value of percentage recirculation. However, undesired or uncontrolled recirculation must be avoided. At all times, legislative constraints must be observed.

Installing a booster fan in one area of a mine will normally cause a reduction in the airflows within other districts of the mine. In extreme cases, reversals of airflow and unexpected recirculations can occur. While examining network predictions involving booster fans, care should be taken to check airflows in all parts of the network and not simply in the section affected most directly by the booster fan.

In addition to modifying airflow patterns, booster fans are also a very effective means of managing the pressure distribution within a network. It is this feature that enables booster fans to influence the leakage characteristics of a subsurface ventilation system. A properly located and
sized booster fan can be far more effective in controlling leakage across worked out areas than sealants used on airway sides or stoppings. Conversely, a badly positioned booster fan can exacerbate leakage problems. These matters are of particular consequence in mines liable to spontaneous combustion (Section 21.4.5). Hence, the pressure differences predicted across old workings or relaxed strata should be checked carefully.

**Steady state effects of stopping the booster fan**
For each proposed location for a booster fan, network simulations should be run to investigate the effect of stopping that fan. In general, the result will be that the airflow will fall in that area of the mine and increase in others. (A cross-cut recirculation booster fan can cause the opposite effect as discussed in Section 4.5.3). In all cases, the ventilation must remain sufficient, without the booster fan, to ensure that all persons can evacuate the mine safely and without undue haste. The implication is that main fans alone must always be able to provide sufficient airflows for safe travel within the mine, and that booster fans merely provide additional airflow to working faces for dilution of the pollutants caused by breaking and transporting rock. It is good practice to employ two or more fans in parallel for a booster fan installation. This allows the majority of the airflow to be maintained if one fan should fail (Section 10.5.2). Further, considerations regarding the transient effects of stopping or starting fans are discussed in Section 10.7.3.

**Economic considerations**
The cost-benefits of a proposed booster fan should be analysed with respect to time. A major booster fan installation should be assured of a reasonable life; otherwise it may be preferable to tolerate short-lived higher operating costs. On the other hand, a cluster of standard axial fans connected in a series/parallel configuration (Section 10.5) can provide an inexpensive booster fan solution to a short term problem. The financial implications of such questions can be quantified by combining total fan operating costs and the capital costs of purchasing and installing the booster fans, using the methods described in Section 9.5.

10.7.3. Monitoring and other safety features
When a booster fan is installed in a subsurface ventilation system, it becomes an important component in governing the behaviour of that system. However, unlike other components such as stoppings, regulators or air crossings, the booster fan is actively powered and has a high speed rotating impeller. Furthermore, it is usually less accessible than a surface fan in case of an emergency condition. For these reasons a subsurface booster fan should be subjected to continuous surveillance. The traditional way of doing this was to post a person at the booster fan location at all times when it was running. This was one of life's more tedious jobs. Some installations reverted to the use of television cameras. A bank of display monitors covering a number of sites could be observed by a single observer at a central location. The problems associated with such methods were that they depended upon the vigilance of human beings and were limited to visual observations. The task may now be undertaken much more reliably and efficiently by electronic surveillance, employing transducers to sense a variety of parameters. These transducers transmit information in a near continuous manner to a central control station, usually located on surface, where the signals are analysed by computer for display, recording and audio-visual alarms when set limits are exceeded. Alarms and, if required, displays may be located both at the control station and also at the fan site.

The monitors used at a booster fan may form part of a mine-wide environmental monitoring system which can, itself, be integrated into a general communication network for monitoring and controlling the condition and activities of conveyors, roof supports, excavation machinery and other equipment. However, sufficient redundancy should be built into the booster fan surveillance such that it remains operational in the event of failures in other parts of the mine monitoring system.
The parameters to be monitored at, or near, a booster fan installation include:

- gas concentrations (e.g. methane) in the air approaching the fan
- carbon monoxide and/or smoke both before and after the fan
- pressure difference across the fan measured, preferably, at the adjoining airlock
- an indication of airflow in the fan inlet or outlet
- bearing temperatures and vibration on both the motor and the fan impeller and
- positions of the airlock and anti-reversal doors.

All monitors should be fail-safe, i.e. the failure of any transducer should be detected by interrogative signals initiated by the computer, resulting in automatic transfer to a companion transducer, display of a warning message and printing of an accompanying hardcopy report. It follows that all transducers must be provided with at least one level of back-up devices.

The monitoring function is particularly important where controlled partial recirculation is practiced or at times of rapid transient change in the ventilation system. The latter may occur by an imprudent operation of doors in other areas of the mine or at a time when either the booster fan itself or other fans in the system are switched on or off. Any sudden reduction in air pressure caused by such activities can cause gases held in old workings or relaxed strata to expand into the ventilating airstream (Section 4.2.2).

A booster fan should be provided with electrical power that is independent of the main supply system to the mine. However, electrical interlocks should be provided according to a predetermined control policy (Section 10.7.4). This policy might include:

- automatic stoppage of the booster fan if the main fan(s) cease to operate
- stoppage of the booster fan if all of the local airlock doors are opened simultaneously
- isolation of electrical power inby if the booster fan ceases to operate.

The second item on this list may also be required in reverse. Hence, for example, the booster fan could not be started if the local airlock doors were all open. However, those doors should open automatically when the booster fan stops, in order to re-establish conventional ventilation by the main fans. The question of interlocks may be addressed by legislation and might also be influenced by existing electrical systems. All electrical equipment should be subject to the legislation pertaining to that mine.

If the fan is to operate in a return airway, then the design will be improved if the motor is located out of the main airstream and ventilated by a split of fresh air taken from an intake airway. In this regard, it is also sensible to ensure that there are no flammable components within the transmission train.

Anti-reversal doors, flaps or louvres should be fitted to a booster fan in order to prevent reversal of the airflow through the fan when it stops.

All materials used in the fan, fan site and within some 50m of the fan should be non-flammable. Furthermore, no flammable materials should be used in the construction of the local airlock, stoppages or air crossings. Finally an automatic fire suppression system should be provided at the fan station.

10.7.4. Booster fan control policy

Continuous electronic monitoring systems provide a tremendous amount of data. During the normal routine conditions that occur for the vast majority of the time, all that is required is that the monitoring system responds to any manual requests to display monitored parameters, and to
record information on electronic or magnetic media. However, the ability to generate and transmit control signals raises the question of how best to respond to unusual deviations in the values of monitored parameters.

The vision of complete automatic control of the mine environment produced a flurry of research activity in the 1960's and '70's (Mahdi, Rustan, Aldridge). This work resulted in significant advances in the development of transducers and data transmission systems suitable for underground use. However, a completely closed loop form of environmental control has seldom been implemented for practical use in producing mines. The difficulty is that there are a number of variables which, although of vital importance in an emergency condition, may not be amenable to direct measurement. These include the locations of personnel and any local actions those personnel might have taken in an attempt to ameliorate the emergency situation.

If an automatic control policy is to be implemented for a booster fan, careful consideration should be given to the action initiated by the system in response to deviations from normal conditions. The control policy that is established should then be incorporated into the computer software (Calizaya, 1989).

The simplest control policy is to activate audio-visual alarms and to allow control signals to be generated manually. Manual over-ride of the system must be possible at all times. In order to examine additional automatic responses, unusual deviations in each of the monitored parameters listed in the previous subsection are considered here:

(a) Methane (or other gas of concern)
Four concentration levels may be chosen or mandated by law

(i) concentration at which personnel are to be withdrawn
(ii) concentration at which power inby the fan is to be cut off
(iii) concentration at which power to the booster fan should be cut
(iv) concentration below which no action is to be initiated.

Levels (i) and (ii) are usually fixed by law. Electrical power inby the fan should be isolated automatically when level (ii) is reached. If the fan is powered by an electric motor which is located within the airstream then the power to the fan should also be cut when level (ii) is reached. However, this may cause even more dangerous conditions for any personnel that are located inby. Having the motor located in a fresh air split allows greater flexibility. At the present time, it is suggested that when level (iii) is exceeded manual control should be established.

Level (iv) is relevant only for those systems where control of the volume flow of air passing through the fan is possible. This may be achieved by automatic adjustment of impeller blade angle, guide vanes or motorized regulators. Level (iv) may be set at one half the concentration at which power inby the fan must be cut. Variations in the monitored gas concentration that occur below level (iv) are to be considered as normal and no action is taken. Variations between levels (iv) and (iii) may result in a PID (proportional/integral/differential) response. This means that control signals will be transmitted to increase the airflow to an extent defined by the level of the monitored concentration, how long that level has existed and the rate at which it is increasing. The objective is to prevent the gas concentration reaching level (iii). The maximum increase in airflow must have been predetermined by network analysis such that the ventilation of the rest of the mine is maintained at an acceptable level.

(b) Smoke or carbon monoxide
The system should be capable of distinguishing hazardous levels of smoke or carbon monoxide from short-lived peaks caused by blasting or nearby diesel equipment. A monitoring system that initiates alarms frequently and without good cause will rapidly lose its effectiveness on human operators. There is no single answer to the response to be initiated if these alarms indicate an actual fire. Similar alarms arising from detectors in other parts of the mine may allow the probable
location of the fire to be determined. Permitting the booster fan to continue running may accelerate the spread of the fire and products of combustion. On the other hand, stopping the fan might allow toxic gases to spread into areas where personnel could have gathered, particularly if damage to the integrity of air crossings or stoppings is suspected. The only realistic action in these circumstances is to revert to manual control.

(c) Pressure differential and airflow
An increase in the pressure developed by the fan will usually indicate an increase in resistance inby. This might be caused by normal face operations, partial blockage of an airway by a vehicle or other means, or obstruction of the fan inlet grill. No action is required other than a warning display and activation of an alarm if the fan pressure approaches stall conditions. A rapid fall in fan pressure accompanied by an increase in airflow is indicative of a short circuit and local recirculation. A warning display should be initiated together with checks on the positions of air lock doors followed, if necessary, by a manual inspection of the area. A simultaneous rapid fall in pressure and airflow suggests a problem with the fan itself. The power supply should be checked and, again, the fan should be inspected manually. Such circumstances will usually be accompanied by indications of the source of the difficulty from other transducers.

(d) Bearing temperatures, vibration
Indications of excessive (or rising) values of bearing temperatures or vibration at either the motor or the fan impeller should result in isolation of power to the fan and all electrical equipment inby the fan.

(e) Positions of airlock doors
An airlock door which is held open for more than a pre-determined time interval should initiate a warning display and, if uncorrected, should warrant a manual inspection.

Bibliography


ASHRAE (1985). Laboratory methods of testing fans for ratings. ANSI/AMCA. Atlanta, Georgia, USA.


APPENDIX A10.1

Comparisons of exhausting and forcing fan pressures to use in pressure surveys.

As stated in Section 10.2 of this chapter, it is fan static pressures that should be employed when a main surface fan is included in the route of a pressure survey. It is not immediately obvious why this should be so when total pressures are used at all other survey measuring points. The apparent anomaly arises from the way in which fan pressures are defined.

In this Appendix, we will consider an identical main fan at a mine surface in each of three configurations. In each case the fan operates at the same speed and passes the same airflow against a constant mine resistance. In such circumstances the fan will deliver the same air power and, hence, produce the same rise in total (facing tube) pressure. In this illustrative example we will take the increase in total pressure across the fan to be $\Delta p_t = 4000 \text{ Pa}$ while the air velocity at the fan measuring station produces a velocity pressure $p_v = 500 \text{ Pa}$.

**Configuration 1.**

Mine Resistance

![Diagram of fan with mine resistance](image)

- $p_v, out = 500 \text{ Pa}$
- $p_s, out = 0$
- $p_t, out = p_v + p_s, out = 500 \text{ Pa}$

- $\Delta p_t = p_t, out - p_t, in = 500 - (-3500) = 4000 \text{ Pa}$

- $p_v, in = p_t, in - p_s, in = -3500 - (-4000) = 500 \text{ Pa}$

(Fig 10.1(c))

Figure A10.1 shows the fan operating in an exhausting configuration with the same inlet and exhaust cross sectional areas and no outlet cone. At the point of exit to the outside atmosphere the velocity pressure remains at 500 Pa but the gauge static pressure is $p_s, out = 0$, giving the total pressure at this location to be

$$p_{t, out} = p_{v, out} + p_{s, out} = 500 + 0 = 500 \text{ Pa}$$

But as $\Delta p_t = p_{t, out} - p_{t, in} = 4000 \text{ Pa}$, it follows that the total pressure shown by a facing tube at the fan inlet must be

$$p_{t, in} = p_{t, out} - \Delta p_t = 500 - 4000 = -3500 \text{ Pa}$$

(negative with respect to the external atmospheric pressure)

and the corresponding static pressure is $p_{s, in} = p_{t, in} - p_{v, in} = (-3500) - 500 = -4000 \text{ Pa}$

As the kinetic energy of the outlet air is dissipated entirely within the external atmosphere, it does not contribute to the ventilating pressure applied to the mine. Hence, that effective ventilating pressure becomes 3500 Pa rather than 4000 Pa and is indicated by the reading on the facing tube gauge, i.e. $p_{t, in}$. Note, however that because of the way in which fan pressures are defined, the total pressure at the fan inlet in a surface exhausting fan is equal to the Fan Static Pressure, FSP (Figure 10.1(c)).
**Configuration 2.**

**Mine Resistance**

![Diagram of a fan with mine resistance](image)

**Figure A10.2 Exhausting fan with an outlet cone, slowing discharge air and reducing exit velocity pressure to 300 Pa.**

Figure A10.2 illustrates the effect of adding an outlet cone to the previous example of a surface exhausting fan. The mine resistance, airflow and fan power remain unchanged. Hence the increase in total pressure across the fan remains at 4000 Pa. However, as shown in the figure, the exit total pressure is now 300 Pa. Hence the gauge inlet total pressure becomes

\[ p_{t,in} = 300 - (-3700) = 4000 \, \text{Pa} \]

Furthermore, as the inlet velocity pressure has remained unchanged at 500 Pa, the inlet static pressure is now

\[ p_{s,in} = p_{t,in} - p_{v,in} = -3700 - 500 = -4200 \, \text{Pa} \]

As in Configuration 1, the kinetic energy of the outlet air is dissipated entirely in the external atmosphere and does not contribute to the ventilating pressure applied to the mine. However, the corresponding exit velocity pressure has now been reduced from 500 Pa to 300 Pa, increasing the effective ventilation pressure applied to the mine from the previous 3500 Pa to 3700 Pa, measured as the total pressure at the fan inlet.

This example further illustrates the advantage of adding an outlet cone to an exhausting fan. Note, however, that losses due to excessive turbulence and/or wall friction can more than counteract this advantage if the cone is too long or diverges too rapidly.²

² Angles of divergence (i.e. between opposite sides) are normally in the range 7 to 15° for mine fans.
Figure A10.3 shows the same fan in a forcing mode with all other parameters as in the previous two configurations. In this case we take the external station to be outside the duct entrance as part of the fan power must be used to accelerate the air from zero to a velocity pressure of 500 Pa\(^3\).

As the total pressure at the external station, \(p_{t,\text{in}}\), is zero and the increase in total pressure across the fan remains at 4000 Pa, it follows that the total pressure at the fan outlet, \(p_{t,\text{out}}\), must be 4000 Pa and the corresponding static pressure becomes

\[
\Delta p_i = p_{t,\text{out}} - p_{t,\text{in}} = 4000 - 0 = 4000 \text{ Pa}
\]

\[
p_{v,\text{out}} = p_{v,\text{in}} = 0
\]

\[
p_{s,\text{in}} = 0
\]

\[
p_{t,\text{in}} = 0
\]

\[
\Delta p_i = p_{t,\text{out}} - p_{t,\text{in}} = 4000 - 0 = 4000 \text{ Pa}
\]

\[
p_{s,\text{out}} + 3500 \text{ Pa (FSP)}
\]

\[
p_{t,\text{out}} + 4000 \text{ Pa (FTP)}
\]

\[
p_{v,\text{out}} = p_{v,\text{in}} - p_{s,\text{out}} = 4000 - 3500 = 500 \text{ Pa}
\]

This is the effective pressure that remains to ventilate the mine and, hence, the pressure to be used if the forcing fan is included in a pressure survey. Once again, because of the way in which fan pressures are defined, this is also the pressure designated as the Fan Static Pressure, FSP, for a forcing fan (Figure 10.1(b)).

\[^3\text{It is also the case that part of the airpower developed by an exhausting fan is utilized in accelerating the air at the intake portals of the mine. However, in that case it is normally measured separately as shock losses at those portals.}\]
APPENDIX A10.2

Derivation of the isentropic temperature-pressure relationship for a mixture of air, water vapour and liquid water droplets.

It is suggested that the reader delays working on this appendix until Chapter 14 (Psychrometry) has been studied.

During a flow process, the work done by expansion or compression of the air is given by $V dP$. In the case of fogged air containing $X$ kg of water (vapour plus liquid) per kg of dry air, this becomes $V_s dP$ where the specific volume of the $(1 + X_s)$ kg of the air/vapour mixture is

$$V_s = \frac{(R + R_v X_s) T}{P} \quad \text{(see equation (14.14))}$$

where

- $P$ = barometric pressure (Pa)
- $R$ = gas constant for dry air (287.04 J/kg K)
- $R_v$ = gas constant for water vapour (461.5 J/kg K)
- $T$ = Absolute temperature (K)
- $X_s$ = mass of water vapour associated with each 1 kg of "dry" air in the saturated space (kg/kg dry air)

The volume of the $(X - X_s)$ kg of liquid water is negligible compared to that of the gases. In a fogged airstream, the wet and dry bulb temperatures are equal.

The steady flow energy equation (10.35) for an isentropic process ($dF = dq = 0$) then becomes

$$V_s dP = \frac{(R + R_v X_s) T}{P} dP = dH \frac{J}{kg} \quad (A10.1)$$

The enthalpy of the $(1 + X)$ kg of air/vapour /liquid mix, based on a datum of 0 °C, is the combination of

$$H = C_p t + X C_w t + L X_s \frac{J}{kg} \quad \text{(A10.3)}$$

where

- $C_p$ = Specific heat of dry air (1005 J/kgK)
- $C_w$ = Specific heat of liquid water (4187 J/kgK)
- $L$ = Latent heat of liquid water at temperature $t$ °C

$$= (2502.5 - 2.386 t)1000 \quad \text{J/kg} \quad \text{(see equation (14.6))}$$

Differentiating,

$$dH = (C_p + X C_w) dt + X_s dL + L dX_s \frac{J}{kg} \quad (A10.4)$$

It is assumed that no water is added or removed during the process. Hence, $X$ remains constant.

We must now seek to formulate this expression in terms of temperature and pressure. First, we evaluate $dL$ and $dX_s$ as functions of temperature and pressure.
(a) Change of latent heat of evaporation, $dL$
From equation (A10.2),
\[ dL = -B \, dt \]
where $B = 2386$  \hspace{1cm} (A10.5)

(b) Change of vapour content, $dX_s$
From equation (14.4) for saturation conditions
\[ X_s = G \frac{e_s}{(P - e_s)} \]  \hspace{1cm} (A10.6)

where  $G = 0.622$
$e_s$ = saturation vapour pressure (Pa) at $t$ °C
and  $P$ = absolute (barometric) pressure (Pa).

By partial differentiation,
\[ dX_s = \frac{\partial X_s}{\partial e_s} \, de_s + \frac{\partial X_s}{\partial P} \, dP \]  \hspace{1cm} (A10.7)

From equation (A10.6)
\[ \frac{\partial X_s}{\partial e_s} = G \left[ \frac{1}{(P - e_s)} + \frac{e_s}{(P - e_s)^2} \right] = \frac{GP}{(P - e_s)^2} \]  \hspace{1cm} (A10.8)
and
\[ \frac{\partial X_s}{\partial P} = -\frac{Ge_s}{(P - e_s)^2} \]  \hspace{1cm} (A10.9)

Furthermore, the Clausius-Clapeyron equation (14.5) gives
\[ de_s = \frac{Le_s}{R_v T^2} \, dt \hspace{1cm} \text{Pa} \]  \hspace{1cm} (A10.10)

where  $R_v = $ gas constant for water vapour (461.5 J/kg K)
and  $T = (273.15 + t)$ \hspace{1cm} (K)
giving  $dt = dT$

Substituting from equations (A10.8, A10.9, and A10.10) into equation (A10.7) gives
\[ dX_s = \frac{GP}{(P - e_s)^2} \frac{Le_s}{R_v T^2} \, dT - \frac{Ge_s}{(P - e_s)^2} \, dP \]  \hspace{1cm} (A10.11)
But as $X_s = \frac{Ge_s}{(P-e_s)}$ from equation (A10.6),

equation (A10.11) becomes

$$dX_s = \frac{X_s P}{(P-e_s)} \frac{L}{R_v T^2} dT - \frac{X_s}{(P-e_s)} dP$$  \hspace{1cm} (A10.12)$$

Having evaluated $dL$ (equation (A10.5)) and $dX_s$ (equation (A10.12)), we can substitute these expressions into equation (A10.4) and, hence, (A10.1)

$$dH = (C_p + X C_w) dt - BX_s dt + \frac{X_s P}{(P-e_s)} \frac{L^2}{R_v T^2} dt - \frac{L X_s}{(P-e_s)} dP = \left( R + R_v X_s \right) \frac{T}{P} dP$$

Collecting the $dt$ and $dP$ terms, and replacing $dt$ by $dT$ for consistency (the two are identical),

$$\frac{dT}{dP} = \frac{\left[ \frac{(R+R_v) X_s}{P} + \frac{L X_s}{(P-e_s)} \right]} {C_p + X C_w - B X_s + \frac{L^2 P}{(P-e_s) R_v T^2}} \ \ ^\circ C \ \ Pa$$  \hspace{1cm} (A10.13)$$

This is a general isentropic relationship describing the rate of change of temperature with respect to pressure for a saturated mixture of air, water vapour and liquid water. All of the variables $X_s, L,$ and $e_s$ depend only upon $T$ and $P$.

For unsaturated air involving no liquid water, $X_s$ becomes $X$ and there is no change of phase. Hence, the terms that include $L$ become zero. Then

$$\frac{dT}{dP} = \frac{(R + R_v X)}{C_p + (C_w - B) X} \frac{T}{P} \ \ ^\circ C \ \ Pa$$  \hspace{1cm} (see footnote ) \hspace{1cm} (A10.14)$$

Inserting values for the constants gives

$$\frac{dT}{dP} = \frac{R}{C_p} \left( 1+1.6078 X \right) \frac{T}{P} \ \ ^\circ C \ \ Pa$$

Expansion of $(1 + 1.7920 X)^1$ and ignoring terms of $X^2$ and smaller leads to the approximation

$$\frac{dT}{dP} = \frac{R}{C_p} \left( 1-0.2 X \right) \frac{T}{P} = 0.286 \left( 1 - 0.2 X \right) \frac{T}{P}$$  \hspace{1cm} (A10.15)$$

\footnote{By considering an isothermal-isobaric change of phase, it can be shown that $C_w - B = C_{pv}$, where $C_{pv}$ = specific heat at constant pressure of water vapour. This gives a value for $C_{pv}$ to be 4187 - 2386 = 1801 J/kgK. The 4.4 per cent deviation from the normally accepted value of 1884 J/kgK indicates the uncertainty of $C_{pv}$ within the atmospheric range of temperatures.}
For dry air, $X = 0$ leaving

$$\frac{dT}{dP} = 0.286 \frac{T}{P} \quad \text{(A10.16)}$$

(This correlates with equation (10.54) when $dT = \Delta T$, $dP = p_r$ and $\eta _{isen} = 1$).

Equation (A10.15) shows that if moist but unsaturated air is assumed to be dry then the fractional error involved is approximately $0.2X$. Hence, for an error of one per cent

$$0.2X = 0.01$$

or

$$X = 0.05 \, \text{kg/kg dry air}$$

This vapour content represents fully saturated conditions at 40 °C and 100 kPa. It is unlikely that such moisture content will exist in mining circumstances without liquid water being present. Thus, for unsaturated air in the normal atmospheric range, assuming dry air will give an accuracy of within one per cent for the temperature/pressure gradient.