MECHANICAL EQUIPMENT
Second Edition
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MECHANICAL CONCEPTS

1.1 FORCE

Newton described force as an action on a body tending to change its state of rest or its uniform motion in a straight line. Thus the change in motion of a body is due to the action of a force. A force can be considered as the action of one body on another, whether in contact (a crowbar being used to move a rock) or not in contact (a magnet attracting a piece of steel - or gravitational pull on a body or its weight).

Force is related to the fundamental quantities of mass, length and time through Newton’s second law \( F = ma \). From strict derivation, the unit of force is therefore \( \text{kg m/s}^2 \). However, as a tribute to the work of the great scientist, the \( \text{kg m/s}^2 \) unit has been given the special name of newton (N) in the SI system.

One newton is that force which, when applied to a mass of one kilogram, produces an acceleration of \( 1 \text{ m/s}^2 \).

1.2 WEIGHT

The weight, or gravitational force on a body, may be determined using Newton’s second law:

\[
\text{Force} = \text{mass} \times \text{acceleration}
\]

where the acceleration is the gravitational acceleration, generally accepted as \( 9.8 \text{ m/s}^2 \). So:

\[
\text{Weight (in newtons)} = \text{mass (in kilograms)} \times 9.8
\]

**EXAMPLE**

Calculate the weight of a rocket, which has a mass of 3.12 tonnes (or \( 3.12 \times 10^6 \text{ kg} \)) when:

(a) on the earth’s surface \( (g = 9.8 \text{ m/s}^2) \)
(b) on the moon’s surface \( (g = 1.62 \text{ m/s}^2) \)

(a) Weight = mass \times \text{gravitational acceleration}

\[
= (3.12 \times 10^6) \times 9.8
= 30.6 \times 10^6 \text{ N} = 30.6 \text{ MN}
\]

(b) Weight = (3.12 \times 10^6) \times 1.62

\[
= 5.05 \times 10^6 \text{ N} = 5.05 \text{ MN}
\]

1.3 FRICTION

Friction arises from two sources:

1. Two surfaces in contact, not being perfectly flat, have only a very small area of true contact. Under load, the material around the area of true contact deforms slightly, thus allowing some atoms of the materials to come sufficiently close together to allow inter-molecular forces of attraction to become significant, as in Figure 1.1

The force required to break all such forces of attraction is the minimum force required to produce motion. Once in motion, these ‘welds’ are continuously created and destroyed.
2. Two surfaces in contact, not being perfectly smooth, have interlocking protrusions, as in Fig 1.2.

The magnitude of the friction force depends upon:

(a) The "area of true contact" of the surfaces, which is much less than the total area.
(b) The nature of the surfaces (rough or smooth).
(c) The load perpendicular to the friction surface - greater load produces greater "areas of true contact".
(d) The shear strength of the material.

Friction is NOT dependent on the actual surface area of the material.

Factors (a), (b) and (d) above may be combined into one constant for a material under a given set of conditions, called the **coefficient of friction**. Symbol $\mu$ (Greek letter mu).

Thus the friction force, or force resisting friction = coefficient of friction $\times$ perpendicular force. ($F = \mu R_N$)

**STATIC AND KINETIC FRICTION**

As shown in Figure 1.3, if a force is gradually applied to a body, the body will eventually move (slide) with an applied force which is less than that required to originally start the body moving.

In this situation two coefficients of friction exist, initially a coefficient of static friction and then a coefficient of kinetic friction. Table 1.1 gives some typical values of the varying coefficient of friction.

<table>
<thead>
<tr>
<th>MATERIALS</th>
<th>STATIC</th>
<th>KINETIC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hard steel on steel</td>
<td>0.78</td>
<td>0.42</td>
</tr>
<tr>
<td>Aluminium on steel</td>
<td>0.61</td>
<td>0.47</td>
</tr>
<tr>
<td>Glass on glass (dry)</td>
<td>0.94</td>
<td>0.40</td>
</tr>
<tr>
<td>Glass on glass (lubricated)</td>
<td>0.35</td>
<td>0.09</td>
</tr>
<tr>
<td>Cast iron on cast iron (dry)</td>
<td>1.1</td>
<td>0.15</td>
</tr>
<tr>
<td>Cast iron on cast iron (lubricated)</td>
<td>0.2</td>
<td>0.07</td>
</tr>
<tr>
<td>Teflon (PTFE) on steel</td>
<td>0.04</td>
<td>0.04</td>
</tr>
</tbody>
</table>

Table 1.1 Friction Coefficients
Chapter One

EXAMPLE

A block of 20 kg mass is acted on by a force of 30 N, which causes it to move at a uniform horizontal velocity of 3 m/s. Determine the coefficient of friction for the surfaces in contact.

Friction force (or force resisting friction) = coefficient of friction x perpendicular force

\[ F = \mu R_N \]
\[ 30 = \mu \times (20 \times 9.8) \]
\[ \mu = \frac{30}{20 \times 9.8} = 0.15 \]

Note that the velocity of 3 m/s is not utilised in the calculation but indicates that the value obtained is for the coefficient of kinetic friction.

EXAMPLE

What force parallel to the slope would be necessary to move a 10 tonne ingot of metal up a 14° slope if the coefficient of friction between the two surfaces is 0.15?

The best way to solve this type of problem is to draw a sketch of the factors to be considered, as in Figure 1.4.

Fig. 1.3 Static and Kinetic Friction

Fig. 1.4
Pull up the slope = component force \((C)\) + friction force \((F)\)

Component force \((C)\) = \((10 \times 10^3 \times 9.8) \times \sin 14^\circ\)

Reaction force \((R_N)\) = \((10 \times 10^3 \times 9.8) \times \cos 14^\circ\)

Friction force = \(\mu R_N = 0.15 \times (10 \times 10^3 \times 9.8 \cos 14^\circ)\)

Pull = \((10 \times 10^3 \times 9.8 \sin 14^\circ) + (10 \times 10^3 \times 9.8 \cos 14^\circ)\)

= \(37.9 \times 10^3\) N = 37.9 kN

1.4 PRESSURE

\[
\text{Pressure} = \frac{\text{Force}}{\text{Area}}
\]

As shown in Figure 1.5, pressure occurs when a force is applied to an area.

Fig. 1.5 Pressure Components

Using the base units of newtons \((N)\) for force and square metres \((m^2)\) for area, the unit of pressure is \(N/m^2\) but this unit is called, for convenience, a pascal \((Pa)\). Because a pascal is an extremely small pressure unit (car tyre pressures are around 200 000 Pa) the more commonly used unit is the kilopascal \((kPa)\) or megapascal \((MPa)\).

Pressures in pascals are used when specifying fan pressures or pressures in ventilation ducting.

You will find reference made in publicity material, magazines and some textbooks to the pressure unit bar \((b)\). Although this unit is not acceptable for engineering use within the Australian metric system, you should be aware of its relationship with the pascal.

\[1\text{ bar} = 100\text{ kilopascals}\]

It is interesting to note that the bar is popular as a pressure unit, particularly in Europe, because it approximates to the atmospheric pressure.

\[1\text{ bar} = 100\text{ kilopascals}\]
\[1\text{ atmosphere} = 101.3\text{ kilopascals}\]

A sub-multiple of the bar was originally used in meteorological work, namely the millibar \((mb)\), for barometric pressures:

\[1000\text{ millibars} = 1\text{ bar}\]
and the standard atmospheric pressure is 1013 millibars. The modern standard for barometric pressure is hectopascals, where, 101.3 kilopascals = 1013 hectopascals.

**POSITIVE AND NEGATIVE PRESSURE**

Because we are subjected to atmospheric pressure, the pressure indicated on a gauge can be either positive or negative.

Above atmospheric pressure is positive and is normally called gauge pressure. Below atmospheric pressure is negative and is usually called vacuum, but the vacuum gauge would be calibrated in - kPa (negative).

Figure 1.6 indicates the relationship between the pressure/vacuum ranges and also introduces the concept of one pressure range starting from absolute zero pressure, called the *absolute pressure* range.

Understanding of these pressure ranges is important because while we normally express pressures in terms of gauge pressure, before these values may be used in any calculation concerning a gas, the value must be converted into an absolute pressure, by adding the accepted value for atmospheric pressure.

\[
\text{absolute pressure (kPa)} = \text{gauge pressure (kPa)} + 101.3
\]

It is important, when specifying pressures or using pressure values in a calculation, to determine if the values given are in terms of gauge pressure (sometimes written kPa g or kPa gauge) or absolute pressure (sometimes written kPa abs).

One other method of pressure specification used is in terms of *head*. Often the characteristics given for a water pump state that the pump will produce a *head* of a given number of metres. This means that the pump will lift the pumped fluid to a certain height.

The relationship between pressure and head is dependent on the mass density of the pumped fluid.

In formula form:

\[
\text{pressure} = (\text{mass density}) \times (\text{gravitational acceleration}) \times (\text{head})
\]

Where:

- pressure is in N/m\(^2\) (symbol \(p\))
- mass density is in kg/m\(^3\) (symbol \(\rho\))
- gravitational acceleration is 9.8 m/s\(^2\) (symbol \(g\))
- head is in metres of fluid (symbol \(h\) or \(H\))
EXAMPLE

A pump is stated to produce a maximum head of 30 metres of water. What is the actual pressure at the pump? Mass density of water is 1000 kg/m$^3$.

Pressure = $\rho g h$

= 1000 x 9.8 x 30

= 294 x 103 N/m$^2$

= 294 kN/m$^2$

Pressure = 294 kPa

Note that this pressure is a gauge pressure. That is, if a gauge were placed in the pump outlet it would indicate 294 kPa and this is the pressure above atmospheric pressure.

EXAMPLE

The maximum gauge pressure available to supply a diver is 180 kPa. To what depth could the diver descend in seawater and still receive an air supply? (The diver must receive air at a pressure equal to the head of water above him.) Mass density of seawater is 1020 kg/m$^3$.

\[
\begin{align*}
\text{pressure} &= \rho gh \\
\frac{h}{\rho g} &= \frac{180 \times 10^3}{1020 \times 9.8} \\
\text{head} &= 18 \text{ metres}
\end{align*}
\]

1.5 TORQUE

Torque = force x radius

Torque produces or tends to produce rotation. It is the input to or output from a drive system that may rotate partially or continuously, as in Figure 1.7.

Using base units of newtons for force and metres for radius, the units for torque are newton metre or N m.

Note that the unit found on many torque wrenches of kg m is not an acceptable torque unit, as it utilises mass x distance, instead of force x distance.

![Fig. 1.7 Components of Torque](image-url)
EXAMPLE

Calculate the torque available from a winch that provides a force of 8 kN at 140 mm radius.

\[
\text{Torque} = \text{Force} \times \text{Radius} \\
= (8 \times 10^3) \times \left(\frac{140}{1000}\right) \\
= 1120 \text{ N m} \text{ or } 1.12 \text{ kN m}
\]

1.6 WORK DONE, ENERGY AND POWER

When a constant force overcomes the resistance of a body and moves that body, work is said to have been done on the body. Thus work done is defined as:

\[\text{force} \times \text{distance moved}\]

the units of work done being newton-metres (N m), also called joules (J).

Note that the basic statement for work done concerns bodies moving on a horizontal plane and where bodies undergo elevation the required additional component force must be determined.

EXAMPLE

A vehicle has a mass of 8 tonnes. Calculate the work done in moving the car one kilometre up an incline of 4°. A force of 900 N is necessary to move the car on a horizontal road.

Work done moving vehicle = force x distance along road

\[= (900) \times 1000 \]
\[= 900 \times 10^3 \text{ N m}\]

Work done in raising = force x distance vehicle up incline

\[= (8000 \times 9.8) \times (1000 \sin 4^\circ) \]
\[= 5.47 \times 10^6 \text{ N m}\]

Total work done on vehicle = (900 x 10^3) + (5.47 x 10^6)

= 6.37 MJ

Work done can also be determined from a graphical representation of force and distance, as in Figure 1.8.

Fig. 1.8 Force/Distance Graph
If the work done is considered as the area beneath a thin vertical strip of the graph, the total work done is the total area under the curve. This system is used when considering the work done during the cycle of a machine, such as an engine or a compressor.

Although the axis units are different, as in Figure 1.9, the multiple (or area) of pressure and volume is still work done.

WORK DONE BY A TORQUE

Work may also be done by a constant torque overcoming a resistance:

\[
\text{Work done} = \text{applied torque} \times \text{angle turned}
\]
\[
\text{N m (or J)} \quad (\text{N m}) \quad (\text{radians})
\]

**EXAMPLE**

Calculate the work done per second when a torque of 200 N m rotates a shaft at 750 r/min.

\[
750 \, \text{r/min} = 750 \times 2\pi \, \text{radians/minute}
\]
\[
= \frac{750 \times 2\pi}{60} \, \text{radians/second}
\]

Work done = applied torque × angle turned
\[
= 200 \times \frac{750 \times 2\pi}{60}
\]
\[
= 15.7 \, \text{kJ per second}
\]

The work done units (joules) is also used for energy, the two most frequently used forms of energy in mechanical engineering being potential energy and kinetic energy.

Potential energy is that energy in a body due to its position, normally above a datum as in Figure 1.10.
Potential energy of the mass due to its position equals its gravitational force x height above the datum, and the body could dissipate this amount of energy in falling to the datum.

Kinetic energy of a body is due to its velocity, as in Figure 1.11.

**EXAMPLE**

A vehicle of 8 tonnes mass is travelling at 72 km/h on the horizontal. Resistance to motion is 900 N. If the engine is switched off, how far would it travel up a 4° slope?

Initial kinetic energy = final potential energy + work done against friction

\[ \frac{1}{2}mv^2 = mgh + \text{work done against friction} \]

Substituting 72 km/h = 20 m/s and letting travel up slope = \( x \) metres

\[ \frac{1}{2} \times (8000 \times 20^2) = (8000 \times 9.8 \times x \sin 4°) + (900x) \]

\[ 1.6 \times 10^6 = (5.5 \times 10^3)x + 900x \]

\[ x = \frac{1.6 \times 10^6}{(5.5 \times 10^3) + 900} = 251 \text{ metres} \]

Power is the rate of doing work. It may also be considered as the rate at which energy is provided or at which energy is consumed.

\[ \text{Power} = \frac{\text{work done}}{\text{time taken}} = \frac{J}{s} \]

Alternatively:

\[ \text{Power} = \frac{\text{force applied} \times \text{distance travelled}}{\text{time taken}} \]
Mechanical Concepts

1.10 velocity-time

Distance = \( \text{velocity} \times \text{time} \)

Power = force applied \( \times \) resultant velocity

Note that the base unit of power is the watt, but in practice the larger multiple is normally used, such as the kilowatt or the megawatt.

**EXAMPLE**

Calculate the power required to maintain the speed of a car at 72 km/h if the force required to overcome road and air resistance is 1.3 kN.

\[
\text{Power} = \text{force} \times \text{velocity}
\]

and 72 km/h = 20 m/s

\[
\text{Power} = (1.3 \times 10^3) \times 20 = 26 \times 10^3 \text{ W or } 26 \text{ kW}
\]

Considering rotational power, as previously stated:

work done = torque \( \times \) angle of rotation (radians)

and work done per second = torque \( \times \) radians per second

But,

work done per second = power

and radians per second = revolutions per second \( \times \) 2\( \pi \)

So,

power = \(2\pi \times \text{revolutions per second} \times \text{torque}\)

Or, in symbol form:

\[
\text{Power} = 2\pi NT
\]

**EXAMPLE**

Calculate the torque available from an electric motor with an output of 5.5 kW at 1440 r/min.

\[
\text{Power} = 2\pi NT
\]

\[
5.5 \times 10^3 = 2\pi \times \frac{1440}{60} \times T
\]

\[
T = \frac{(5.5 \times 10^3) \times 60}{2\pi \times 1440} = 36.5 \text{ N m}
\]

1.7 EFFICIENCY

In all practical machines, some power is lost in overcoming friction or is dissipated as heat. The efficiency of a machine is the ratio of output power to the input power, or the actual output power compared with the theoretical output power.
Efficiency = \frac{\text{output power}}{\text{input power}}

and the efficiency is normally expressed as a percentage.

Note that the value of efficiency is always less than 100% and this fact should be recalled when calculating the efficiency of a machine. A value in excess of 100% indicates a calculation error.

**EXAMPLE**

An electric motor-driven hoist consumes 4 kW when lifting a mass of 500 kg through 2 m at constant velocity in 4 seconds. Calculate the overall efficiency of the hoist.

\[
\text{Input power} = 4 \text{ kW} \\
\text{Output power} = \frac{\text{force} \times \text{distance}}{\text{time}} \\
= \frac{(500 \times 9.8) \times 2}{4} \\
= 2450 \text{ W} \\
\text{Efficiency} = \frac{\text{output power}}{\text{input power}} \\
= \frac{2450}{4000} = 0.6125 \text{ or } 61.25\%
\]

**EXAMPLE**

Water is raised through a height of 15 m by a centrifugal pump. Input to the pump is 20 kW and the system efficiency is 75%. Calculate the quantity of water pumped in one hour. (1 litre of water has a mass of 1 kg.)

\[
\text{Input power} = 20 \text{ kW} \\
\text{Efficiency} = \frac{\text{output power}}{\text{input power}} \\
= \frac{75}{100} \times \frac{\text{output power}}{20 \times 10^3} \\
\text{Output power} = 15 \text{ kW} \\
\text{Output power} = \frac{\text{force} \times \text{distance}}{\text{time}}
\]

Let \( x \) = the mass of water raised and consider what happens over one second:

\[
15 \times 10^3 = \frac{\text{force} \times 15}{1} \\
\text{force/second} = \frac{15 \times 10^3}{15} = 1000 \text{ N/s}
\]

To calculate the quantity of water lifted, knowing that 1 litre of water is 1 kilogram:
Mechanical Concepts

\[ 1000 \text{ N/s} = \frac{1000}{9.8} \text{ kg/s} = \frac{1000}{9.8} \text{ L/s} \]

\[ = \frac{1000 \times 3600}{9.8} \text{ litres/hour} \]

\[ = 367 \text{ kL per hour} \]
Chapter Two

FLUIDS AND THERMAL CONCEPTS

2.1 PROPERTIES OF A FLUID

Matter can exist in three basic forms, as either a solid, a liquid or a gas.

*Solids* retain their shape and have no tendency to flow.

*Liquids and gases* flow readily and are known as *fluids*.

Liquids offer considerable resistance to compression and are not greatly affected by changes in temperature. The reduction in volume of a liquid because of compression is so small that it is usually ignored. Gases are easily compressed and are subject to change owing to temperature.

Consequently, although it is not entirely correct, it is usual to consider a liquid as an incompressible fluid and a gas as a compressible fluid.

*Hydraulics* is the general term for the study of fluids - it is not necessarily confined to water but includes oil as well.

Just as the compressibility of a liquid is small, its coefficient of expansion is also low and thus it undergoes little change in volume because of temperature change. Thus a liquid may be described as:

1. Being practically incompressible.
2. Occupying a definite volume.

The *mass density* of a fluid is its mass per unit volume, normally kilograms per cubic metre.

Symbol \( \rho \) (Greek letter rho).

For example:

- mass density of water at 4°C = 1000 kg/m\(^3\)
- mass density of air = 1.23 kg/m\(^3\)

The *relative density* (originally called specific gravity) of a fluid is the mass density of the fluid compared with the mass density of water:

For example:

- relative density of petrol = 0.8
- relative density of water = 1
- relative density of mercury = 13.6

The *specific volume* of a fluid is its volume per unit mass, units m\(^3\)/kg, and is thus the reciprocal of the mass density:

\[
\text{Specific Volume} \ (\text{m}^3/\text{kg}) = \frac{\text{mass density of fluid}}{\text{mass density of water}}
\]
The viscosity of a fluid is its ability to resist shearing forces when it is in motion.

Considering Figure 2.1, and assuming that

(a) the fluid velocity at the stationary surface is zero, and
(b) the rate of change of velocity is related to the distance from the stationary surface,

then the force to move the plate is proportional to the absolute viscosity of the fluid

\[
\text{force} \propto \text{absolute viscosity} \times \text{fluid thickness}
\]

where absolute viscosity = \( \frac{\text{shear stress}}{\text{shear strain}} \)

or absolute viscosity = \( \frac{\text{force}}{\text{area} \times \text{fluid thickness}} \)

In unit form, this is:

\[
\text{absolute viscosity} = \frac{\text{N m}^2}{\text{m s}} = \text{Pa s}
\]

Because a Pa s is a large unit, the usual form for absolute viscosity is \( \text{mPa s} \), which is itself known as a centipoise (cP):

For example:

Absolute viscosity of water = 1.12 cP
Absolute viscosity of an oil = 273 cP

The absolute viscosity of a liquid may be measured by the method shown in Figure 2.1, sliding a plate with a measured force and velocity, but the system tends to be cumbersome to operate. Equipment used to measure viscosity (or viscometers) measures the kinematic viscosity, which is basically the absolute viscosity divided by the fluid's mass density:

\[
\text{kinematic viscosity} = \frac{\text{absolute viscosity}}{\text{mass density}}
\]
In unit form, this would be:

\[
\text{kinematic viscosity} = \frac{\text{Ns/m}^2}{\text{kg/m}^3} = \text{m}^2/\text{s}
\]

A more practical unit for kinematic viscosity is mm\(^2\)/s, which has the name of centistoke (cSt):

For example:
- kinematic viscosity of water = 1.12 cSt
- kinematic viscosity of an oil = 279 cSt

A more familiar viscosity unit is the range of SAE numbers used for motor vehicle lubricants. It should be realised that these numbers relate to a viscosity range, rather than a precise viscosity rating.

Table 2.1 provides a comparison between SAE numbers (Society of American Engineers) and cSt values:

<table>
<thead>
<tr>
<th>SAE</th>
<th>cSt at 0°C</th>
<th>cSt at 100°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>10W</td>
<td>1 300 - 2 600</td>
<td>5.7 - 9.6</td>
</tr>
<tr>
<td>20W</td>
<td>2 600 - 10 400</td>
<td>9.6 - 12.9</td>
</tr>
<tr>
<td>30W</td>
<td>3 260 - 21 700</td>
<td>12.9 - 16.7</td>
</tr>
<tr>
<td>20</td>
<td>5.7 - 9.6</td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>9.6 - 12.9</td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>12.9 - 16.7</td>
<td></td>
</tr>
<tr>
<td>140</td>
<td>25 - 42</td>
<td></td>
</tr>
</tbody>
</table>

Table 2.1 Relationship Between SAE and cSt

The compressibility of a fluid is its resistance to change in shape, the change in volume being a function of its original volume and the applied pressure. Mathematically, the modulus of elasticity or bulk modulus of a fluid at a point is defined by:

\[
\text{bulk modulus} = -\frac{\Delta p}{\Delta V}
\]

where the negative sign indicates that as the pressure increases the volume decreases. It is usually assumed that the bulk modulus is insensitive to pressure but is affected by temperature.

For water, the bulk modulus is approximately 2 MPa and for steel the modulus of elasticity is approximately 200 MPa, so it can be seen that water is relatively compressible.

### 2.2 THERMAL CONCEPTS

The temperature of a body is its hotness or coldness expressed in terms of the expansion or contraction of a gas, a liquid or a solid.

The concept of hotness or coldness originated from the sensation felt by a human body on coming into contact with some other body, but temperature is now more frequently defined as a measure of the activity of the molecules within the body.
The temperature of a body is measured by one of the methods detailed in Chapter 4 and a value given in degrees celsius, with one hundred divisions between freezing point (0°C) and boiling point (100°C) of water.

In thermodynamics, there exists a constant need for temperature expressed in absolute values, which has been determined from a datum at which it is estimated a gas has no volume. In practice, this point may not be achieved because a gas would liquefy and then solidify before reaching the datum point, but this point may be extrapolated from experimental results as shown in Figure 2.2.

The volume of a gas is measured at varying temperatures and the results graphed. The graph intersects the temperature at -273°C (at zero volume) and this point is called absolute zero temperature and a new scale formed with units of kelvins (symbol K). Note that absolute temperature is expressed in kelvins and not degrees kelvin. Thus:

\[
\text{Absolute temperature (K)} = (\text{°C} + 273)
\]

Heat energy passes from one body to another when a temperature difference exists between them. Because temperature is a measure of the activity of the molecules within the body, heat is said to flow from the more active molecules to the less active molecules, or from the hotter to the cooler parts of the body.

Heat may be generated by the combustion of a fuel. This heat is then supplied to the working substance that is used by a heat engine as a medium for converting heat energy into mechanical energy.

The quantity of heat energy required to achieve a specific change in a material is its specific heat. The quantity of heat supplied to or removed from a body depends on three variables:

1. Mass of the body.
2. Specific heat of the body's material.
3. Change in temperature that occurs.

Consider a quantity of water being heated by a bunsen burner that is adjusted to supply heat at a constant rate. A definite time will be required to raise the water to boiling point. If the mass of water is increased by a known ratio, then the heating time will also increase by the same ratio while the change in temperature in both cases remains the same.
The Specific Heat of a substance is defined as the amount of heat passing into or out of a unit mass of the substance in order to change its temperature by one degree celsius. Because the addition or removal of heat increases or decreases the internal energy of the body, heat itself must be a form of energy and therefore must be expressed in energy units:

For example: joules

Thus:

\[ Q = mc(T_2 - T_1) \]

Where

\( Q \) = heat transferred to produce the temperature change is measured in joules (J)
\( c \) = Specific heat of substance in joules per kilogram kelvin (J/kg K)
\( m \) = mass of substance in kilograms (kg)
\( T_2 \) = final body temperature in kelvins (K)
\( T_1 \) = initial body temperature in kelvins (K)

or \( c = \frac{Q}{m(T_2 - T_1)} \) with units of \( \frac{[J]}{[kg \ K]} \)

The specific heat of a substance is a characteristic of that substance. The values of specific heat for various substances are tabulated below for future reference.

<table>
<thead>
<tr>
<th>SUBSTANCE</th>
<th>( c ) [J/kg K]</th>
<th>SUBSTANCE</th>
<th>( c ) [J/kg K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>917</td>
<td>Glycerine</td>
<td>2429</td>
</tr>
<tr>
<td>Benzene</td>
<td>1718</td>
<td>Lead</td>
<td>0 130</td>
</tr>
<tr>
<td>Brass</td>
<td>0 385</td>
<td>Mercury</td>
<td>0 138</td>
</tr>
<tr>
<td>Cast Iron</td>
<td>0 447</td>
<td>Nickel</td>
<td>0 460</td>
</tr>
<tr>
<td>Copper</td>
<td>0 394</td>
<td>Steel</td>
<td>0 480</td>
</tr>
<tr>
<td>Ether</td>
<td>2219</td>
<td>Tin</td>
<td>0 230</td>
</tr>
<tr>
<td>Ethyl Alcohol</td>
<td>2303</td>
<td>Zinc</td>
<td>0 394</td>
</tr>
<tr>
<td>Glass</td>
<td>0 674</td>
<td>Water</td>
<td>4187</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Ice</td>
<td>2040</td>
</tr>
</tbody>
</table>

Table 2.2 Specific Heat Values

As will be shown later, it is necessary to show the specific heats of gases under conditions of constant pressure or constant volume:

<table>
<thead>
<tr>
<th>GAS</th>
<th>( c_p ) [J/kg K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>992</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>842</td>
</tr>
<tr>
<td>Carbon Monoxide</td>
<td>1013</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>14 319</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>984</td>
</tr>
<tr>
<td>Oxygen</td>
<td>1013</td>
</tr>
</tbody>
</table>

where \( c_p \) is the specific heat of the gas under constant pressure conditions.
EXAMPLE

Calculate the quantity of heat required to raise 10 kg of air from 30°C to 500°C. Assume that constant pressure conditions prevail:

\[
\begin{align*}
m &= 10 \text{ kg} \\
T_1 &= 30 + 273 = 303 \text{ K} \\
T_2 &= 500 + 273 = 773 \text{ K} \\
(T_2 - T_1) &= 773 - 303 = 470 \text{ K}
\end{align*}
\]

Also 500°C - 30°C = 470°C so, in this case, it is not absolutely necessary to convert the temperature into the kelvin scale. Extreme care must be exercised if this short cut is to be used because most formulae to be introduced later MUST be calculated using the kelvin scale.

\[
Q \text{ [J]} = m \times c \times (T_2 - T_1) \text{ [K]}
\]

\[
= 10 \times 992 \times 470
\]

\[
= 4.66 \times 10^6 \text{ [J]}
\]

\[
= 4.66 \text{ MJ}
\]

2.3 WATER EQUIVALENT

The water equivalent of a body may be defined as the mass of water that requires the same quantity of heat to raise its temperature by one degree as would be required to raise the temperature of the body one degree.

EXAMPLE

Find the water equivalent of an iron container which has a mass of 10 kg if the specific heat of the iron is 477 J/kg K.

\[
Q = \text{Mass} \times \text{Specific Heat} \times \text{Temperature Change}
\]

for iron

\[
Q_I = m_I \times c_I \times (T_2 - T_1)_I
\]

for water

\[
Q_W = m_W \times c_W \times (T_2 - T_1)_W
\]

but

\[
Q_I = Q_W \quad \text{and} \quad (T_2 - T_1)_I = (T_2 - T_1)_W
\]

Thus

\[
m_W \times c_W = m_I \times c_I
\]

Mass of water =

\[
\frac{m_W \times c_I}{c_W} = \frac{10 \times 477}{4187} = 1.14 \text{ kg}
\]

Water equivalent = 1.14 kg

This means that 1.14 kg water requires the same quantity of heat as 10 kg of iron to raise its temperature by 1°C or 1 K. Water equivalent is often used in calculating the results of laboratory experiments.

EXAMPLE

You are required to find the specific heat of an unknown metal. A copper calorimeter, of mass 0.3 kg, contains 0.5 kg of water at 10°C and is perfectly insulated. Small particles of the unknown metal, of mass 0.6 kg are heated to 60°C and then immersed in the water in the calorimeter. The resulting temperature is 12.5°C. What is the specific heat of the metal?
Because the calorimeter is perfectly insulated it can be assumed that there are no heat losses, that is, all the heat in the hot metal flows into the cooler water and calorimeter until a heat balance is obtained.

\[
\text{Water Equivalent} = \frac{\text{Mass of Body} \times \text{Specific Heat of Body}}{\text{Specific Heat of Water}}
\]

\[
\text{W.E. of calorimeter} = \frac{0.3 \times 394}{4187} = 0.028 \text{ kg}
\]

Thus, exactly the same effect will be obtained if the hot metal is placed directly into \((0.028 + 0.5) = 0.528\) kg of water at \(10^\circ\text{C}\).

Now the heat lost by the metal is:

\[
Q_M = m_M \times c_M \times (T_2 - T_1)_M
\]

\[
= 0.6 \times c_M \times (60 - 12.5)
\]

\[
= 28.5 \text{ } c_M \text{ J}
\]

and the heat gained by the calorimeter and water is:

\[
Q_W = m_W \times c_W \times (T_2 - T_1)_W
\]

\[
= 0.528 \times 4187 \times (12.5 - 10)
\]

\[
= 5526 \text{ J}
\]

but \(Q_M = Q_W\) because there are no losses

\[
28.5 \text{ } c_M = 5526
\]

\[
\text{c}_M = 194 \text{ J/kg K}
\]

**LATENT HEAT**

When a substance is changing its state, such as from solid to liquid or liquid to gas, energy (heat) must be supplied to overcome the mutual attraction between the atoms and molecules comprising the substance. Because of this, no temperature change occurs, even though large quantities of heat may be added during the change of state. The reverse applies when condensing a vapour or freezing a liquid.

When a substance changes its physical state from solid to liquid, the process is called *melting* or *fusion* and the quantity of heat supplied to overcome the mutual attraction between the atoms and molecules is called the *latent heat of fusion*.

At \(0^\circ\text{C}\) the Latent Heat of Fusion of Ice = 335 kJ/kg:

So 335 kJ of heat are required to change 1 kg of ice at \(0^\circ\text{C}\) into 1 kg of water at \(0^\circ\text{C}\).

Similarly, 335 kJ of heat must be removed from 1 kg of water at \(0^\circ\text{C}\) to convert it to ice at \(0^\circ\text{C}\). The process of boiling or evaporation occurs when a substance changes its state from a liquid to a vapour. The heat supplied, at constant temperature, to effect this change of state is called the *latent heat of evaporation*. 
At atmospheric pressure:

Latent Heat of Evaporation of Water = 2256.7 kJ/kg:

So 1 kg of water at 100°C requires 2256.7 kJ of heat to convert it to 1 kg of steam at 100°C. In the same way, 1 kg of steam at 100°C must have 2256.7 kJ of heat removed from it to condense the steam to 1 kg of water at 100°C.

The boiling temperature of a liquid and its Latent Heat of Evaporation are definitely dependent upon the pressure exerted on the liquid surface. Boiling temperature of water increases as the pressure increases but the Latent Heat of Evaporation decreases as the pressure increases.

<table>
<thead>
<tr>
<th>ABSOLUTE PRESSURE</th>
<th>BOILING TEMPERATURE</th>
<th>LATENT HEAT OF EVAPORATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>101.3 kPa</td>
<td>100°C</td>
<td>2256.7 kJ/kg</td>
</tr>
<tr>
<td>1 MPa</td>
<td>179.9</td>
<td>2015</td>
</tr>
<tr>
<td>10 MPa</td>
<td>311</td>
<td>1317</td>
</tr>
<tr>
<td>22.12 MPa</td>
<td>374.15</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 2.3 Latent Heat of Evaporation

**EXAMPLE**

How much heat must be supplied to a block of ice of mass 10 kg to convert it into steam at atmospheric pressure? The initial temperature of the ice was -20°C.

When heat is supplied and it results in a temperature change, it is often referred to as *sensible heat* thus distinguishing between Latent Heat where no temperature change occurs and heat where a temperature change does occur. Consider the rise in temperature of the ice from -20°C to 0°C:

\[
\text{Sensible Heat} = m_i \times c_i \times (T_2 - T_1) = 10 \times 2040 \times (0 - (-20)) = 408000 \text{ J} = 408 \text{ kJ}
\]

Now consider the Latent Heat of Fusion required to change the ice at 0°C into water at 0°C:

\[
Q_2 = 10 \times 335 = 3350 \text{ kJ}
\]

Now the water must be raised from 0°C to 100°C:

\[
\text{Sensible Heat} = m_w \times c_w (T_2 - T_1)_w = 10 \times 4187 \times (100 - 0) = 4187 \text{ kJ}
\]

Finally, the water at 100°C must be changed into steam at 100°C:

\[
Q_4 = 10 \times 335 = 22567 \text{ kJ}
\]
Thus the total heat supplied
\[ Q_1 + Q_2 + Q_3 + Q_4 = 408 + 3350 + 4187 + 22567 = 30.512 \text{ MJ} \]

**EXAMPLE**

0.5 kg of ice at -10°C is put into 10 kg of water at 30°C, the water being contained in a glass vessel of mass 5 kg. Finally, a steel ball of mass 2 kg and temperature 150°C is added. Assuming no losses of heat, what is the final temperature of the water?

Sensible Heat required to raise the temperature of the ice to 0°C
\[ m_i \times c_i \times (T_2 - T_1) = 0.5 \times 2040 \times (0 + 10) = 10.2 \text{ kJ} \]

Total Latent Heat required to melt the ice
\[ = \text{Mass} \times \text{Latent Heat} = 0.5 \times 335 = 167.5 \text{ kJ} \]

Heat absorbed by converting ice to water at 0°C
\[ = 10.2 + 167.5 = 177.7 \text{ kJ} \]

Let \( \theta \)° C = Final temperature after the steel ball has been added:

Sensible Heat to raise 0.5 kg water to \( \theta \)°C
\[ m_w \times c_w \times (T_2 - T_1) = 0.5 \times 4187 \times (\theta - 0) = 2.049 \theta \text{ kJ} \]

Total Heat absorbed by ice = (177.7 + 3.09\( \theta \)) kJ

Water equivalent of glass vessel = \( \frac{\text{mass of vessel} \times \text{SH of glass}}{\text{SH of water}} \)
\[ = \frac{5 \times 674}{4187} = 0.805 \text{ kg} \]

Sensible Heat given up by 10 kg of water and glass vessel
\[ m_w \times c_w \times (T_1 - T_2) = (10 + 0.805) \times 4187 \times (30 - \theta) = 45.241 (30 - \theta) \text{ kJ} \]
Sensible Heat given up by steel ball
\[ = m_s \times c_s \times (T_1 - T_2)_s \]
\[ = 2 \times 480 \times (150 - \theta) \]
\[ = 0.96 (150 - \theta) \text{ kJ} \]

Because there are no heat losses, the heat gained by the ice must equal the heat given up by the 10 kg of water, the glass vessel and the steel ball:

Heat absorbed = Sensible Heat given + Sensible Heat given up by steel ball
by ice up by water + glass

(177.7 + 2.09\theta) = 45.241 (30 - \_\_) + 0.96 (150 - \theta)
= 1357 - 45.24\_ + 144 - 0.96 \theta

\[ \theta = 27.4^\circ \text{C} \]

Although it is a simplification, it is not necessary to calculate the problem using the concept of the water equivalent. The glass vessel can be considered as another body giving up heat:

Sensible Heat given up by glass vessel
\[ = m_G \times c_G \times (T_1 - T_2)_G \]
\[ = 5 \times 674 \times (30 - \theta) \]
\[ = 3.37 (30 - \theta) \text{ kJ} \]

Sensible Heat given up by 10 kg water
\[ = m_w \times c_w \times (T_1 - T_2)_w \]
\[ = 10 \times 4187 (30 - \theta) \]
\[ = 41.87 (30 - \theta) \text{ kJ} \]

The previous statement about heat gained = heat given up is still valid:

\[ \text{heat absorbed by ice} = \text{sensible heat given} + \text{sensible heat given} + \text{sensible heat given} \]
up by steel ball up by glass up by water

(177.7 + 2.09\theta) = 0.96 (150 -\theta) + 3.37 (30 - \theta) + 41.87 (30 -\theta)

\[ \theta = 27.4^\circ \text{C as before} \]

2.4 GAS LAWS

Boyle’s Law states that "The absolute pressure of a given mass of a perfect gas varies inversely as its volume if the temperature remains unchanged".

Expressing this mathematically:

\[ p = \frac{1}{V} \]
Chapter Two

Thus \[ p = \text{constant} \times \frac{1}{V} \]

\[ pV = \text{constant} \]

\[ p_1 V_1 = p_2 V_2 = p_3 V_3 = \text{etc.} \]

**EXAMPLE**

A cylinder contains 4 m\(^3\) of gas at a pressure of 150 kPa and a gas-tight piston compresses the gas to a volume of 1 m\(^3\) at constant temperature. Construct a pressure-volume diagram showing the relationship between pressure and volume during the movement of the piston.

Consider four states of volume.

Initial: \[ 4 \text{ m}^3 \quad V_1 \]
\[ 3 \text{ m}^3 \quad V_2 \]
\[ 2 \text{ m}^3 \quad V_3 \]

Final: \[ 1 \text{ m}^3 \quad V_4 \]

For a volume of 4 m\(^3\) the pressure is 150 kPa absolute

\[ pV = \text{Constant} \]
\[ = 150 \times 4 \]
\[ = 600 \]

For a volume of 3 m\(^3\)

\[ p_2 V_2 = 600 \]
\[ p_2 = 600 \div 3 = 200 \text{ kPa abs.} \]

For a volume of 2 m\(^3\)

\[ p_3 V_3 = 600 \]
\[ p_3 = 600 \div 2 = 300 \text{ kPa abs.} \]

For a volume of 1 m\(^3\)

\[ p_4 V_4 = 600 \]
\[ p_4 = 600 \div 1 = 600 \text{ kPa abs.} \]

Figure 2.3 shows the four pressures plotted on a pressure/volume graph. The diagram shows the variation of pressure with change of volume. The curve produced is a rectangular hyperbola and the operation is termed *isothermal* compression.
Charles’ Law states that "The volume of a given mass of a perfect gas varies directly as its absolute temperature if the pressure remains unchanged". Also, "the absolute pressure varies directly as the absolute temperature if the volume remains unchanged".

From the above statement we have:

For constant pressure \( V \propto T \) (the symbol \( \propto \) means ‘proportional to’)

Thus \( \frac{V}{T} = \text{constant} \)

Hence \( \frac{V_1}{T_1} = \frac{V_2}{T_2} \)

or \( \frac{V_1}{V_2} = \frac{T_1}{T_2} \)

For constant volume \( p \propto T \)

Thus \( \frac{p}{T} = \text{constant} \)

Hence \( \frac{p_1}{T_1} = \frac{p_2}{T_2} \)

or \( \frac{p_1}{p_2} = \frac{T_1}{T_2} \)

**EXAMPLE**

The pressure of oxygen in a gas cylinder is 1500 kPa gauge and the temperature is 80°C. A fire in the vicinity causes the temperature of the cylinder and its contents to rise to 150°C. Find the new pressure of the oxygen if the atmospheric pressure is 105 kPa absolute. Neglect the increase in volume of the cylinder.
\[ T_1 = 80 + 273 = 353 \text{ K} \]
\[ T_2 = 150 + 273 = 423 \text{ K} \]
\[ p_1 = 1500 + 105 = 1605 \text{ kPa abs} \]

For constant volume:

\[ \frac{p_1}{p_2} = \frac{T_1}{T_2} \]
\[ p_2 = \frac{p_1 T_1}{T_2} = \frac{1605 \times 423}{353} = 1927 \text{ kPa absolute} \]
\[ = 1822 \text{ kPa gauge} \]

**COMBINATION OF BOYLE’S AND CHARLES’ LAWS**

Each one of these laws states how one quantity varies with another if the third quantity remains unchanged. If the three quantities change simultaneously, it is necessary to combine these laws in order to determine the final conditions of the gas.

Referring to the first example, let the gas be compressed from its initial state of pressure \( p_1 \), volume \( V_1 \), and temperature \( T_1 \), to its final state of \( p_2 \), \( V_2 \) and \( T_2 \). To arrive at the final state, let it pass through two stages, the first to satisfy Boyle's Law and the second to satisfy Charles' Law.

Imagine the piston pushing inward to compress the gas until it reaches its final pressure \( p_2 \) and let its volume then be represented by \( V \) as in Figure 2.4 (a).

![Fig. 2.4 Actual Compression](image)

Normally, the temperature would tend to increase owing to the work done in compressing the gas, but any heat so generated must be taken away from it during the compression so that its temperature stays constant at \( T_1 \), hence following Boyle's Law:

\[ p_1 V_1 = p_2 V_2 \]
Now apply heat to raise the temperature to $T_2$ and at the same time draw the piston outwards to prevent a rise of pressure and keep it constant at $p_2$ (Figure 2.4 (b)). The volume will increase in direct proportion to the increase in absolute temperature, according to Charles’ Law:

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \quad \text{(ii)}$$

By substituting the value of $V$ from (ii) into (i)

$$V = \frac{V_2 T_1}{T_2}$$

Substituting in (i):

$$p_1 V_1 = p_2 \frac{V_2 T_1}{T_2}$$

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

This combined law of Boyle’s and Charles’ is true for a given mass of any perfect gas subject to any form of expansion or compression. Figure 2.4 (c) represents the actual direct compression diagram.

**CHARACTERISTIC EQUATION**

Since the ratio $\frac{pV}{T}$ is a constant, its numerical value can be determined for any quantity of a perfect gas.

Since a fixed mass of gas was considered:

$$\frac{pV}{T} = \text{mass constant}$$

or $$\frac{pV}{T} = mR$$

where $m$ is the mass in kilograms

and $R$ is the *gas constant* in kJ/kg K.

For example, $R$ for air = 0.287 kJ/kg K

**EXAMPLE**

Air at 17°C and 125 kPa absolute pressure occupies 2.46 m$^3$. If the air is compressed to a volume of 1 m$^3$ and a pressure of 700 kPa absolute, calculate its final temperature and mass.

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

$$\frac{(125 \times 10^3) \times 2.46}{(17 + 273)} = \frac{(700 \times 10^3) \times 1}{T_2}$$

$$T_2 = 660.3 \text{ K}$$

$$= 387.3 \text{°C}$$
Using initial conditions:

\[
\frac{pV}{T} = mR
\]

\[
\frac{(125\times10^3)\times2.46}{(17 + 273)} = m \times (0.287\times10^3)
\]

\[
m = 3.69 \text{ kg}
\]

### 2.5 EXPANSION AND COMPRESSION OF GASES

Work may be obtained from an internal combustion engine by allowing a gas to expand against a piston and drive it forward. When a gas is compressed or expanded it follows the law \( pV^n = c \) where \( c \) is a constant.

It is fairly easy to check this by taking actual values of the pressure/volume relationship in a cylinder and graph the values, as in Figure 2.5.

![Fig. 2.5 Pressure/Volume Relationship](image)

The law relating pressure and volume may be determined by further plotting the pressure/volume values but on logarithmic scale paper, as in Figure 2.6. The slope of the straight-line graph produced provides a value for \( n \), called the index of compression.

![Fig. 2.6](image)
If the graph is a straight line, then the expansion or compression curve must be of the form:

$$pV^n = c$$

Taking logarithms of both sides:

$$\log p + n \log V = \log c$$
$$\log p = n \log V + \log c$$

Which is of the form

$$y = mx + c$$

This is a straight line of slope $-n$

Figure 2.7 shows a variety of ways that a gas may expand in accordance with the law $pV^n = c$ from an initial pressure $p$, and initial volume $V$. Note that the reverse conditions occur during compression.

There are an infinite number of such expansions, each with a different value of $n$, but two of them are of particular importance, the ISOTHERMAL expansion and the ADIABATIC expansion.

Isothermal Expansion occurs when the value of $n = 1$

If $n = 1$, then $pV = c$

But for a gas, the characteristic equation states $pV = mRT$

So, $mRT = c$ when $n = 1$

Now, since the mass of the gas remains constant, and $R$ is the characteristic gas constant, then during this process $T$ must be constant.

Fig. 2.7 Types of Gas Expansion
An ISOTHERMAL expansion or compression of a gas is one in which the temperature of the gas remains constant throughout the process.

In practice, an isothermal expansion is approached when the operation is carried out very slowly, so that heat can be taken in and the temperature remains constant.

Adiabatic Expansion occurs when the value of \( n = \frac{c_p}{c_v} \) which is the ratio of the specific heat of the gas at constant pressure \( (c_p) \) and constant volume \( (c_v) \).

The ratio \( \frac{c_p}{c_v} \) is given the Greek symbol gamma (\( \gamma \)).

By definition, ADIABATIC expansion occurs when no heat flow takes place across the boundaries of the system.

In practice, adiabatic expansion and compression are only approached when the operation is carried out very quickly, so that there is no time for heat flow to occur. All expansions or compressions of the form \( pV^n = c \), other than the particular isothermal or adiabatic operations, are called polytropic operations.

Fig 2.7 shows a number of polytropic expansions, those lying above the isothermal curve having a value of \( n \) less than 1; those lying between the isothermal and adiabatic curves having a value of \( n \) greater than 1 but less than gamma; and those lying below the adiabatic curve having a value of \( n \) greater than gamma.

Since for a polytropic process \( pV^n = c \), and also for a gas the characteristic equation gives \( \frac{pV}{T} = \) constant, we can obtain a useful relationship between temperatures and pressures, or temperatures and volumes, for any two points during the process as follows:

\[
pV^n = c \quad \text{so,} \quad p_1V_1^n = p_2V_2^n \quad \text{...............(i)}
\]

and also

\[
\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2} \quad \text{...............(ii)}
\]

From (ii)

\[
\frac{T_2}{T_1} = \frac{p_2V_2}{p_1V_1} \quad \text{...............(iii)}
\]

To eliminate volumes from (i):

\[
\left( \frac{V_2}{V_1} \right)^n = \frac{p_1}{p_2}
\]

So

\[
\frac{V_2}{V_1} = \left( \frac{p_1}{p_2} \right)^{\frac{1}{n}} = \left( \frac{p_2}{p_1} \right)^{-\frac{1}{n}}
\]

and substituting in (iii)
Fluids and Thermal Concepts

So \( \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right) \left( \frac{p_1}{p_2} \right)^{\frac{1}{\gamma}} \),

\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right) \left( \frac{p_1}{p_2} \right)^{\frac{1}{\gamma}} \]

Or, eliminating pressures from equation (i) and substituting in (iii)

\[ \frac{T_2}{T_1} = \left( \frac{V_1}{V_2} \right)^{\frac{\gamma}{\gamma-1}} \]

\[ \frac{T_2}{T_1} = \left( \frac{V_1}{V_2} \right)^{\frac{\gamma}{\gamma-1}} \]

Giving the combined relationship for any polytropic process:

\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma}{\gamma-1}} = \left( \frac{V_1}{V_2} \right)^{\gamma-1} \]

This very useful equation is well worth committing to memory.

The above equations are reproduced from HEAT ENGINES AND APPLIED HEAT by F. Metcalfe, with the permission of Cassell Ltd.

EXAMPLE

2.5 litres of gas is expanded in the cylinder of an engine from 700 kPa absolute pressure and 1100°C to a final pressure of 270 kPa absolute, according to the law \( pV^{1.3} = c \). If \( R = 0.289 \text{ kJ/kg K} \), calculate:

(a) final temperature
(b) final volume
(c) mass of the gas.

(a)

\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma}{\gamma-1}} \]

\[ \frac{T_2}{(1100 + 273)} = \left( \frac{(270 \times 10^3)}{(700 \times 10^3)} \right)^{\frac{1.3}{1.3-1}} \]

\[ T_2 = 1102 \text{ K} = 829°C \]

(b)

\[ \left( \frac{p_2}{p_1} \right)^{\frac{\gamma}{\gamma-1}} = \left( \frac{V_1}{V_2} \right)^{\gamma-1} \]

\[ \left( \frac{(270 \times 10^3)}{(700 \times 10^3)} \right)^{\frac{1.3}{1.3-1}} = \left( \frac{(2.5 \times 10^{-3})}{V_2} \right)^{1.3-1} \]

\[ 0.8^{\frac{1}{1.3}} = \frac{(2.5 \times 10^{-3})}{V_2} \]

\[ V_2 = 5.26 \times 10^{-3} \text{ m}^3 \]
Note that the formula $\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$ could have been used by utilising the previously calculated $T_2$.

(c)

\[
\frac{pV}{T} = mR \\
m = \frac{pV}{TR} \\
= \frac{(700\times10^3)\times(2.5\times10^{-3})}{(1100 + 273)\times(0.289\times10^3)} \\
= 0.004 \text{ kg}
\]
STEAM

3.1 PROPERTIES OF STEAM

The properties of steam depend very much on its pressure and temperature.

Consider 1 kg of water initially at 0°C being heated in a vessel fitted with a movable piston such that the pressure remains constant, as in Figure 3.1.

During the first stages of heating the temperature of the water will rise until the water boils, at a temperature known as the saturation temperature which depends on the pressure in the vessel.

After the boiling temperature is reached, steam begins to form but the temperature remains constant, as in Figure 3.2.

Until the point is reached at which all the water is converted to steam, the contents of the vessel will be a mixture of water and steam known as wet steam.

Eventually all the water, including those droplets held in suspension, will be evaporated and at this instant, the substance is known as dry saturated steam. The substance has, in theory, become a gas at the same temperature as that at which the water boiled.

As heating continues further, the temperature begins to rise again and the steam is now known as superheated steam, and behaves as a gas. To define the condition of superheated steam it is necessary to state both the
3.2 ENTHALPY OF STEAM

A very useful property of fluids is enthalpy which is defined as:

$$ H = u + pV $$

Enthalpy = Internal Energy + Work Done

For steam, the zero of enthalpy is taken arbitrarily to occur at a temperature of 0°C and tables usually give the values of the enthalpy of 1 kg of boiling water (h_f) and dry saturated steam (h_g) at various pressures.

For 1 kg of water which is not boiling, the specific enthalpy may be estimated from the formula:

$$ h = 4187t \text{ joules} $$

where \( t \) is the water temperature. This neglects the flow work term \( pV \), which is very small except at high pressures. The continuous formation of steam in a boiler is a flow process in which a supply of feed water is pumped into the boiler equivalent to the amount of steam drawn off.

Neglecting changes of potential and kinetic energy, the general energy equation states:

$$ Q = H_2 - H_1 $$

This means that the heat taken in is equal to the change of enthalpy. The heat taken in during the process of converting 1 kg of boiling water into dry saturated steam is known as the latent heat of evaporation (l) and is the difference between the specific enthalpies of boiling water and dry saturated steam:

$$ l = h_g - h_f $$

In wet steam, although all the water has boiled, only part of it has received its latent heat and changed into steam. That fraction of the initial 1 kg of water which has received its latent heat and has become steam, is called the dryness fraction (x) of the steam. The dryness fraction may have any value from zero (corresponding to boiling water) to unity (corresponding to dry saturated steam).

The specific enthalpy of wet steam is therefore given by:

$$ h_{wet} = h_f + x(h_g - h_f) $$

For example, the enthalpy of 1 kg of steam of dryness fraction 0.9 at 2 MN/m² is:

$$ h_{wet} = 908.6 + (0.9 \times 1888.6) $$

$$ = 2608.3 \text{ kJ/kg} $$

For superheated steam, since theoretically the substance has now become a gas, the increase of enthalpy of 1 kg of steam during the superheating state may be calculated from the equation for the heating of gases at constant pressure, that is, specific heat capacity multiplied by the degrees of superheat.

Hence the specific enthalpy of superheated steam is given by:

$$ h_{sup} = h_g + c_p(t_{sup} - t_{sat}) $$
where  \( c_p \) = mean specific heat capacity over a range the range of superheating
\( t_{sup} \) = temperature of superheated steam
\( t_{sat} \) = temperature of steam formation.

The value of the specific heat capacity of superheated steam is not the same over different ranges of
temperature, and the specific enthalpy of superheated steam is best obtained from tables.

**EXAMPLE**

Determine the enthalpy of superheated steam at 1 MPa absolute pressure and 400\( ^\circ \)C if the saturation
temperature (\( t_{sat} \)) is 179.9\( ^\circ \)C and the mean specific heat capacity over the temperature range is 2.21 kJ/kg. The
enthalpy of dry saturated steam (\( h_g \)) under the given conditions is 2776.2 kJ/kg.

\[
H_{sup} = h_g + c_p (t_{sup} - t_{sat})
\]
\[
h_{sup} = 2776.2 + 2.21(400 - 179.9) = 3262.6 \text{ kJ/kg}
\]

**NOTE:**
1. Temperatures are not changed to kelvins because a difference in temperature is required.
2. The enthalpy values are not changed to base units (J/kg) because a factor of 10\(^3\) occurs on both sides
   of the equation.

### 3.3 VOLUME OF STEAM

The specific volume of wet steam is written as \( V_{wet} \).

1 kg of wet steam of dryness fraction \( x \) consists of \( x \) kg of pure steam and \((1 - x)\) kg of water held in
suspension:

\[
V_{wet} = x V_g + (1 - x) V_f
\]

where \( V_g \) is the specific volume of dry saturated steam
and \( V_f \) the specific volume of water at formation pressure.

In most cases, except when the steam is very wet, the volume of the water in relation to that of the steam is so
small that it may be neglected, and we write:

\[
V_{wet} = x V_g
\]

The specific volume of superheated steam (\( V_{sup} \)) may be determined directly from tables, or calculated from
the formula:

\[
V_{sup} = \frac{230.8(h_{sup} - 1941)}{p} \text{ m}^3/\text{kg}
\]

where \( h_{sup} \) is the enthalpy in kJ/kg
\( p \) is the pressure in N/m\(^2\)

**EXAMPLE**

Determine the volume of 1 kg of steam at 500 kPa absolute superheated to 500\( ^\circ \)C. The enthalpy of
superheated steam is 3484 kJ/kg.
3.4 USE OF STEAM TABLES

Under normal conditions, a steam engine or turbine consumes steam at the same rate at which it is generated; therefore the steam may be considered to be generated at constant pressure. The temperature at which water boils (the vaporisation point) depends strictly on the pressure exerted on it. The higher the pressure, the higher the temperature.

EXAMPLE

<table>
<thead>
<tr>
<th>Pressure (kPa absolute)</th>
<th>2.3</th>
<th>19.9</th>
<th>101.3</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boiling Point (°C)</td>
<td>2.0</td>
<td>60</td>
<td>100</td>
<td>120.2</td>
</tr>
</tbody>
</table>

The temperature of the steam produced at any given pressure is the same as the temperature of the boiling water at the same pressure. Thus, if a boiler is operating at 200 kPa absolute, the water begins to boil as its temperature reaches 120.2°C and steam is generated at the same temperature.

Steam which is in contact with the boiling water from which it is produced is called saturated steam and the temperature at which it is produced is called the saturation temperature.

Steam without any water particles held in suspension is called dry saturated steam. Normally, the steam does contain water particles and is known as wet steam and its quality is expressed as a dryness fraction where:

\[
\text{dryness fraction} = \frac{\text{latent heat supplied}}{\text{latent heat required for conversion}}
\]

In order to completely dry the steam and raise its temperature above saturation temperature, the steam must be taken completely away from contact with water and extra heat added, in a superheater. Thus, steam whose temperature is higher than the temperature of saturated steam at the same pressure is called superheated steam.

The value of superheating may be realised when it is considered that steam is a means of conveying energy, and the higher the temperature of the steam, the more energy it can transfer.

A lot of experimental work has been done on the properties of steam and the results have been published in various forms under the title of Steam Tables. A section from some steam tables is shown in Tables 3.1 and 3.2 and will be used in examples.

Note that not all the columns shown will be explained because not all of the information is relevant to this subject, but full use of the tables is made in modules such as Thermodynamics.
Chapter Three

Table 3.1  Saturated Water and Steam Tables

<table>
<thead>
<tr>
<th>$p$ [bar]</th>
<th>$t_s$ [°C]</th>
<th>$v_g$ [m$^3$/kg]</th>
<th>$u_f$ [kJ/kg]</th>
<th>$u_g$ [kJ/kg]</th>
<th>$h_f$ [kJ/kg]</th>
<th>$h_g$ [kJ/kg]</th>
<th>$s_f$ [kJ/kg K]</th>
<th>$s_g$ [kJ/kg K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>99.6</td>
<td>1.694</td>
<td>417</td>
<td>2506</td>
<td>417</td>
<td>2258</td>
<td>2675</td>
<td>1.303</td>
</tr>
<tr>
<td>1.1</td>
<td>102.3</td>
<td>1.549</td>
<td>429</td>
<td>2510</td>
<td>429</td>
<td>2251</td>
<td>2680</td>
<td>1.333</td>
</tr>
<tr>
<td>1.2</td>
<td>104.8</td>
<td>1.428</td>
<td>439</td>
<td>2512</td>
<td>439</td>
<td>2244</td>
<td>2683</td>
<td>1.361</td>
</tr>
<tr>
<td>1.3</td>
<td>107.1</td>
<td>1.325</td>
<td>449</td>
<td>2515</td>
<td>449</td>
<td>2238</td>
<td>2687</td>
<td>1.387</td>
</tr>
<tr>
<td>1.4</td>
<td>109.3</td>
<td>1.236</td>
<td>458</td>
<td>2517</td>
<td>458</td>
<td>2222</td>
<td>2690</td>
<td>1.411</td>
</tr>
<tr>
<td>1.5</td>
<td>111.4</td>
<td>1.159</td>
<td>467</td>
<td>2519</td>
<td>467</td>
<td>2216</td>
<td>2693</td>
<td>1.434</td>
</tr>
<tr>
<td>1.6</td>
<td>113.3</td>
<td>1.091</td>
<td>475</td>
<td>2521</td>
<td>475</td>
<td>2211</td>
<td>2696</td>
<td>1.455</td>
</tr>
<tr>
<td>1.7</td>
<td>115.2</td>
<td>1.031</td>
<td>483</td>
<td>2524</td>
<td>483</td>
<td>2206</td>
<td>2699</td>
<td>1.475</td>
</tr>
<tr>
<td>1.8</td>
<td>116.9</td>
<td>0.9774</td>
<td>491</td>
<td>2526</td>
<td>491</td>
<td>2201</td>
<td>2702</td>
<td>1.494</td>
</tr>
<tr>
<td>1.9</td>
<td>118.6</td>
<td>0.9292</td>
<td>498</td>
<td>2528</td>
<td>498</td>
<td>2196</td>
<td>2704</td>
<td>1.513</td>
</tr>
<tr>
<td>2.0</td>
<td>120.2</td>
<td>0.8856</td>
<td>505</td>
<td>2530</td>
<td>505</td>
<td>2192</td>
<td>2707</td>
<td>1.530</td>
</tr>
<tr>
<td>2.1</td>
<td>121.8</td>
<td>0.8461</td>
<td>511</td>
<td>2531</td>
<td>511</td>
<td>2188</td>
<td>2709</td>
<td>1.547</td>
</tr>
<tr>
<td>2.2</td>
<td>123.3</td>
<td>0.8100</td>
<td>518</td>
<td>2533</td>
<td>518</td>
<td>2183</td>
<td>2711</td>
<td>1.563</td>
</tr>
<tr>
<td>2.3</td>
<td>124.7</td>
<td>0.7770</td>
<td>524</td>
<td>2534</td>
<td>524</td>
<td>2179</td>
<td>2713</td>
<td>1.578</td>
</tr>
<tr>
<td>2.4</td>
<td>126.1</td>
<td>0.7466</td>
<td>530</td>
<td>2536</td>
<td>530</td>
<td>2174</td>
<td>2715</td>
<td>1.593</td>
</tr>
<tr>
<td>2.5</td>
<td>127.4</td>
<td>0.7186</td>
<td>535</td>
<td>2537</td>
<td>535</td>
<td>2169</td>
<td>2717</td>
<td>1.607</td>
</tr>
<tr>
<td>2.6</td>
<td>128.7</td>
<td>0.6927</td>
<td>541</td>
<td>2539</td>
<td>541</td>
<td>2165</td>
<td>2719</td>
<td>1.621</td>
</tr>
<tr>
<td>2.7</td>
<td>130.0</td>
<td>0.6686</td>
<td>546</td>
<td>2540</td>
<td>546</td>
<td>2160</td>
<td>2720</td>
<td>1.634</td>
</tr>
<tr>
<td>2.8</td>
<td>131.2</td>
<td>0.6462</td>
<td>551</td>
<td>2541</td>
<td>551</td>
<td>2156</td>
<td>2722</td>
<td>1.647</td>
</tr>
<tr>
<td>2.9</td>
<td>132.4</td>
<td>0.6253</td>
<td>556</td>
<td>2543</td>
<td>556</td>
<td>2151</td>
<td>2724</td>
<td>1.660</td>
</tr>
<tr>
<td>3.0</td>
<td>133.5</td>
<td>0.6057</td>
<td>561</td>
<td>2544</td>
<td>561</td>
<td>2146</td>
<td>2725</td>
<td>1.672</td>
</tr>
</tbody>
</table>

The first column is the absolute pressure in bars (where 1 bar = 100 kPa) symbol $p$.

The second column gives the saturation temperature $(t_s)$ in °C.

Consider the atmospheric pressure, which at 1.013 bars is between 1 and 1.1 on the table; the corresponding temperature should be 100°C and is shown as between 99.6°C and 102.3°C.

Note that more detailed tables are available to provide intermediate values but for the majority of engineering work the required value may be estimated from the information given.

The third column $V_g$ gives the specific volume of the steam in m$^3$/kg.

Columns six, seven and eight are concerned with enthalpy and all have units of kJ/kg.

The sixth column provides values for the sensible heat $h_f$ measured above 0°C. It is measured above the fusion point of water because it is extremely unlikely that, for instance, feed water being pumped into a boiler would contain lumps of ice.

The $h_f$ value indicates the heat required to raise 1 kg of water from freezing (or fusion) point to saturation temperature at the required pressure.

It should be noted that the tables provide a more precise value for the specific heat of water.
If you consider the top and bottom lines of Table 3.1:

\[ at \quad t_s = 99.6^\circ C \]

Specific heat \[ \frac{h_f}{t_s} = \frac{417}{99.6} = 4.186 \text{kJ/kg} \]

\[ at \quad t_s = 133.5^\circ C \]

Specific heat \[ \frac{h_f}{t_s} = \frac{561}{133.5} = 4.2 \text{kJ/kg} \]

which indicates the variance from the generally accepted value of 4.187 kJ/kg.

Column seven \( (h_{fg}) \) gives the latent heat of evaporation, or the amount of heat energy to completely evaporate 1 kg of water into 1 kg of steam at the saturation temperature given in column two.

Column eight \( (h_g) \) gives the total enthalpy of dry saturated steam, and is the sum of columns six and seven:

\[ h_f + h_{fg} = h_g \]

Examples of the use of the values in Table 3.1:

1. Determine the enthalpy per kilogram of boiling water at 110, 190 and 280 kPa absolute pressure.

\[ h_f = 429, 498, 551 \text{ kJ/kg} \]

2. Determine the temperature at which water boils when subjected to a pressure of 200 and 300 kPa absolute

\[ t_s = 120.2^\circ C \text{ and } 133.5^\circ C \]

3. Determine the volume per kilogram of dry saturated steam at 140 and 240 kPa absolute pressure.

\[ V_g = 1.236 \text{ and } 0.7466 \text{ m}^3/\text{kg} \]

4. Calculate the enthalpy per kilogram of steam at 110 kPa absolute, 0.9 dry.

\[ h_{WET} = h_f + (x.h_{fg}) \]

\[ = 429 + (0.9 \times 2251) \]

\[ = 2455 \text{ kJ/kg} \]

5. Calculate the change in enthalpy per kilogram when water at 25\degree C is converted to 0.95 dry steam at a constant pressure of 270 kPa absolute.

Initial enthalpy of water \[ = 25 \times 4.187 = 104.7 \text{ kJ/kg} \]

Final enthalpy of wet steam \[ h_f + (x.h_{fg}) \]

\[ = 546 + (0.95 \times 2174) = 2611.3 \text{ kJ/kg} \]

Change in enthalpy \[ = 2611.3 - 104.7 = 2506.6 \text{ kJ/kg} \]
6. Calculate the heat required to raise 2 kg of water initially at 30°C to dry saturated steam at 220 kPa absolute.

Initial enthalpy/kg of water = 30 \times 4.187 = 125.7 \text{ kJ/kg}

At 220 kPa \quad h = 2711 \text{ kJ/kg}

Change in enthalpy = 2711 - 125.7 = 2585.3 \text{ kJ/kg}

Total heat required for 2 kg = 2585.3 \times 2 = 5170.6 \text{ kJ}

---

**Superheated Steam**

<table>
<thead>
<tr>
<th>( p ) (atm)</th>
<th>( t ) (°C)</th>
<th>350</th>
<th>375</th>
<th>400</th>
<th>425</th>
<th>450</th>
<th>500</th>
<th>600</th>
<th>700</th>
</tr>
</thead>
<tbody>
<tr>
<td>80 (295-0)</td>
<td>v_e = 0.02352</td>
<td>v \times 10^4</td>
<td>2.994</td>
<td>3.220</td>
<td>3.428</td>
<td>3.625</td>
<td>3.812</td>
<td>4.170</td>
<td>4.839</td>
</tr>
<tr>
<td></td>
<td>h_e = 2758</td>
<td>h</td>
<td>2990</td>
<td>3067</td>
<td>3139</td>
<td>3207</td>
<td>3272</td>
<td>3398</td>
<td>3641</td>
</tr>
<tr>
<td>90 (303-3)</td>
<td>v_e = 0.02048</td>
<td>v \times 10^4</td>
<td>2.578</td>
<td>2.794</td>
<td>2.991</td>
<td>3.173</td>
<td>3.346</td>
<td>3.673</td>
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<td>3256</td>
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<td>100 (311-0)</td>
<td>v_e = 0.01802</td>
<td>v \times 10^4</td>
<td>2.241</td>
<td>2.453</td>
<td>2.639</td>
<td>2.812</td>
<td>2.972</td>
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<td>110 (318-0)</td>
<td>v_e = 0.01598</td>
<td>v \times 10^4</td>
<td>1.960</td>
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<td>2.350</td>
<td>2.514</td>
<td>2.666</td>
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<td>1.931</td>
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<td>3.159</td>
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<td>3134</td>
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<td>130 (330-8)</td>
<td>v_e = 0.01278</td>
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<td>1.509</td>
<td>1.726</td>
<td>1.901</td>
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<td>1.321</td>
<td>1.548</td>
<td>1.722</td>
<td>1.872</td>
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<td>2.250</td>
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<td>3322</td>
<td>3590</td>
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<td></td>
<td>s_e = 5.373</td>
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<td>5.946</td>
<td>6.079</td>
<td>6.193</td>
<td>6.390</td>
<td>6.716</td>
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<td>v_e = 0.01035</td>
<td>v \times 10^4</td>
<td>1.146</td>
<td>1.391</td>
<td>1.566</td>
<td>1.714</td>
<td>1.844</td>
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<td>2.487</td>
</tr>
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<td>3157</td>
<td>3309</td>
<td>3581</td>
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<td>5.443</td>
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<td>6.023</td>
<td>6.142</td>
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<td>6.677</td>
</tr>
</tbody>
</table>

Table 3.2 Saturated Steam Table

**SUPERHEATING**

As previously stated in Part 3.1, additional heat may be added to steam to increase its energy content, and the level of superheating is often referred to as degrees of superheat.

Table 3.3 shows a portion of the Steam Tables that are devoted to superheated steam.

Note in Table 3.3 that pressure and saturated temperature are located together and the values of \( v_g \) and \( h_g \) are given for a range of temperatures.
EXAMPLES

1. Determine the saturation temperature of steam at 12 MPa absolute and the specific volume and enthalpy of superheated steam at 425°C.

From Table 3.3 using 120 bar (12 MPa)

\[
\begin{align*}
t & = 324.6°C \\
v & = 2.265 \times 10^{-2} \text{ m}^3/\text{kg} \\
h_g & = 3134 \text{ kJ/kg}
\end{align*}
\]

2. Calculate the increase in enthalpy when 2 kg of steam initially at 0.8 dryness is superheated to 425°C at a constant pressure of 9 MPa absolute.

Using values from Table 3.2 for 90 bar (9 MPa):

\[
\begin{align*}
\text{Initial enthalpy per kg} & = h_i + x h_g \\
& = 1364 + (0.8 \times 1379) \\
& = 2467 \text{ kJ/kg}
\end{align*}
\]
Chapter Three

For superheated steam at 9 MPa and 425°C, from Table 3.3.

Final enthalpy = 3189 kJ/kg

Change in enthalpy = 3189 – 2467 = 722 kJ/kg

So, for 3 kilograms:
Change in enthalpy = (722 x 3) = 2166 kJ
MEASUREMENT OF MECHANICAL VARIABLES

4.1 TORQUE MEASUREMENT

The measurement of torque from an engine or motor may be achieved with a device called a dynamometer, the two categories of dynamometer being absorption type and transmission type.

The absorption dynamometer converts work done on it (by the input) into heat. This conversion is accomplished by frictional means or the conversion of work into electrical energy which is subsequently converted to heat by passage through an electrical resistance. In either case the heat is dissipated to atmosphere and is regarded as a waste product of testing.

The transmission dynamometer operates in conjunction with some external load upon which the power source is performing economically useful work. In this case the dynamometer's sole function is to measure the torque being transmitted and does not itself absorb any of the power source's output.

THE ABSORPTION DYNAMOMETER

The majority of routine engine test-work utilises an absorption dynamometer.

As shown in Figure 4.1, the algebraic sum of the torques acting in a system which is in equilibrium must be zero.

The shaft exerts a turning motion or torque $T$ and has affixed to one end an arm of effective length $L_B$. A weight $W$ is suspended from the arm as shown:

\[
\text{clockwise torque } (WL_B) = \text{anti-clockwise torque } (T)
\]

\[
WL_B = T
\]

Fig. 4.1 Torque Components
The unknown torque $T$ may be determined from the product $W \times L_B$. In practice of course, the engine output shaft is not stationary but the same principle may be applied.

A simple method of measuring torque is shown in Figure 4.2.

![Fig. 4.2 Measurement of Torque](image)

The torque applied is the product of the drum radius and the spring balance reading charge ($W - w$).

Figure 4.3 shows two methods of attaching a drum to an output shaft.

![Fig. 4.3 Drum Dynamometers](image)

The drum is surrounded by a steel band attached to a lever arm. The band is lined with a friction material, usually a fibre composition, and equipped with a clamping screw to enable the force between the friction material and the drum to be adjusted.
A rider weight suspended from the band ensures that the centre of gravity of the band and lever arm assembly is vertically in line with the centre of rotation. Two fixed stops limit the rotation of the braking assembly.

With the engine running, the frictional force between the drum and the friction material will carry the band round in a clockwise direction until the lever arm is restrained by the upper stop. Slippage between the drum and friction material will then commence, converting the work output of the engine into heat. If weights are now added to the lever arm until the arm “floats” in a horizontal position between the stops, the torque exerted by the band assembly is a product of the force caused by the weights and the perpendicular distance to that force from the centre of rotation.

In the left-hand diagram of Figure 4.3, the braking effect is carried by changing the spring tension and in the lower diagram of Figure 4.3, the braking effect is varied by tightening the hand assembly.

One problem with this basic form of dynamometer is that the heat produced causes the drum to reach a high temperature very quickly. Imagine attempting to check the output of a 10 kW electric motor - the drum would have to absorb a heat input equivalent to 10 single-element electric radiators. Change in drum temperature causes a change in the coefficient of friction of the braking material and the system becomes unstable.

Figure 4.4 shows a typical method of reducing the heat build-up, by circulating water through the inside of the drum. This type of equipment also includes provision for the automatic adjustment of band tension. If the small arm moves to touch $S_1$ or $S_2$, the band tension is increased or decreased respectively.

With this dynamometer the input torque is $(W \times L_B) - (w \times l)$ newton metres, where $W$ and $w$ are the weights in newtons and $L_B$ and $l$ their distances from the shaft centre in metres.

**THE HYDRAULIC DYNAMOMETER**

Figure 4.5 shows a section of a Heenan-Froude hydraulic dynamometer, a device which converts the input torque to a flow of water, the kinetic energy transmitted by the water being measurable in terms of force applied.

It consists of a shaft carrying a rotor which revolves inside a water-tight casing. In each face of the rotor there are a number of radial vanes set obliquely to the axis of the rotor, and the spaces between these vanes form cups of semi-elliptical section. Each face of the casing is also provided with vanes forming a similar set of cups. The rotor vanes face forward in the direction of rotation; the casing vanes face in the opposite direction. See Figure 4.6 for details.
Measurement of Mechanical Variables

Fig. 4.5 Heenan-Froude Dynamometer

When the rotor is driven, water is flung outwards by centrifugal force and streams obliquely from the outer edge of each cup with a velocity compounded by axial and tangential components. These streams are flung into the cups in the casing and exert a force on it, tending to drag it around with the rotor. The cups in the casing serve as guides to return the water to the inner part of the rotor, where it is again flung out, and thus the work done by the engine is absorbed and converted into heat by a continuous series of whirling eddies in the water.

The heat thus generated is disposed of by arranging a continuous flow of water fed to the back of the cups in the casing by a flexible rubber pipe which offers negligible constraint to the casing, while an overflow pipe at the top provides an exit for the warm water that is displaced.

The rate of flow should be regulated by a valve on the overflow pipe and not at the inlet valve, which should be kept wide open. The makers recommend running at an outflow temperature which should normally not exceed 60°C.

Fig. 4.6 Heenan-Froude Dynamometer Details

The load on the engine may be varied by inserting or withdrawing thin sluice plates between rotor and casing, thus rendering more or less of the casing cups operative in the formation of the power-absorbing eddies. These sluice plates are operated by a hand wheel on the outside of the casing. The rotor shaft runs in bearings.
carried by the casing and is furnished with a water-tight gland where it enters and leaves. The casing itself is
supported on large ball-bearing trunnions, and to it is attached an arm at which the torque is measured.

The power of the engine is absorbed in three ways:

1. shaft-bearing friction
2. gland friction
3. the continuous change in direction and momentum of the water,

and since all three tend to turn the casing in the direction in which the engine is running, it is clear that the
total turning effort may be measured by a single reading of a spring balance attached to the arm.

Care should be taken that the dynamometer is accurately balanced on its trunnions, so that when at rest, but
with the water flowing through the casing, the reading of the spring balance is zero. Readings of the spring
balance should be taken only when the balance arm is quite horizontal, and for this purpose an adjustment is
provided.

A pointer is attached to a fixed part of the framework, which registers with a mark on the moving balance arm
when it is in the desired position. Another precaution that is sometimes overlooked is to make certain that no
operator has his or her hand upon the load-adjusting wheel or any other part of the swinging casing when a
reading is being taken.

THE ELECTRIC DYNAMOMETER

The true electric dynamometer, shown in Figure 4.7, is basically a generator which is driven by the power
source to be measured. The generator output is loaded-up electrically (the output voltage being coupled to a
resistance circuit). The torque applied to the generator is determined from the force required on the spring
balance to keep it in the position shown.

As the torque increases, the generator casing tends to rotate in its trunnion bearings. As shown, the device is a
fixed installation to which the input power source is coupled for the duration of the test only. Where torque
transmission needs to be monitored on a permanent basis, the output torque from an electric motor may be
measured with a system similar to that shown in Figure 4.7.
The electric motor is mounted on trunnion bearings and a ‘torque-arm’ affixed; once again the torque output is determined by the spring-balance reading multiplied by the distance to its line of action from the centre of rotation.

THE TRANSMISSION DYNAMOMETER

When a shaft is subjected to torque, it will twist to some extent, depending on the physical dimensions of the shaft, the material from which it is made and, of course, the applied torque. The actual relationship is:

\[
\frac{T}{I} = \frac{G\theta}{l} = \frac{f}{r}
\]

where

- \( T \) = applied torque (N m)
- \( \theta \) = angle of twist (radians)
- \( I \) = second moment of area of the shaft (m^4)
- \( G \) = modulus of rigidity of the shaft material (N/m^2)
- \( l \) = shaft length (m)
- \( f \) = maximum shear stress for the material (N/m^2)
- \( r \) = shaft radius (m)

It follows that if a shaft of known dimensions and material is transmitting a torque, and if the resulting angle of twist can be measured, the torque may be calculated.

Consider the case of an electric motor driving a pump. A portion of the drive shaft could be replaced with a specially-made shaft designed to give maximum twist within the permissible stress range of the material. In operation, the angle of twist that occurs is measured optically or electronically.

The optical method consists in having identical discs, calibrated in degrees, fixed to opposite ends of the shaft with the calibrations in phase when the shaft is unloaded. Under running conditions, the discs are viewed through the eyepiece of an optical system which, once in each revolution, briefly presents an image of the two discs super-imposed. Due to the stroboscopic effect, the discs appear stationary and the phase difference of the two ends of the shaft may be read off directly in degrees of twist of the shaft.

For electronic indication of twist angle, similar discs attached to the ends of the shaft are used. In place of the degree calibrations, these discs have a number of equi-spaced slots cut in their peripheries and arranged so that each slot in one disc is exactly in phase with its opposite number in the other disc when the shaft is unloaded. Each disc is monitored by a transducer, each rigidly fixed in a similar angular position. The transducers are commonly photo-transistors responding to individual light sources, located on the opposite side of each disc, the light-beam being modulated by the slots.

Under running conditions, and with appropriate circuitry, the transducers each produce an electrical impulse as each slot passes through the light beam. If the shaft is unloaded, each pair of impulses will be in phase but, as load is applied a phase difference proportional to shaft twist angle will arise. This phase difference is displayed by a phase meter from which angle of twist may be read off directly.

POWER MEASUREMENT

As shown below, if the output torque of a system can be measured and also its rotational speed, the output power of a system may be calculated.

Considering Figure 4.8:
Work done = force applied \times distance travelled \quad (1)

Distance travelled around the circle = 2\pi \times radius \quad (2)

Substituting 1 and 2 in the original expression:

\[
\text{Power} = \frac{\text{force applied} \times 2\pi \times \text{radius}}{\text{time taken}}
\]

But force applied \times radius = torque

\[
\text{Power} = \frac{2\pi \times \text{torque}}{\text{time taken}}
\]

\[
\text{Power} = 2\pi \times \text{rotational speed}
\]

This is usually written in symbol form as:

\[
\text{Power} = 2\pi N T
\]

where: \(N\) = speed (r/s) \quad \text{Note: NOT r/min}
\(T\) = torque (N m)

**EXAMPLE**

A rope brake type of dynamometer has an initial load of 10 N and a running load of 120 N. The brake drum is 250 mm diameter and rotates at 100 r/min. Calculate the power at the brake (or brake power).

Force due to rope around the drum = 120 – 10 = 110 N

\[
\text{Torque} = \text{force} \times \text{radius} \\
= 110 \times \left( \frac{250 \times 10^{-3}}{2} \right) \\
= (110 \times 0.125) \text{ N m}
\]
**Measurement of Mechanical Variables**

\[
\text{Power} = 2\pi NT \\
= 2\pi \times \frac{100}{60} \times (110 \times 0.125) \\
= 143.9 \text{ W}
\]

**NOTE:** The division of the speed by 60 to change the value from r/min to r/s.

**EXAMPLE**

An electronic torque measuring device indicates an angle of twist of 0.75° in a shaft 30 mm diameter and 500 mm long, rotating at 200 r/min. The Modulus of Rigidity for the shaft material is 80 GN/m². Calculate:

(a) The transmitted torque.
(b) The transmitted power.

(a) From the original formula:

\[
\frac{T}{I} = \frac{G\theta}{l} \\
T = \frac{G\theta I}{l}
\]

For a circular shaft \( I = \frac{\pi d^4}{32} \)

Also, because 360° = 2π radians

0.75° = \( \frac{2\pi \times 0.75}{360} \) radians

Substituting into the formula,

\[
T = \frac{G\theta I}{l} \\
= \frac{(80 \times 10^6)}{0.5} \times \frac{2\pi \times 0.75}{360} \times \frac{\pi \times 0.03^4}{32} \\
= 166.6 \text{ N m}
\]

(b) \( \text{Power} = 2\pi NT \)

\[
= 2\pi \times \frac{200}{60} \times 166.6 \\
= 3.488 \text{ kW}
\]

**4.2 PRESSURE MEASUREMENT**

Before considering the actual methods of measuring pressure, it is essential to make clear the meaning of pressure and units used, and the difference between gauge pressure and absolute pressure.

The ‘formula’ for pressure is:

\[
\text{Pressure} = \frac{\text{Force}}{\text{Area}}
\]

Thus, a pressure is simply a force applied over a specified area. Using a force in newtons and area in square metres:
Chapter Four

\[ \text{Pressure} = \frac{N}{m^2} = \text{N/m}^2 \]

The unit N/m\(^2\) is commonly given the name of a pascal (abbreviation Pa).

\[ \text{N/m}^2 = \text{Pa} \]

A pascal is a relatively small unit of pressure so multiples are normally used. For instance:

- typical car tyre pressure = 200 kPa
- typical oil hydraulic system pressure = 15 MPa

In thermodynamics (the study of heat and heat engines) a pressure unit called a bar is frequently used, where 1 bar = 100 kPa. You will have heard sub-multiples of a bar used in weather forecasting, where the barometric pressure used to be quoted in millibars. The current unit used for barometric pressure is the hectopascal, which is equal to a millibar. The bar unit has gained some popularity overseas because it is very close to one ‘atmosphere’:

\[ 1.013 \text{ bars} = 1 \text{ atmosphere} \]

**EXAMPLE**

Express 21 360 N/m\(^2\) in Pa, kPa, bar and MPa

\[ 21\ 360 \text{ N/m}^2 = 21\ 360 \text{ Pa} \]

\[ = 21.36 \text{ kPa} \]

\[ = 0.2136 \text{ bar} \]

\[ = 0.021\ 36 \text{ MPa} \]

Consider the terms *gauge* pressure and *absolute* pressure. Gauge pressure is that which is normally displayed on a pressure gauge. On such a gauge a reading of zero hides the fact that even at zero gauge pressure a pressure value does exist, the atmospheric pressure, nominally taken as 101.3 kPa.

The value of 101.3 kPa is itself an absolute pressure, being the pressure above absolute zero pressure or the situation where no pressure exists - just a space empty of gas or liquid molecules.

Figure 4.9 displays the two scales side by side.

**Fig. 4.9** Gauge Pressure versus Absolute Pressure
It appears from Figure 4.9 that a vacuum is negative and this is quite correct - vacuum gauges are calibrated in negative units. In fact, it makes more sense to talk about positive and negative (vacuum) pressures.

Thus:

Absolute pressure = gauge pressure + atmospheric pressure

where atmospheric pressure is normally taken as 101.3 kPa.

PRESSURE MEASUREMENT DEVICES

Manometers

Liquids in a system always seek a common level; thus, if some liquid is poured into an upright U-tube it will settle at a common level in the two branches of the tube, as in Figure 4.10.

If one leg of the U-tube is connected to a vessel or system and the other leg left open to atmosphere, the U-tube becomes a manometer, as in Figure 4.11.

Consider the case where the pressure in the vessel is below atmospheric pressure (left-hand diagram of Figure 4.11). The fluid in the vessel exerts a pressure on the liquid in the left-hand branch of the U-tube. The atmosphere exerts atmospheric pressure on the liquid in the right-hand branch of the U-tube and forces it down. Provided that the U-tube is long enough to prevent the liquid entering the vessel, the liquid column will come to rest as shown.

At the reference level, the pressure in the left-hand column is caused by the pressure in the vessel plus the pressure due to the column of liquid of height $h$. 
In the right-hand column, the pressure is a result of the atmospheric pressure. Now, pressure is transmitted uniformly throughout a liquid; therefore, the pressure at the reference level in both legs of the U-tube is the same:

Atmospheric Pressure = Vessel Pressure + Pressure due to a column of liquid of height $h$

thus

Vessel Pressure = Atmospheric Pressure - Pressure due to a column of liquid of height $h$

In a similar manner, the vessel pressure may be determined when the pressure is above atmospheric pressure.

At reference level, (right-hand diagram of Figure 4.11)

Vessel Pressure = Atmospheric Pressure + Pressure due to a column of liquid of height $h$

The pressure owing to the column of liquid is calculated from the formula:

$$ p = \rho gh $$

where:

- $p$ = pressure (N/m$^2$ or Pa)
- $\rho$ = mass/density of liquid (kg/m$^3$)
- $g$ = gravitational acceleration (normally 9.8 m/s$^2$)
- $h$ = height of column of liquid (m)

**EXAMPLE**

A manometer containing mercury has a height of 140 mm. Calculate the applied pressure. The mass density of mercury is 13.6 tonnes per cubic metre.

$$ p = \rho gh $$

$$ = (13.6 \times 10^3) \times 9.8 \times 0.14 $$

$$ = 18.66 \times 10^3 \text{ Pa} \text{ or } 18.66 \text{ kPa} $$

Note that this is the gauge pressure above (or below) atmospheric. To obtain an absolute pressure value we need to add 101.3 kPa.

Manometers can also be used to determine the difference in pressure between two systems by connecting one leg of the U-tube to each system, as in Figure 4.12. An example would be the measurement of pressure difference (or pressure drop) across a filter unit, to determine if the filter was becoming blocked.
As we are considering the pressure due to a column of liquid, a slight modification to the formula occurs when two liquids are used in the manometer. This occurs, for instance, when the system contains water - the manometer must contain a liquid that does not mix with water, such as mercury, as in Figure 4.13.

If we wish to determine the pressure at A (symbol $p_A$),

\[
\begin{align*}
\text{pressure at } X & = p_A + \rho_{1ghA} \\
\text{pressure at } Y & = \rho_{2gh}
\end{align*}
\]

But pressure at X and Y are equal, being on the same level.

So

\[
\begin{align*}
p_A + \rho_{1ghA} & = \rho_{2ghA} \\
p_A & = \rho_{2gh} - \rho_{1ghA}
\end{align*}
\]

Note that the atmospheric pressure has been disregarded and the value of $p_A$ will be a gauge pressure.

Although a manometer is a simple type of pressure-indicating device, its usable range is limited. At very low pressure, the conventional U-tube manometer is fairly difficult to read. For instance, if checking the performance of a fan or measuring the pressure inside an air duct, pressures below 50 Pa are common.
Converting 50 Pa to the height of water in a manometer

\[
p = \rho g h
\]

\[
h = \frac{p}{\rho g} = \frac{50}{1000 \times 9.8} = 5.1 \text{ mm}
\]

To accommodate such small manometer readings, the tube may be angled to extend the lengths, as shown in Figure 4.14 of an inclined manometer.

The inclined scale provides a magnification of about 20 times, depending on the angle used. This instrument may be used to measure a single pressure at A, the pressure difference between A and B, or the negative pressure or vacuum if connected to B only.

![Inclined Manometer](image)

Fig. 4.14 Inclined Manometer

Considering the higher pressure limitations of a U-tube manometer, if it were required to check the pressure in a vehicle tyre, say in the region of 200 kPa gauge pressure, we would need a water-filled manometer that could indicate a height of about 20 metres!

\[
p = \rho g h
\]

\[
h = \frac{p}{\rho g} = \frac{200 \times 10^3}{1000 \times 9.8} = 20.4 \text{ m}
\]

To obtain a reading of 20 metres of water would require a manometer 40 metres high! If mercury were used in the manometer instead of water, the value of \(h\) would be about 1.5 metres (mass density of mercury = 13.6 x 10\(^3\) kg/m\(^3\)) but even that height would make the instrument rather cumbersome.

Thus, for pressures above about 10 kPa, it is more convenient to use a Bourdon tube gauge, as shown in Figure 4.15.

The gauge consists of a curved tube which has an elliptical cross-section, made from bronze for lower pressures and steel for higher pressures (over 5 MPa). The open end of the tube is connected to a quadrant by a link. Pressure inside the elliptical tube tends to straighten out the curve. This causes the free, closed end to move every time the pressure changes. This small movement is magnified and turned into a rotary movement of the pointer by the quadrant and pinion mechanism. A scale mounted behind the pointer is calibrated in the appropriate units and the fluid pressure is read from the scale.
The Bourdon pressure gauge can be used for measuring negative pressure (vacuum) but the pointer moves in an anti-clockwise direction. Gauges are also available which measure pressure and vacuum on the same scale; these types are called compound gauges.

Australian Standard 1349 provides technical information on the construction and accuracy of Bourdon tube gauges. One important recommendation is that the "usable" range of a gauge should be within the middle 50% of its scale. Thus, a 1000 kPa range gauge should only be used for pressures between 250 kPa and 750 kPa if acceptable accuracy is required.

4.3 TEMPERATURE MEASUREMENT

On the Celsius scale, water freezes at 0°C and boils at 100°C and the range in between is divided into 100 equal parts, each part being known as one degree celsius [1°C].

All bodies expand or contract when they are subjected to a temperature change and use is made of this property in temperature-measuring instruments.

MERCURY IN GLASS THERMOMETER

This is the most common instrument used for temperature measurement. (Figure 4.16)

In most glass thermometers, mercury is used as the expanding medium.

It has a boiling point of 356°C and a freezing point of -38°C and can be used for measuring any temperature between these limits.

Normally, the space in the capillary above the mercury is a vacuum but if nitrogen is introduced at high pressure into the space, the higher pressure increases the boiling point of mercury and the temperature range of the thermometer is increased. The strength of the glass then puts the upper limit on the temperatures which can be measured by glass thermometers.
In addition to its good temperature range, mercury is used because it does not wet glass and it has a good expansion rate. Alcohol, usually dyed red, can be used to measure temperatures down to its freezing point of \(-113^\circ C\), but it has a low boiling point of \(78^\circ C\) and so alcohol thermometers are very limited in their use.

**FILLED-SYSTEM THERMOMETER**

Use can be made of expanding fluids to operate a Bourdon pressure gauge calibrated in temperature instead of pressure units. A suitable fluid is contained in an enclosed bulb which is connected to the pressure gauge by a fine capillary. A temperature change at the bulb causes the fluid to expand or contract and thus creates a pressure change which acts on the Bourdon pressure gauge, as in Figure 4.17.

The fluid may be a liquid or a gas. A popular version uses mercury in steel, which may be pressurised for temperature measurement up to \(600^\circ C\).

![Fig. 4.17 Filled System Thermometer](image)

The output scale is normally linear and the pointer output is sufficiently powerful enough to operate a pen recorder. Vapour-filled systems commonly use freon, alcohol or ether, which partly fills the system as a liquid and the remainder is vapour-filled. It has a temperature range from \(-50^\circ C\) to \(260^\circ C\) but a non-linear output scale.

Gas-filled thermometers usually employ nitrogen or helium under high pressure, the pressure being proportional to the absolute temperature at constant volume. The usual temperature range is \(-50^\circ C\) to \(430^\circ C\) and the output scale is linear.

**BIMETALLIC THERMOMETER**

The construction of a bimetallic thermometer is shown in Figure 4.18.

The principle of the bimetallic thermometer involves two strips of metal bonded together. The metals are of widely differing coefficients of linear expansion, such as Invar (a special low expansion alloy) and brass. As the composite strip is heated, the differing expansion rates cause the material to deflect. By using a coil of bimetallic material even small changes in temperature cause an effective change in the bimetallic coil position.
PYROMETERS

Pyrometry is the name given to the study and measurement of high temperatures. There are three types of pyrometer in general use:

1. Resistance pyrometers for temperatures ranging from $-120^\circ C$ to $1000^\circ C$.

2. Thermo-electric pyrometers for temperatures up to $1600^\circ C$.

3. Radiation and optical pyrometers for temperatures up to $1350^\circ C$.

Resistances Pyrometer

This instrument is based on the fact that the electrical resistance of a metal increases as the temperature increases.

The resistance pyrometer consists of a coil of wire suitably wound, insulated and protected. For low temperatures up to $100^\circ C$, silk-covered nickel wire is varnished and enclosed in a sheath. For temperatures up to $1000^\circ C$, platinum wire is used. Measurements are made by passing a standard voltage across the coil and measuring the current flow directly in terms of relative temperature. For more precise measurements the resistance element is placed across one leg of a Wheatstone Bridge, as shown in Figure 4.19.

Fig. 4.18 Bimetallic Thermometer

Fig. 4.19 Resistance Pyrometer
Resistors $R_2$ and $R_3$ have the same value, so when the galvanometer is zeroed by adjusting $R_1$, the value of $R_1$ equates to the value of the pyrometer resistance, which in turn relates to the temperature.

In practice, the Wheatstone Bridge may be balanced automatically and a direct temperature reading obtained.

**Thermoelectric Pyrometer (Thermocouple)**

The principle on which this instrument is based was discovered by Seebeck in 1821. He found that if two dissimilar materials were joined at their ends, and one of the junctions is heated, an electric current will flow through the circuit. The value of the electro-motive force (e.m.f) generated depends upon the materials used and the temperature difference between the hot and cold junctions. Extremely precise and convenient readings up to 1650°C can be readily obtained and may either be indicated, recorded or used as a control function.

The couple consists of two wires of the chosen materials, welded together at one end, insulated and often enclosed in a protective sheath. In many cases the wires are connected directly to a millivoltmeter, but where precise temperature measurements are required the system is as in Figure 4.20.

The hot junction is placed in the region to be measured and the cold junctions are maintained at a pre-set temperature, say freezing point, so that performance of the unit is predictable.

![Fig. 4.20 Thermocouple](image)

For temperatures up to 250°C, wires of copper (+) and Constantan (-) may be used. Constantan is an alloy of 40% nickel and 60% copper. For temperatures up to 850°C wires of iron (+) and Constantan (-) may be used. For very high temperatures, such as molten steel up to 1650°C, wires of platinum and a platinum/13% rhodium alloy may be used.

Surface Pyrometers have been designed to take the temperatures of heated surfaces under working conditions in rubber, paper and textile industries. Each instrument consists of a thermocouple in the form of a thin flat strip with the junction at its mid-point, stretched across the ends of a bow spring and connected to a millivoltmeter. The thermocouple is pressed into contact with the hot surface and the reading taken. The instrument can be applied to moving surfaces with no risk of damage by scratching.

**Optical Pyrometer**

Various types of this instrument exist but the most common is the "disappearing filament" type. As shown in Figure 4.21, it consists of a telescope inside which is a small electric lamp. The telescope is focused on the region to be measured and then the current through the lamp filament is adjusted until the colour of the filament merges with the colour of the region being assessed for temperature.

The ammeter is calibrated directly in temperature units.

The optical pyrometer has the advantage that temperatures may be measured without actually touching the hot region, such as inside a furnace, but care must be taken that smoke or fumes do not obstruct the light received at the pyrometer.
Measurement of Mechanical Variables

Radiation Pyrometer
In this type of pyrometer, radiant energy from a hot body is focused onto a thermocouple which is connected to a millivoltmeter, which is calibrated directly in temperature units. (See Figure 4.22.) As with the optical pyrometer, care must be taken in use to ensure that the path of the radiation is not obstructed.

<table>
<thead>
<tr>
<th>BODY COLOUR</th>
<th>APPROXIMATE TEMP (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dazzling</td>
<td>1500</td>
</tr>
<tr>
<td>White</td>
<td>1300</td>
</tr>
<tr>
<td>Yellow</td>
<td>1100</td>
</tr>
<tr>
<td>Orange</td>
<td>1000</td>
</tr>
<tr>
<td>Bright Red</td>
<td>900</td>
</tr>
<tr>
<td>Cherry Red</td>
<td>800</td>
</tr>
<tr>
<td>Dark Cherry Red</td>
<td>700</td>
</tr>
<tr>
<td>Dark Red</td>
<td>600</td>
</tr>
<tr>
<td>Red (just visible)</td>
<td>500</td>
</tr>
</tbody>
</table>

Fig. 4.21 Optical Pyrometer

Fig. 4.22 Radiation Pyrometer

The following table indicates the relationship between the radiant source colour and temperature.
**Fusion Pyrometer**

Chemical substances have various melting points, and this property may be used for temperature indication. Mixtures of clay, metals and salts are made up into slender cones, about 75 mm high called *Seger Cones*. By varying the composition, the collapsing temperature of the cone may be predicted.

Thus, to check when a furnace has reached 1000°C, a series of cones rated at 800°C, 900°C and 1000°C would be inserted. As the last cone "collapsed", the furnace would be at the correct temperature. The cones cannot be re-used.

**Thermal Crayons**

Thermal crayons and paints are available which may be applied to a structure to assess temperature, the change in colour of the marking being related to a known scale. This method is useful for large structures that require localised heat-treatment, perhaps before welding (or after welding to relieve stresses).

### 4.4 ROTATIONAL SPEED

A revolution counter is the most basic method of measuring rotational speed. The device shown in Figure 4.23 is held onto the end of a rotating shaft and the number of revolutions measured during a certain time, normally a minute.

![Fig. 4.23 Rotation Counter](image)

One disadvantage of this method is that it involves the use of the revolution counter and a stop watch, so inaccuracies may occur. Also, the final result provides the average rotational speed and would not detect any surges or variations in speed during the time period.

A *tachometer* gives a direct indication of rotational speed and may be operated mechanically or electronically. Mechanical versions normally use the effect of centrifugal forces acting against a spring - as rotational speed increases, the spring is increasingly compressed and its resultant position is linked to output display.

Another type of mechanical tachometer uses the viscous drag of one cup rotating inside another, separated by an air gap. The second cup is restrained by a hair spring and its partial rotation is displayed on a scale, in units of r/min.

Mechanical tachometers, especially where physical contact is involved, are limited to a speed range of up to 5000 r/min, and, of course, are subject to mechanical wear. Also, the output reading must be taken at the rotating shaft, which may be in a dark or dangerous location.

*Electrical/Electronic tachometers* have the advantage that the readout may be located at some distance from the measurement point. Furthermore, electronic tachometers are available which will measure the rotational speed without actually needing to touch the rotating shaft and operate in excess of 40 000 r/min.
When a direct drive is acceptable, a small alternator is driven by the shaft to be measured. The output voltage is rectified and read directly on a voltmeter - the displayed voltage being proportional to speed, as shown in Figure 4.24. An accuracy of ±0.25% may be achieved.

An alternative direct-drive electric tachometer is the *eddy-current* tachometer as shown in Figure 4.25. It is widely used for vehicle speedometers. A rotating magnet is close to an aluminium disc (or cup), the disc being directly coupled to an output pointer and restrained by a hair-spring. The eddy-currents induced into the disc by the rotating magnet set up a magnetic field which resists the magnetic field of the magnet. The rotational forces produced (torque) cause the disc to rotate relative to magnet speed.

This unit can provide an accuracy of ±10 r/min when operating at 3000 r/min.

Non-contacting tachometers are used when the shaft to be measured must not be subjected to additional torsional load, or a drive from the end of a shaft is not possible.

The *inductive pickup tachometer* uses the pulses produced from a pickup adjacent to the rotating shaft, as shown in Figure 4.26. A variation of this principle utilises a small magnet fixed to the rotating shaft and provides measurable pulses.
The light-operated tachometer utilises a beam of light that is reflected from a rotating shaft onto a phototransistor. One half of the periphery of the shaft may be treated with matt-black paint to make it non-reflecting or alternatively, an area of a black shaft may be covered with a white reflective tape. Thus, for every revolution, the photo-transistor will receive one burst of reflected light which, in turn, switches a transistor and the speed is computed and displayed on a meter or a digital readout.

An electronic stroboscope may be used to measure rotational speed.

A stroboscope consists of a gas discharge tube that emits a bright flash when "fired" (similar to the timing gun used with vehicles). The flash is controlled by an adjustable oscillator, and the flash rate may be preset in terms of frequency (Hz) or cycles per second.

In use, some distinguishing mark is made on the shaft. The stroboscope is adjusted so that the flashing frequency is higher than the rotational frequency of the shaft. If the stroboscope light is then directed upon the rotating shaft, the eye of the observer will register the image of the marked shaft only while the latter is illuminated by the high-intensity flashes. Since flashes are occurring more than once per shaft revolution, and owing to the observer's persistence of vision, the marked shaft will appear to be rotating slowly backwards.

The frequency of the stroboscope flashes is then reduced until the shaft appears stationary and the single mark made upon it is clearly visible. The flash frequency is then read off from the calibrated control on the stroboscope. This is also the rotational speed of the shaft.

A certain degree of skill is required to use a stroboscope in this way successfully for it will be appreciated that if, for example, the flash frequency is set to twice the rotational frequency, the shaft will again appear to be stationary but will apparently display two marks instead of one.

Conversely, if the flash frequency is set to, say, one-third that of the shaft, the latter will be illuminated, when in the same angular position, once in every three revolutions. The shaft will again appear stationary and, this time, only the single mark will be visible.

The only indications of the difference in frequency between flash and shaft rotation will be the generally lower level of illumination and consequent lack of visual definition. This effect may be slight, however, and the best technique is to start operations with a higher flash frequency, reducing this gradually until the first image of the shaft with one mark appears.

The electronic stroboscope is unsuitable for measuring low rotational speed but its short-duration, high-intensity flash can be used to “freeze” the motion of any visible machine mechanism which is operating at a constant rate.
Examples of stroboscopic "freezing" are:

- valve gear to show valve bounce;
- pump mechanisms to indicate impeller movements;
- water flow to display the passage of air bubbles.
FUELS

5.1 CALORIFIC VALUE OF FUELS

Fuels may be defined as a composition of carbon and hydrogen, because these elements combine readily with oxygen under suitable conditions and give out appreciable amounts of heat.

The heating or calorific value of these fuels is a measure of the heat evolved by burning 1 kg of the fuel in an adequate supply of oxygen (air) and for the two elements mentioned the approximate calorific values are:

- Hydrogen  144 MJ/kg
- Carbon  34 MJ/kg

Fuels may be broadly classified into solid, liquid and gaseous.

5.2 SOLID FUELS

Solid fuels in the first place absorbed their energy from the sun, in the form of vegetation.

**Wood**

Wood consists largely of carbon and hydrogen chemically formed by the action of sunlight, and was at one time used extensively as a fuel. Other solid fuels are derived from partial decomposition of wood and vegetable matter together with the compressing action of superimposed layers as the original wood was buried under earth and rock. The form of the fuel indicates the relative time for which these processes have continued.

**Peat**

This represents the first stage at which the fuel derived from wood and vegetable matter is recovered from the earth. It is a fibrous watery substance found close to the surface and needs to be dried off before it will burn satisfactorily.

**Lignite or Brown Coal**

This is the next stage, and is an inferior type of coal containing less than 70% carbon and a much higher proportion of moisture than true coal. It has been used extensively as a low grade fuel. Calorific value is about 23 MJ/kg after drying.

**Bituminous Coal**

The next stage in the development, and is the main fuel mined. It is shiny black in appearance, often showing signs of its vegetable origin, and consists of about 80% carbon and 3% to 5% hydrogen. Calorific value is about 31.5 MJ/kg.

**Anthracite**

It is regarded as the final stage in the development. It is hard and brittle and consists of about 90% carbon and 3% to 4% hydrogen. It is difficult to ignite but burns with little smoke. Calorific value is about 35 MJ/kg.

When coal is heated it breaks up into two distinct fuels:

(a) Volatile hydrocarbons (compounds of carbon and hydrogen of the form C\(_n\)H\(_m\)) which ignite easily and burn with a long yellow flame.

(b) Fixed carbon, which burns with hardly any flame and is difficult to ignite.
Fuels

Bituminous coal when heated gives off from 20% to 30% volatiles, whereas anthracite contains very few; thus bituminous coal is much easier to ignite than anthracite coal.

The distillation temperature of the volatiles (the temperature at which they leave the coal), is much below the ignition temperature of the fixed carbon, so that it is important to ensure that after distillation they pass through a zone of high temperature and burn; otherwise they pass away as smoke and soot with a consequent wastage of heat.

Coke

Coke is the substance, consisting almost entirely of fixed carbon and ash, which remains after the greater part of the hydrocarbons and moisture has been distilled from the coal, as in the production of coal gas. Calorific value is about 28 MJ/kg.

Coke is manufactured by enclosing up to 20 tonnes of coal in a large box-like oven (10 m long by 3 m high and 0.6 m wide) and heating at 1000°C. As the coal is heated, tar, ammonia compounds, naphthalene, benzol, toluene and gas are driven off. Up to 40% of the gas produced may be recirculated to heat the oven.

A series of ovens is used, each being sequentially charged with coal, heated for 12-24 hours, then discharged and the coke quenched in water. Coke used for the production of steel is called metallurgical coke and is made from low-sulphur coal.

Charcoal

Charcoal (or char) is a porous material containing 85-98% carbon, produced by heating carbonaceous materials such as cellulose, wood, peat and certain coals at 500-600°C in the absence of air. For charcoal that is to be used as a filter media, heating may be done in carbon dioxide or steam atmosphere at 900°C for a shorter period. Charcoal used for filtering can have a surface area as high as 1000 square metres per gram.

Determination of the Calorific Value

The calorific value of a fuel can only be satisfactorily determined by experiment, and in the case of solid fuels a bomb calorimeter is used. While a number of different makes of bomb calorimeter are on the market, all of them consist essentially of the same parts, as shown in Fig. 5.1, namely,

1. A strong container called a "bomb", in which a measured quantity of fuel may be fired in oxygen at high pressure.

2. An electrical arrangement whereby insulated leads are carried to the fuel container inside the bomb, from which a fuse wire dips into the fuel. A switching device is available so that the current may be switched on at a given instant, causing the fuse wire to glow and ignite the fuel.

3. A well-insulated calorimeter containing a known amount of water into which the bomb may be immersed so as to absorb the heat given out from the fuel. Arrangements must be made for uniform stirring of the water so that the heat is well distributed.

4. A sensitive thermometer which may be well immersed into the water in the calorimeter and give an accurate measurement of the change in water temperature during the experiment.

In operation, a small amount of dry powdered fuel is placed in the bomb calorimeter and a thin wire is attached to act as an igniter. The bomb is filled with oxygen and the fuel is ignited by passing a current through the thin wire.
A series of water-temperature readings are taken. The calorific value of the fuel may now be calculated from the general equation:

\[
\text{Heat released from the fuel} = \text{Stored energy gained by water and calorimeter} \\
\text{mass of } \times \text{ calorific fuel value} = (\text{mass of water} + \text{water equivalent of calorimeter}) \\
\times \text{ specific heat capacity } \times \text{ rise in temperature of water}
\]

The water equivalent of the calorimeter is established by first using a fuel of known calorific value.

The calorific value determined during the experiment is the gross (or higher) calorific value since it includes the latent heat of the steam formed by the combustion of the hydrogen present as it cooled at the temperature of the surrounding water. The net (or lower) calorific value can be calculated, providing the mass of hydrogen in the fuel is known, from the formula:

\[
\text{Net c.v.} = \text{Gross c.v.} - (9H_2 \times 2.453) \text{ MJ}
\]

where \( H_2 \) is the mass of hydrogen per kg of fuel.

### 5.3 LIQUID FUELS

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Most of the liquid fuels used are derived from one or the other of the following sources:

1. Crude petroleum.
2. By-products from coal gas manufacture.
3. Vegetable matter.
1. CRUDE PETROLEUM

The historical origin of crude petroleum is in some doubt, but it is found in natural oil reserves situated in the earth's crust and from which it is pumped. Crude petroleum is the basis from which the great majority of oil fuels are obtained; the separation of this crude petroleum into the numerous oils used in industry is achieved by distillation (by heating the crude petroleum and condensing the vapour which evaporates) at various temperatures and pressures.

Below 65°C: All those ‘fractions’ which distil at temperatures in this range at atmospheric pressure are removed first and are too volatile for use in engine cylinders or heating apparatus.

65°C - 220°C: Those fractions which boil between these limits form the petros of industry. Their chemical composition varies widely as does their specific gravity, volatility and ignition temperature.

220°C - 345°C: Fractions distilled between these limits form the heating and lighting fuels, such as paraffin and kerosene.

345°C - 425°C: The fuel oils used in compression-ignition engines and in the furnaces of oil-fired boilers are distilled in this range. The products which now remain if distilled under a vacuum give mineral lubricating oils.

Over 425°C: Finally, such products as greases, paraffin wax, bitumen or asphalt are obtained at very high boiling temperatures, depending upon the type of crude oil used.

Cracking Distillation is a process whereby the yield of the lighter and more valuable fractions may be increased. The heavier fractions are heated under high pressure in the absence of air, so that the heavy hydrocarbon molecules are decomposed into simple molecules.

2. BY-PRODUCTS FROM COAL-GAS MANUFACTURE

Tar is an important by-product from the manufacture of coal gas, and it may be re-distilled to produce valuable fuels like Benzene (C₆H₆) and Toluene (C₇H₈). The important feature of such oils is that they are much less liable to detonation than standard petros and form a good alternative to petrol in internal combustion (I.C.) engines.

3. VEGETABLE MATTER

Alcohol is formed by fermentation of vegetable matter, and has been widely used in some parts of the world as a commercial fuel. Since it has a high ignition temperature, arrangements have to be made for easy starting when used in I.C. engines, but it has the advantage that detonation is less likely so that higher compression ratios may be used and hence higher thermal efficiencies obtained. An important feature is that supply is practically unlimited, which cannot be said for natural crude petroleum deposits.

PROPERTIES OF LIQUID FUELS

Among the properties of liquid fuels, those which concern us are calorific value, flash point, ignition temperature, octane number and cetane number.

Calorific Value of Liquid Fuels

This may be obtained experimentally in the same way as for solid fuels using the bomb calorimeter, except that to avoid shattering the crucible when using liquid fuels the fuse wire should not actually dip into the fuel but should be connected to it by a piece of cotton. In this case, allowance is made for the heating value of the cotton by weighing the piece used, given the calorific value of cellulose = 173 MJ/kg.
Some approximate values of calorific values (gross) for the fuels mentioned are:

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Petrol</td>
<td>46.5 MJ/kg</td>
<td>Kerosene</td>
<td>44 MJ/kg</td>
</tr>
<tr>
<td>Alcohol</td>
<td>30 MJ/kg</td>
<td>Diesel Oil</td>
<td>44 MJ/kg</td>
</tr>
<tr>
<td>Benzol</td>
<td>40 MJ/kg</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Flash Points**

This is the temperature at which inflammable vapour is given off by a fuel, and is important from storage considerations.

The Flash Point of a fuel is normally determined using the Pensky-Martens apparatus, as shown in Figure 5.2.

![Pensky-Martens Apparatus](image)

**Fig. 5.2 Pensky-Martens Apparatus**

To perform a flash point test, the cup is filled with the liquid to be tested, up to a set mark. Normally the **closed** flash point is determined: this means that the ignition temperature of an enclosed fuel is to be found, so a top is placed on the cup.

A small gas flame is positioned over the top of a sliding shutter, near the top of the cup. The fuel is then heated slowly (at 5-6 degrees celsius per minute) and stirred at 60 revolutions per minute. For each degree rise in temperature (or each three degrees rise above 100°C), the test flame is offered to the top of the fuel. This is done by operating the hand switch, which opens the sliding shutter and lowers the flame into the fuel vapour. When a distinct flash is observed, the temperature is noted.

If the cover is removed and the temperature increase continued, then the **open flash point** may be determined, which is 10-20 degrees higher than the closed flash point.
Above this temperature, by about 15-30 degrees, is the fire test point, the temperature at which the vapour generated is sufficient to support a continuous flame.

Typical flash points are:

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Petrol</td>
<td>below 0°C</td>
</tr>
<tr>
<td>Kerosene</td>
<td>46°C</td>
</tr>
<tr>
<td>Diesel distillate</td>
<td>75°C</td>
</tr>
</tbody>
</table>

**Ignition Temperature**

This is the lowest temperature at which the fuel will ignite spontaneously, that is, without the help of a flame or spark.

**Octane Number**

Some fuels have a greater tendency to detonation than others, depending upon their ignition temperature and the rate of the combustion reaction. A measure of their tendency to resist detonation is termed the octane number of the fuel.

In the determination of the octane number of a fuel, the fuel "iso-octane" is considered to be 100% anti-detonating and the fuel "heptane", which makes an engine detonate very readily, is considered to be 100% detonating.

The octane number of a fuel is measured by that percentage of octane in an octane-heptane mixture which detonates under the same conditions as the fuel under test.

A petrol engine with an adjustable cylinder head, so that the compression ratio can be altered, and fitted with a "bouncing-pin" device, which shows the onset of detonation, is used for the test. Running first on the fuel under test, the cylinder head is screwed down and the compression ratio thereby increased until the bouncing-pin device shows detonation to be just beginning.

Using this same head setting, the engine is run first on iso-octane and then an increasing percentage of heptane is added to the mixture until detonation occurs again. The percentage of iso-octane in the mixture when detonation recurs is termed the octane number of the fuel.

Certain dopes, in particular tetra-ethyl of lead, may be added to a fuel and have been found to increase the resistance to detonation, and thereby improve the octane number of the fuel.

**Cetane Number**

This is a measure of the suitability of oil fuels for use in compression-ignition engines. In the operation of high-speed diesels there is always a period of ignition lag between the beginning of fuel injection and the instant when ignition starts, resulting in a severe rise in pressure, and causing the characteristic sound of such engines known as "diesel knock".

It is desirable that this delay period should be as short as possible to avoid excessive pressure rise. The period of ignition lag can be measured using an accurate indicator. The fuel to be tested is used under standard conditions in the engine, and the period of ignition lag is measured in degrees of crank angle. A standard fuel consisting of cetane which has a very low period of lag and C_{10}H_{7}CH_{3} with a large period of lag is then used in the fuel supply line.

Starting with 100% cetane, the proportion of C_{10}H_{7}CH_{3} is increased until the same period of ignition lag is obtained as that of the fuel under test. The percentage cetane in the standard mixture gives the cetane number.
for the fuel being tested. Thus, the higher the cetane number, the more suitable is the fuel for compression-ignition work.

5.4 GASEOUS FUELS

Fuels under this heading include town gas, producer gas and natural gas.

**TOWN GAS**

Town gas is produced from coal, and was used to run gas engines as well as heating systems until mainly superseded in Western Australia by the abundant supply of natural gas. It is obtained from the distillation of bituminous coal, and has as its main constituents hydrogen (H\textsubscript{2}) and methane (CH\textsubscript{4}). Other gases, in particular water gas and blast furnace gas and more recently natural gas, are often mixed with coal gas to form town gas.

Water gas is obtained by blowing steam over a bed of burning fuel so that the steam breaks down into its constituents hydrogen and oxygen, the oxygen combining with carbon to form carbon monoxide.

Blast furnace gas is a by-product of the smelting process of iron ore.

A typical analysis of a town gas is as follows:

<table>
<thead>
<tr>
<th>Constituent</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydrogen (H\textsubscript{2})</td>
<td>53.0%</td>
</tr>
<tr>
<td>Methane (CH\textsubscript{4})</td>
<td>23.0%</td>
</tr>
<tr>
<td>Carbon Monoxide (CO)</td>
<td>12.5%</td>
</tr>
<tr>
<td>Nitrogen (N\textsubscript{2})</td>
<td>6.3%</td>
</tr>
<tr>
<td>Carbon Dioxide (CO\textsubscript{2})</td>
<td>2.8%</td>
</tr>
<tr>
<td>Oxygen (O\textsubscript{2})</td>
<td>0.4%</td>
</tr>
<tr>
<td>Hydrocarbons</td>
<td>2.0%</td>
</tr>
</tbody>
</table>

Gross calorific value (c.v.) of town gas is about 18.5 MJ/m\textsuperscript{3} at normal temperature and pressure (n.t.p.).

**PRODUCER GAS**

Producer Gas is used in gas engines, and is formed by the partial combustion of anthracite in a generator through which air is drawn by the intake suction of an engine. A fire is created at the base of the generator and as the carbon dioxide produced rises through the bed of burning anthracite or charcoal, it is dissociated (split up) into carbon monoxide and oxygen. The oxygen combines with carbon to make more carbon monoxide.

The main constituents of producer gas are carbon monoxide, hydrogen and nitrogen. The gross calorific value is low, in the order of 4.8 MJ/m\textsuperscript{3}

**NATURAL GAS**

This gas is found in porous rocks in oil-bearing regions in many parts of the world and has become a very valuable source of heat for industrial and town use. It is particularly important in Western Australia where substantial reserves have been discovered in the North West, to be used not only for domestic consumption but to be liquefied and shipped overseas.

**CALORIFIC VALUE OF GASEOUS FUELS**

Calorific value of a gas may be determined in a Boy's gas calorimeter, as shown in Fig. 5.3.

The gas to be tested is supplied through a sensitive meter and is pressure-regulated to give a constant supply of gas to the burners at the base of the calorimeter. The hot gases from the burners pass over a set of cooling tubes through which a controlled flow of water is passing and give up their heat to the cooling water before passing off at the top of the calorimeter.
The flow of cooling water is regulated by a constant head weir and its rise of temperature during its passage through the calorimeter is measured by sensitive thermometers. A small outlet pipe is fitted at the base of the calorimeter from which the condensed steam formed from the combustion of the hydrogen in the gas may be collected.

The calorific value of the gas is given by the general equation:

\[
\text{Heat released from the gas} \quad = \quad \text{Stored energy gained by the gas} \\
\text{Volume of gas burned} \quad (J/m^3 \text{ at n.t.p.}) \quad \times \quad \text{Calorific value} \quad = \quad \text{Mass of water passing through the cooling tubes} \times \text{rise in temperature} \times \text{specific heat capacity}
\]

It is essential when stating the calorific value of a gaseous fuel to state also the conditions of pressure and temperature to which it refers, for if the pressure and temperature were to be different from those specified then the volume of gas referred to would also be different.

It is usual to state the calorific value of town gas in joules per cubic metre at n.t.p. (that is, 101.3 kN/m\(^2\) and 15°C). In the test therefore, it is necessary to measure the temperature and pressure at which the gas is actually supplied so that the measured volume can be corrected to n.t.p. conditions. Readings taken during the test are:

- Gas supply temperature \(T_1\)
- Barometric pressure (mm of Hg) \(h_a\)
- Inlet gas pressure to meter (mm of H\(_2\)O) \(h_W\)
- Mass of water in measuring vessel \(m_W\)
- Mass of condensate collected during the test \(m_c\)
- Inlet and outlet water temperatures
The calorific value of the gas can then be calculated as follows:

1. **Temperature difference of water** $t$ equals mean outlet temperature minus mean inlet temp.

2. **Absolute pressure of gas supplied** $p_1$

   $$h_u = \frac{h_w}{13.6} \text{ mm Hg}$$

   Volume of gas supplied under test conditions = $V_1$

   Temperature of gas supplied = $T_1$

   Volume of gas supplied converted to n.t.p. conditions $V_n$ can be calculated from:

   $$\frac{p_n V_n}{T_n} = \frac{p_1 V_1}{T_1}$$

   where $p_n = 101.3 \text{ kN/m}^2$ and $T_n = 288 \text{ K}$

3. **Heat released by gas** = Stored energy gained by water.

   So

   $$V_n \times (\text{gross calorific value [J/m}^3\text{] at n.t.p.}) = m_W [\text{kg}] \times t_d [\text{deg C}] \times 4187$$

4. Since the condensate ($m_c$) collected is the condensed steam resulting from the combustion of the hydrogen in gas, the lower calorific value of the gas can be calculated from:

   $$\text{Net c.v. = Gross c.v. - } m_c \times 2.453 \times 10^6 \text{ J/m}^3 \text{ at n.t.p.}$$

**EXAMPLE**

In an experiment to determine the calorific value of town gas using a Boy's calorimeter, the following results were obtained:

- **Mean inlet water temperature** = 20.72°C
- **Mean outlet water temperature** = 39.36°C
- **Mass of water in measuring vessel** = 2130 g
- **Mass of condensate collected** = 4.93 g (in 10 L of gas)
- **Volume of gas metered** = 10 L
- **Barometer reading** = 740 mm Hg
- **Gas supply pressure** = 15 mm water
- **Gas supply temperature** = 20°C

Determine the values of Gross c.v. and Net c.v. for the gas per cubic metre at n.t.p.

**Temperature difference of water** = 39.36 - 20.72 = 18.64°C

**Volume of gas converted to n.t.p. conditions**:

$$V_n = \frac{p_1 V_1 T_n}{T_1 p_n}$$

$$p_1 = 740 + \frac{15}{13.6} = 741.1 \text{ mm Hg}$$

$$V_n = \frac{741.1 \times 10 \times 288}{760 \times 293} = 9.583 \text{ L}$$
Moisture collected = \( \frac{4.93 \times 100}{1000} \) = 0.493 kg/m\(^3\) of gas

Heat released by gas = Stored energy gained by water

\[ \frac{9.583}{10^3} \times \text{gross c.v.} = 2.13 \times 18.64 \times 4187 \]

\[ \text{gross c.v.} = 17.35 \text{ MJ/m}^3 \text{ at n.t.p.} \]

\[ \text{net c.v.} = \text{gross c.v.} - 0.493 \times 2.453 \]

\[ = 16.14 \text{ MJ/m}^3 \text{ at n.t.p.} \]

NOTE:
In this experiment, the town gas referred to consisted primarily of coal gas. An average value of calorific value of natural gas made available for domestic use is about 38 MJ/m\(^3\) at n.t.p.

5.5 TO CALCULATE THE GROSS CALORIFIC VALUE OF ANY FUEL

The only satisfactory way of determining the calorific value of a fuel is the experimental method described using a bomb calorimeter. An approximate value can, however, be obtained from a knowledge of the chemical analysis of the fuel, as follows:

Calculate the gross (or higher) calorific value of an oil fuel having the following composition by mass.
Carbon 85.5%, hydrogen 12.5%, oxygen 2.0%. Take c.v. of hydrogen as 144 MJ/kg and of carbon as 33.7 MJ/kg.

We first assume that any oxygen present in the fuel is already combined with some of the hydrogen, so that the heat release from this part of the hydrogen is not available.

1 kg of H\(_2\) requires 8 kg of O\(_2\) and produces 9 kg H\(_2\)O

0.020 kg of O\(_2\) will have combined with \( \frac{0.02}{8} \) kg of H\(_2\) = 0.0025 kg

Hydrogen available for heat release = 0.125 - 0.0025 = 0.1225 kg

Calorific value of hydrogen = 144 MJ/kg

So Heat release from 0.1225 kg of H\(_2\) = 0.1225 x 144 = 17.64 MJ

Calorific value of carbon = 33.7 MJ/kg

So Heat release from 0.855 kg of C = 0.855 x 33.7 = 28.81 MJ

Gross c.v. per kg fuel = 17.64 + 28.81 = 46.45 MJ

5.6 COMBUSTION OF FUELS

It is easy to cause fuels to burn - every smoker achieves this many times a day, but it takes a good deal of "know-how" to burn fuel efficiently, with minimum wastage. There are two good reasons why every self-respecting engineer should ensure that fuel is burnt efficiently:

1. Fuel is expensive, and inefficient combustion means expensive wastage.
2. Inefficient combustion results in pollution of the atmosphere with noxious gases.
The clue to the correct burning of fuels lies in a knowledge of the combustion equations which show how fuels combine with oxygen to release heat. We shall need to refer to molecular weight of a substance, and so the values are shown again in the following table:

<table>
<thead>
<tr>
<th>Element</th>
<th>Symbol</th>
<th>Atomic Weight</th>
<th>Molecular Weight</th>
<th>Calorific Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>C</td>
<td>12</td>
<td>12</td>
<td>33.7 MJ/kg</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>H₂</td>
<td>1</td>
<td>2</td>
<td>114</td>
</tr>
<tr>
<td>Oxygen</td>
<td>O₂</td>
<td>16</td>
<td>32</td>
<td>-</td>
</tr>
<tr>
<td>Sulphur</td>
<td>S</td>
<td>32</td>
<td>32</td>
<td>9.3</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>N₂</td>
<td>14</td>
<td>28</td>
<td>-</td>
</tr>
</tbody>
</table>

When a molecule is made up of atoms of different elements, the molecular weight can be calculated by adding the atomic weights of the elements concerned. For example:

1 molecule of carbon dioxide (CO₂) consists of 1 atom of carbon + 2 atoms of oxygen.

So: Molecular weight of CO₂ = 12 + (2 x 16) = 44

Similarly:

- Molecular wt. of carbon monoxide (CO) = 28
- Methane (CH₄) = 16
- Sulphur dioxide (SO₂) = 64
- Steam (H₂O) = 18
- Petrol (C₇H₁₆) = 100

The Burning of Carbon to Carbon Dioxide

This occurs when carbon is burnt with a sufficient supply of oxygen:

\[ C + O₂ = CO₂ \]

This equation states that 1 molecule of carbon combines with 1 molecule of oxygen and produces 1 molecule of carbon dioxide. Inserting the molecular weights, the equation becomes:

\[ 12 \text{ units by mass of Carbon} + 32 \text{ units by mass of Oxygen} = 44 \text{ units by mass of Carbon Dioxide} \]

\[ 1 \text{ kg of Carbon} + 8/3 \text{ kg of } O₂ = 11/3 \text{ kg of } CO₂ \]

Experiments with the bomb calorimeter shows that the calorific value of carbon is about 33.7 MJ/kg, so that:

1 kg of carbon needs 8/3 kg of O₂ and produces 11/3 kg of CO₂ (and releases 33.7 MJ)
PISTON ENGINES

6.1 ENGINE TERMINOLOGY

A large number of terms are used in relation to internal combustion engines and it is important to clarify these terms before dealing with specific types of engine.

*Power* of an engine is its rating in kilowatts (power) or kilojoules per second (energy per second). The term power is used as a generalisation but more specific forms of power occur.

*Brake Power* is the actual output power of an engine, measured at the output shaft as the name implies with some form of braking or loading device, a dynamometer. Thus brake power could be also called the usable power.

*Indicated Power* is the theoretical power of an engine. It may be calculated from the cylinder bore, stroke, speed and pressure as shown:

\[
\text{Indicated Power} = \frac{p_m \times L \times A \times N}{1000}
\]

where Indicated Power is in kilowatts

\[
p_m = \text{mean effective cylinder pressure (N/m}^2)\]

\[
L = \text{length of stroke (m)}
\]

\[
A = \text{area of piston (m}^2)\]

\[
N = \text{power strokes per second (r/s)}
\]

Note that the formula applies for single cylinders but for multi-cylinders the value is increased accordingly. Also note that the speed is in revolutions per second to maintain coherent units.

The indicated power of an engine may also be obtained by fixing a device to the engine which produces a graph of engine cylinder pressure and stroke volume. Then, the area of the graph provides an indication of the work done:

\[
\text{Pressure (N/m}^2) \times \text{Volume (m}^3) = \text{Work done (N m)}
\]

and if this value is multiplied by the engine speed (r/s), the indicated power is determined.

The indicated power does not provide a figure that may be used as a design factor but it does provide an indication of the efficiency of the engine when compared with the actual or brake power.

Torque was detailed in Chapter One. It is a function of power and engine speed in the relationship:

\[
\text{Power (kW)} = \frac{2\pi \times \text{speed (r/s) \times torque (N m)}}{1000}
\]

However, the torque output of an engine is not exactly proportional to the speed because an engine has a peak torque speed range. In other words, the engine is most powerful (speed x torque) at a particular speed, usually below maximum speed.

*Volumetric Efficiency* is the ability of the engine to take in a full charge of air or air/fuel mixture in the short time that the appropriate valve is open.
Bore and Stroke - as shown in Figure 6.1, the bore is the internal diameter of the cylinder and the stroke is the distance travelled by the piston.

Top Dead Centre (abbreviated T.D.C.) is the position the piston assumes when it reaches its maximum distance from the centre-line of the crankshaft. This position is also referred to as the top of the stroke.

Bottom Dead Centre (abbreviated B.D.C.) is the opposite end of the stroke to T.D.C. and is known as bottom of the stroke.

Compression Ratio is the relationship between the volume in the cylinder when the piston is at the bottom of its stroke (B.D.C.) and the volume of the cylinder when the piston is at the top of its stroke (T.D.C.).

The amount of clearance volume above the piston has a major effect on the compression ratio. Compression ratios range from 6:1 to 10:1. The higher ratios provide a higher engine output power but the increased pressures place greater loading on the engine's components.

Displacement of an engine is the volume of air the pistons displace in moving from bottom dead centre to top dead centre. For a multi-cylinder engine the displacement equals:

\[
\text{the piston area} \times \text{stroke} \times \text{number of cylinders.}
\]

Being a volumetric measurement, the displacement of an engine is stated in litres.

Two-Stroke engines have a power stroke each time the piston completes one cycle, up and down. On a single-cylinder engine, one power stroke occurs for each revolution of the crankshaft.
Four-Stroke engines require four strokes of a piston to complete one cycle of operation. On a single-cylinder engine, one power stroke occurs for each two revolutions of the crankshaft.

Spark Ignition engines operate on the principle of an electric spark igniting a mixture of fuel and air.

Compression Ignition engines operate on the principle of fuel being injected into a hot compressed charge of air and spontaneous combustion occurring.

6.2 FOUR-STROKE ENGINES

A four-stroke engine may be either spark ignition (petrol as fuel) or compression ignition (diesel oil as fuel). The four-stroke petrol engine will be considered first because it is the type that is commonly used as a car engine.

Figure 6.2 shows a four-stroke engine with the main components labelled. The camshaft operates the valves at the correct time, relative to piston stroke, to allow the fuel mixture to enter (intake valve) or the exhaust gases to escape (exhaust valve). The carburettor supplies the petrol/air mixture and is controlled by a butterfly valve shown just below the carburettor. The ignition system comprises a high voltage which is switched to the spark plug at the correct time in the piston stroke, causing a spark which ignites the fuel.

![Fig. 6.2 Four-Stroke Engine Components](image)
Fig 6.3 shows the four strokes of the engine during two complete revolutions of the crankshaft.

(A) Induction Stroke
The piston is moving down, the inlet valve is open and a mixture of air and petrol vapour is being drawn into the cylinder through the inlet valve, from the carburettor.

(B) Compression Stroke
The inlet valve has closed, no valves are open, the piston is moving up and the air/petrol mixture is being compressed. The pressure at the end of the compression stroke will be about 1.7 MPa and the temperature in the region of 370°C. The temperature should be as high as possible without causing spontaneous combustion so that the fuel mixture will burn very rapidly when ignited by the spark plug towards the end of the stroke.
(C) **Power Stroke**
An electric spark across the points of the spark plug is timed to ignite the fuel mixture just before the piston reaches top dead centre so that full burning effect takes place almost instantaneously while the piston is at the top of the stroke.

The burning fuel mixture expands rapidly to cause a rise in pressure and temperature. Combustion takes place in about 0.03 seconds and is really like an explosion. The hot gases in the cylinder force the piston down which, in turn, causes the crankshaft to rotate. As the piston descends, the hot gases expand and the internal pressure decreases.

(D) **Exhaust Stroke**
The camshaft has opened the exhaust valve, and the piston now moves up, expelling the exhaust gases. Note that the piston is now being driven by the crankshaft which is itself being driven by rotational energy stored in the flywheel.

**VALVE TIMING**
As previously stated, the spark occurs just before the piston reaches the top of the compression stroke. Likewise, valve operation occurs not, as would be expected, at the end of a stroke, but slightly before or after. Figure 6.4 shows a timing circle, the conventional way to display the valve and spark timing.

![Fig 6.4 Timing Cycle](image)

The diagram really consists of two circles, one superimposed on the other, since the four-stroke cycle is completed in two revolutions. Note that the inlet and outlet valves do not commence to open at the dead centre positions but at some degrees on either side of these positions.

The inlet valve opens before T.D.C. and closes after B.D.C. This is done in an attempt to get as much fuel-air mixture into the cylinder as possible. When the inlet valve opens, the air-fuel mixture outside is stationary and has to be accelerated to the velocity of the descending piston. This takes time and so the inlet valve is opened early to allow for this acceleration.
Piston Engines

Once the air-fuel mixture is moving it possesses kinetic energy which is used at the end of the induction stroke to produce a ramming effect by closing the inlet valve some degrees after B.D.C. More air-fuel mixture is pushed into the cylinder in this way, thus increasing the potential output of the engine.

In order to exhaust the products of combustion as quickly as possible, the exhaust valve opens early, some degrees before B.D.C. Thus, some of the exhaust gas leaves by virtue of its excess pressure above the outside atmospheric pressure and hence it is flowing freely from the cylinder by the time the piston commences the exhaust stroke.

By closing the exhaust valve late, that is, some degrees after T.D.C., the kinetic energy of the exhaust gas can be utilised to assist in maximum exhausting of the cylinder before the exhaust valve closes. Note that the inlet valve begins to open before the exhaust valve closes. This is called valve overlap.

Also note that the igniting spark occurs before T.D.C. This is done so that as full combustion of the fuel takes a short time, the maximum explosive force (pressure) occurs just as the piston commences its downward stroke. Because the engine speed varies, the number of degrees before T.D.C. that ignition occurs is modified to suit engine speed. This is an automatic advance-retard system. If the ignition is too far advanced, excessive cylinder pressures are produced and the resultant shock waves often cause an audible metallic sound called pinking. Also the engine suffers a loss of power and overheats.

6.3 TWO-STROKE ENGINES

As the name implies, all the events in the two-stroke engine are completed in two strokes, or one revolution of the crankshaft. The engine and its cycle are shown in Figure 6.5.

![Fig. 6.5 Two-Stroke Engine Cycle](image)
Control of admission and exhaust in this engine is by means of ports let into the side of the cylinder and also by means of the piston. The piston in this type of engine is also the engine valve.

The crankcase is made gas tight, since the incoming air-fuel mixture passes through the crankcase on its way to the cylinder. There is an inlet port near the base of the cylinder through which the air-fuel mixture passes into the crankcase, or an inlet valve is situated in the crankcase. A transfer port is led from the crankcase into the cylinder through which the air-fuel mixture is transferred from crankcase to cylinder. There is a further port, the exhaust port, from cylinder to atmosphere, through which the products of combustion are exhausted.

Referring to Figure 6.5, the cycle of events is:

(A) Admission of Fuel Vapour
The piston is moving upwards and the piston is sealing the transfer and exhaust ports. There is a fresh air-fuel charge in the cylinder above the piston and this is being compressed. Since the crankshaft casing is sealed, there is a reduction of pressure in the crankcase as the piston rises. As a result of this below-atmosphere pressure, as the piston uncovers the inlet port in the base of the cylinder, a charge of air-fuel mixture flows in. Alternatively, some engines use a simple valve which opens when the pressure difference across the valve exceeds a certain value.

(B) Power Stroke
Just before the top of the stroke, the charge above the piston is ignited. The piston is pushed down by the rapidly expanding gases. Eventually, as the piston descends, the exhaust port is uncovered and the hot gases rush out to exhaust. Sometimes the top of the piston is specially shaped to align with the ports to channel the gas flow in the correct direction. Also during the power stroke, the descending piston covers the inlet port and the trapped air-fuel mixture is then compressed by the descending piston.

(C) Transfer Stage
As the piston approaches B.D.C., several changes occur. The transfer port is uncovered and the compressed air-fuel mixture in the crankcase is fed to the top of the piston. The incoming mixture tends to assist in ejecting the last of the exhaust gases but careful design is necessary to avoid losing too much of the unused air-fuel mixture. On some larger engines the exhaust removal, called scavenging is assisted by injecting compressed air.

(D) Compression
As the piston ascends it compresses the air-fuel mixture ready to commence another power stroke.

Figure 6.6 shows a cycle timing diagram for a two-stroke engine.

Note that with a two-stroke engine the inlet and outlet ports open and close at equal angles on either side of the B.D.C. position. This is because the piston on this type of engine also acts as the inlet and exhaust valves and hence port opening and closing will occur at equal angles on either side of the bottom dead centre.

Note that the angles shown are representative only and vary for different designs of engine.
6.4 DIESEL ENGINES

Like the spark ignition engines previously considered, the diesel engine is produced in both four-stroke and two-stroke versions, but the major difference between petrol and diesel engines is in the method of fuel ignition. A diesel is a compression ignition engine; that is, fuel ignition does not rely on an electrical spark but is ignited by the internal temperature of the engine, which is caused by a high compression ratio, almost double that found in a petrol engine.

In a four-stroke diesel the air enters the cylinder in the same way as it would enter a petrol engine, rushing to a low pressure area in the cylinder. However, it does not pass through a carburettor to receive the fuel because the fuel is forced directly into the cylinder through the fuel injector.

Figure 6.7 shows the sequence of operations for a four-stroke diesel.

(A) The Inlet Stroke
As the piston descends, the inlet valve opens and air flows in to the cylinder, pushed in by the difference between the atmospheric pressure outside and the negative pressure (vacuum) above the piston inside. In practice, many diesels utilise a blower which forces air into the cylinder.

(B) The Compression Stroke
The piston is moving up, both valves are closed and thus the air in the cylinder is being compressed. As the air is compressed its temperature rises, finally achieving a pressure in excess of 3 MPa and a temperature in excess of 500°C.

(C) Power Stroke
Fuel is injected into the hot compressed air and combustion occurs. Fuel injection pressure is between 5 MPa and 20 MPa which allows the fuel to be injected through a very small jet, causing a very fine fuel spray or atomisation. The fine spray allows rapid ignition to occur. These minute droplets of fuel evaporate on the outer surface, surrounding each with a layer of vapour which soon reaches a temperature where ignition occurs.
Ignition causes a rapid rise in temperature and pressure, causing almost complete combustion of fuel still being injected. Injection may continue for 25-30 degrees or approximately 1/10th crankshaft travel.

There are four distinct phases in the combustion of fuel:

1. The delay period during which each droplet of atomised fuel surrounds itself with vapour. This outer layer reaches a temperature high enough for spontaneous combustion to occur - this is ignition point.

2. This second phase is one of very rapid burning as the fuel which has accumulated during the delay period, being surrounded by air, is ignited in uncontrolled combustion (diesel knock).

3. The third phase is the stage of controlled burning, when, owing to the high temperature existing in the cylinder, the rest of the charge of fuel burns immediately it is injected.

4. Burning continues after the complete charge. This fourth stage consists of fuel that has not mixed with sufficient air to ignite it previously, or fuel that is being evaporated from the walls of the combustion chamber.

As only 1/75th part of a second may be available for this complete combustion cycle, the fuel must be correctly atomised to allow the particles to be completely surrounded with sufficient hot air for them to reach
self-ignition in this time. When the fuel is shut off, the gases continue to push the piston down. At the end of fuel combustion the internal temperature is in excess of 1650°C.

(D) Exhaust Stroke
Near the end of the downward stroke the exhaust valve opens. As the piston moves up again, gases are expelled from the cylinder, and at the top of the stroke the induction cycle recommences.

Note that the exhaust, induction and compression strokes are all powered by the kinetic energy within the engine, so a flywheel is essential to maintain the rotation during non-power strokes.

Figure 6.8 shows the timing diagram for a four-stroke engine.

To ensure sufficient combustion it is essential to clear the burnt gases from the cylinder. For this reason the exhaust valve opens before B.D.C. Because the exhausting gas has momentum at the top of the exhaust stroke, when the inlet valve is opened before T.D.C. the outgoing exhaust tends to draw behind it air via the inlet valve. Thus, at this point of the cycle, valve overlap occurs.

Note that the angles given in Figure 6.8 are typical only and do not necessarily occur on all diesel engines.

Fig. 6.8 Four-Stroke Timing Diagram

The two-stroke diesel engine which, as previously discussed, has a power stroke on every revolution, may appear to have twice the power output of a four-stroke engine of the same size. However, owing to problems in clearing the exhaust (scavenging) the two-stroke has about 1.5 times the power output of a comparable four-stroke. This type of engine differs from the four-stroke in that it has no conventional valving but a crankcase flap or reed valve to allow air intake.

Figure 6.9 shows the four major parts of the two-stroke diesel cycle.
(A) Transfer and Scavenging
As the piston covers the transfer port and then the exhaust port, air is compressed above the piston and a negative pressure is created below the piston.

(B) Crankcase Intake
Air enters the crankcase via the flap valve as the crankcase pressure goes more negative.
(C) Injection and Start of Power Stroke
Fuel is injected in an atomised spray into the highly compressed air above the piston and ignition occurs. The flap valve closes as the piston reaches T.D.C.

(D) Power Stroke and Exhaust
The hot gases push the piston down until the exhaust part is uncovered and the gases can exhaust. Beneath the piston, air is compressed and as the transfer port is exposed, air flows into the cylinder from the crankcase. This incoming air helps to sweep the remaining gases out through the exhaust port. However, this method of cleaning the exhaust gas is far from efficient and in many two-stroke diesels the transfer-port method is not used, the scavenging being done by a separate air blower.

A typical timing diagram for a two-stroke diesel is shown in Figure 6.10.

![Fig. 6.10 Two-Stroke Diesel Timing Cycle](image)

6.5 INDICATED POWER OF AN ENGINE

The indicated power is the power actually developed in the cylinders. The brake (or output) power will always be less due to transmission losses, mainly frictional.

To obtain the indicated power it is necessary to know the actual conditions that exist above the piston in terms of pressure and volume. If pressure and volume occurring throughout the engine cycle are displayed graphically, the resulting graph area represents work done by the engine.

Consider a theoretical pressure/volume diagram as shown in Figure 6.11 which is for a four-stroke diesel engine:

![Fig. 6.11](image)
Air intake is from A to B (note a slightly negative pressure, if AL represents atmospheric pressure).

When the piston reaches B.D.C. the inlet valve closes (point B) and the piston then moves up the cylinder, compressing the air (point B to C). Note that the curve of the line BC relates to the polytropic compression discussed in Chapter 2.

When the piston reaches T.D.C., fuel injection occurs. Remember that Figure 6.11 shows a theoretical diesel cycle and in practice injection would occur before T.D.C.

Between points C and D the fuel burns and the diagram shows that if the pressure is kept constant, the volume must increase, and thus the piston starts to move back down the cylinder.

As the piston descends further, the pressure starts to reduce, shown by line DE. At B.D.C. the exhaust valve opens and the piston commences its upwards exhaust stroke. The cylinder volume reduces and pressure goes back to atmospheric (line EA).

At T.D.C. the exhaust valve closes and the inlet valve opens. The engine then commences another cycle of operations. The actual indicator diagram varies from the theoretical because:

- the valves do not give a precise cut-off
- valve overlap occurs
- the temperature and pressure affect the operation.

Figure 6.12 shows some typical diagrams.

![Fig. 6.12 Typical Engine Indicator Diagrams](image)
Piston Engines

The diagram can provide a lot of information about an engine.

The clearance volume is represented by the distance from the pressure axis to the line AC (in Figure 6.11).

The compression ratio is represented by the ratio of the length from the pressure axis to the B.D.C. line relative to the length from the pressure axis to line AC.

The work done by the engine is represented by the graph internal area. This is because, taking units of N/m$^2$ for pressure and m$^3$ for volume, the area of the diagram is:

$$\text{Area} = \frac{N}{m^2} \times m^2 = N \text{ m} = \text{joules}$$

The indicated power of the engine is the indicated work done, divided by the time taken.

A convenient measure of the speed at which the engine operates is its rotational speed (r/min).

So:  \[ \text{Indicated power (W)} = \text{Diagram area (N m)} \times \text{speed (r/s)} \]

Note that the speed must be in r/s to maintain the relationship

\[ \text{Power (W)} = \frac{\text{Work done (N m)}}{\text{Time taken (s)}} \]

The indicated mean effective pressure may be determined from the diagram. The mean effective pressure is the constant pressure which, if it acted over the full length of the stroke, would produce the same amount of work done by the piston as is actually produced during a complete engine cycle.

The mean effective pressure may be determined from the diagram by:

1. dividing the diagram up into vertical strips and taking the average height of the strip (in terms of the pressure scale)
2. dividing the diagram up into vertical strips and using Simpson's rule to find the area. Then the average height (pressure) is the area (work done) divided by the diagram length (volume)
3. using a planimeter to measure the area and then dividing by the length, as in (b) above.

In terms of engine performance checking the indicator diagram is very useful for the early detection of faults within the engine. If a diagram is produced on an engine at regular intervals, as a fault develops it can be identified on the diagrams and then remedied before major damage occurs.

Figure 6.13 shows a diagram for a two-stroke diesel and the effect, dotted, when the fuel is injected too early, causing excessive pressure rise.

Figure 6.14 shows the same engine but now the exhaust valve timing is too early, causing a premature drop in cylinder pressure.
An engine indicator is an instrument used to produce a pressure/volume diagram of an engine.

Note that although reference has only been made to internal combustion engines, the same instrument may be used on steam engines and even to investigate the pressure/volume relationship of a piston-type air compressor.

Figure 6.15 shows the principle of operation of an engine indicator. Pressure above the engine piston acts on the indicator piston and forces it upwards against the indicator piston spring. Because the spring rate has been carefully determined beforehand, the travel of the indicator piston may be directly related to the pressure applied. By a system of linkages, the cylinder volume is represented by movement of the indicator cord.

The actual engine indicator is shown in Figure 6.16.
As in the schematic diagram, engine pressure is indicated by vertical movement of the indicator piston and magnified by the parallel link mechanism. Volume change is related to the rotation of the drum by the link cord. In operation, a combination of vertical pressure display and horizontal volume display is recorded on a card on the drum. The resultant diagram, called an *engine indicator diagram* is then interpreted in terms of pressure and volume.

Owing to the mechanical nature of this type of indicator, it is not suitable for use on high-speed engines. The inertia of moving parts, friction and the presence of a large number of mechanical wear points results in its being restricted to use on engines that operate below 900r/min. Above this speed, electronic measurement and display methods are utilised.

**MORSE TEST**

In multi-cylinder internal combustion engines where all the cylinders are of the same cubic capacity, a reasonable estimate of the indicated power developed in each cylinder can be made by the *Morse Test*. This is most useful in small high-speed engines where indicator diagrams cannot be taken satisfactorily by the standard mechanical indicator.

The test consists of measuring the brake power at the shaft when all cylinders are firing and then measuring the brake power of the remaining cylinders when each one is “cut out” in turn. Cutting out the power of each cylinder is done in petrol engines by shorting the sparking plug, and in diesel engines by by-passing the cylinder fuel supply. The speed of the engine and the petrol throttle or fuel pump setting is kept constant during the test so that friction and pumping losses are approximately constant.
Taking a four-cylinder engine:

With all four cylinders working,

\[
\text{Total b.p.} = \text{Total i.p.} = \text{Total f.p.} = \text{sum of i.p. of 3 cylinders} - \text{sum of f.p. of 4 cylinders}
\]

When the power of one cylinder is cut out,

\[
\text{Total b.p.} = \text{sum of i.p. of 3 cylinders} - \text{sum of f.p. of 4 cylinders}
\]

\begin{align*}
\text{b.p.} & \quad = \quad \text{brake power} \\
\text{i.p.} & \quad = \quad \text{indicated power} \\
\text{f.p.} & \quad = \quad \text{friction power}
\end{align*}

Hence, when one cylinder is cut out, the loss of power at the shaft is the loss of the indicated power of that cylinder which is not firing.

**EXAMPLE**

During a Morse test on a four-cylinder, four-stroke petrol engine, the throttle was set in a fixed position and the speed maintained constant at 2000 r/min by adjusting the brake, and the following readings taken:

- with all cylinders working, b.p. developed = 57 kW
- with sparking plug of No 1 cyl. shorted, b.p. = 37 kW
- with sparking plug of No 2 cyl. shorted, b.p. = 38.5 kW
- with sparking plug of No 3 cyl. shorted, b.p. = 37.5 kW
- with sparking plug of No 4 cyl. shorted, b.p. = 38 kW

Estimate the i.p. of the engine and the mechanical efficiency.

\[
i.p. \text{ of No 1 cyl.} = 57 - 37 = 18.5 \text{ kW} \\
i.p. \text{ of No 2 cyl} = 57 - 38.5 = 20 \text{ kW} \\
i.p. \text{ of No 3 cyl.} = 57 - 37.5 = 19.5 \text{ kW} \\
i.p. \text{ of No 4 cyl.} = 57 - 38 = 19 \text{ kW}
\]

Total i.p. = 77 kW

\[
\text{Mechanical Efficiency} = \frac{\text{b.p.}}{\text{i.p.}} = \frac{57}{77} = 0.74 \text{ or } 74\%
\]
6.6 ENGINE PERFORMANCE

(A) INDICATED POWER: (i.p.)

Let

- \( p_m \) - indicated mean effective pressure
- \( L \) - length of piston stroke
- \( A \) - piston area
- \( N \) - number of power strokes per second

Then, force on piston = \( p_m A \) newtons

Work done per stroke = \( p_m L A \) joules

Work done per second = \( p_m L A N \) watts

Note that for a four-stroke engine, \( N \) equals the engine speed divided by two.

**EXAMPLE**

The area of an indicator diagram taken from one cylinder of a four-cylinder, four-stroke engine is 378 mm\(^2\), the length is 70 mm and the indicator spring rate is 1 mm = 100 kPa. The diameter of the piston is 250 mm, stroke 300 mm and speed 300 r/min. Calculate the indicated power of the engine, assuming all cylinders develop equal power.

\[
\text{mean height of diagram} = \frac{\text{area}}{\text{length}} = \frac{378}{70} = 5.4 \text{ mm}
\]

Indicated \( p_m \) = mean height × spring scale

\[= 5.4 \times 1 = 540 \text{ kPa}\]

\[N = \frac{300 \text{ r/min}}{60 \times 2} = 2.5 \text{ power strokes per minute}\]

Indicated power = \( p_m LAN \)

\[= (540 \times 10^3) \times 0.3 \times \left( \frac{\pi \times 0.25}{4} \right) \times 2.5\]

\[= 79.48 \text{ kW for four cylinders}\]

(B) BRAKE POWER: (b.p.)

As discussed in Chapter One, the brake power of an engine is determined by measuring the engine output torque and speed, and using the formula:

\[\text{Power} = 2\pi NT\]

where \( N \) is rotational speed in r/s

and \( T \) is torque in newton metres

**EXAMPLE**

An engine runs at 250 r/min. A rope brake is applied to the 600 mm diameter flywheel and loaded up against a spring balance. Load on the free end is 425 N and the spring balance reads 72 N. Calculate the brake power.
Braking torque = \((425 - 72) \times 0.3\) = 105.9 N m

Brake power = \(2\pi NT\)
\[= 2\pi \times \frac{250}{60} \times 105.9\]
\[= 2.772\ kW\]

(C) THERMAL POWER

The thermal power of an engine is the amount of power it should develop calculated from the amount of heat energy (fuel) it consumes.

If an engine consumes 1 kg of oil per minute, and the calorific value of the oil is 44 MJ/kg, its thermal power equals:

\[\text{Thermal Power} = \frac{44 \times 10^6}{60}\ \text{joules per second}\]
\[= 733\ kW\]

(D) FRICTION POWER

The difference between the indicated power and the brake power of an engine is called the friction power (or the power lost due to friction).

\[\text{Friction power} = \text{i.p.} - \text{b.p.}\]

(E) MECHANICAL EFFICIENCY

The ratio of the actual output from an engine compared with its indicated power is its mechanical efficiency.

\[\text{Mechanical Efficiency (\%)} = \frac{\text{b.p.}}{\text{i.p.}} \times 100\]

(F) THERMAL EFFICIENCY

The thermal efficiency of an engine must be related to either the brake power or the indicated power conditions. The most useful thermal efficiency is the Brake Thermal Efficiency:

\[\text{Brake Thermal Efficiency} = \frac{\text{heat energy converted into work}}{\text{heat energy applied}} = \frac{\text{brake power}}{\text{thermal power}}\]

EXAMPLE

The following data was taken during a one-hour test of a single-cylinder, four-stroke diesel engine. The engine has a cylinder of 175 mm diameter and 225 mm stroke. Engine speed is 1000 r/min.
Indicated mean effective pressure = 550 kPa  
Diameter of rope brake = 1066 mm  
Load on brake = 400 N  
Spring balance reading = 27 N  
Fuel consumed = 5.7 kg  
Calorific value of fuel = 44.2 MJ/kg

Calculate the indicated power, the brake power, mechanical efficiency and brake thermal efficiency.

Indicated power = \( p_{\text{i.p.}} = \frac{1}{n} \times \frac{\pi \times 0.175^2}{4} \times 1000 \times \frac{550 \times 10^3}{60 \times 2} \) 

= 24.8 kW

Brake power = \( 2\pi NT \) 

= \( 2\pi \times \frac{1000}{60} \times (400 - 27) \times \frac{1.066}{2} \) 

= 20.82 kW

Mechanical Efficiency = \( \frac{\text{b.p.}}{\text{i.p.}} \) 

= \( \frac{20.82 \times 10^3}{24.8 \times 10^3} \) 

= 0.839 or 83.9%

Brake Thermal Efficiency = \( \frac{\text{b.p.}}{\text{thermal power}} \) 

= \( \frac{20.82 \times 10^3}{(44.2 \times 10^6 \times 5.7)/3600} \) 

= 0.297 or 29.7%

**EXAMPLE**

The following results were obtained during an engine test on a four-stroke engine:

Mean height of indicator diagram = 21 mm  
Indicator spring rate = 27 kPa/mm  
Swept volume of cylinder = 14 litres  
Engine speed = 6.6 t/s  
Brake effective loading = 77 kg  
Brake radius = 0.7 m  
Fuel consumption = 0.002 kg/s  
Calorific value of fuel = 44 MJ/kg

Calculate:

(i) Indicated power  
(ii) Brake power  
(iii) Brake thermal efficiency  
(iv) Mechanical efficiency
Indicated mean effective  =  27 x 21  =  567 kPa

(i)

Indicated power = \( p_m \cdot LAN \)

But length \( \times \) area = swept volume

\[
i.p. = (567 \times 10^3) \times (14 \times 10^{-3}) \times \frac{66}{2}
\]

= 26.2 kW

(ii)

Brake power = \( 2\pi NT \)

\[
= 2\pi \times 6.6 \times (77 \times 9.8 \div 0.7)
\]

= 22 kW

(iii)

Brake Thermal Eff'\( y \) = \( \frac{\text{b.p.}}{\text{thermal power}} \)

\[
= \frac{22 \times 10^3}{(44 \times 10^3) \times 0.002}
\]

= 0.25 or 25%

(iv)

Mechanical Efficiency = \( \frac{\text{b.p.}}{\text{i.p.}} \)

\[
= \frac{22 \times 10^3}{26.2 \times 10^3}
\]

= 0.84 or 84%
STEAM AND GAS TURBINES

7.1 TURBINE PRINCIPLES

A turbine is a form of heat engine in which power is generated by the flow of gases or steam passing across curved blades attached to the circumference of a rotor, thus causing the rotor to rotate. The action could be compared to that of the wind on the sails of a yacht sailing across the direction of the breeze.

![Diagram of wind and yacht](image)

Fig. 7.1

Considering a steam turbine first, the available heat energy in the steam is converted into kinetic energy by the expansion of steam in a suitably shaped nozzle, from which it issues as a high velocity jet. A portion of this kinetic energy is then converted into mechanical energy by directing the jet, at a suitable angle, against curved blades mounted on a revolving disc attached to a shaft. Another portion of the energy can be converted by the steam, giving a reactive thrust as it leaves the blades.

The pressure on the blades causing rotary motion is due solely to the change in momentum of the steam jet in its passage through the blades.

There are two types of turbine, the *impulse* and the *reaction*. In both cases the steam is allowed to expand from a high pressure to a low pressure region so that the steam acquires a high velocity at the expense of a fall in pressure. This high velocity steam is directed onto curved section blades which absorb some of the velocity.

In impulse turbines the steam is expanded in nozzles in which a high velocity of steam is achieved before it enters the blades on the turbine rotor. The pressure drop and subsequent increase in velocity take place in the nozzle. As steam passes over the rotor blades it loses velocity but there is no fall in pressure.

In reaction turbines, expansion of the steam takes place as it passes through the moving blades on the rotor, as well as the guide blades fixed to the turbine casing.

In a gas turbine, the kinetic energy in the gases also comes from heat energy content. The heat is obtained by burning fuel in compressed air and the products of combustion, the exhaust, is used to drive a rotor. Part of the power of this type of turbine is fed back into the system to drive the compressor which supplies air to the turbine. The remaining energy in the gases can be used to produce either mechanical power or, in the case of aircraft, thrust.
7.2 IMPULSE TURBINES

The simple impulse turbine consists essentially of one or more nozzles, each supplied with high pressure steam. The discharge jet impinges, at a suitable angle, on a single row of buckets on a revolving disc. The steam expands in the nozzle to the exhaust pressure, and its velocity increases during the expansion. The shape of the blades on the turbine wheel are curved to give a change of direction to the gas. They also impart a corresponding thrust on the wheel circumference to produce rotary motion. Figure 7.2 shows the basic principle of an impulse turbine. The thrust shown causes an end load on the turbine bearings.

Simple impulse turbines are generally limited in output, although units up to 2500 kW have been produced. Operating speed is up to 300 revolutions per second and thus the output is normally geared down to produce a usable output speed. To achieve a higher power rating, a series of rotors are used.

Velocity-compounded turbines utilise a very high velocity jet from a nozzle more efficiently than the simple impulse turbine. For moderate velocities the simple impulse is more efficient. In the velocity-compounded turbine the steam, after passing through one row of buckets on the rotor, is re-directed by a fixed row of blades, buckets or passages against a second row of moving buckets. This reversal and re-impingement may occur several times before the steam leaves the turbine. When velocity compounding consists of two rows of moving blades, with intermediary stationary reversed blading, the form results in the "Curtis" or two-row wheel. (Figure 7.3)
In another type, the reversing passages redirect the steam into the same row of blades that the steam first passed across. This is called the re-entry type. (Figure 7.4)

Helical flow "Terry" turbines employ a forged steel wheel in which semi-circular buckets are milled in the vane at an angle of about 30° to the tangent. The steam expands to exhaust pressure in the nozzles which direct steam into one side of the bucket. The steam gives up part of its energy in its first reversal of 180° and passes to a reversing chamber which redirects the steam back into the buckets. This process is continued until the kinetic energy in the steam is nearly exhausted. The whole operation occurs in a single wheel, which is usually provided with several nozzles and reversing chambers. The nozzles may be provided with shut-off valves, thus giving a means of regulating the turbine power output. (Figure 7.5)
In all velocity-compounded turbines, part of the kinetic energy is absorbed each time the steam passes through a revolving blade or bucket passage. More work per kilogram of steam is thus obtained than in a simple impulse turbine of the same size and bucket speed. Friction losses, however, increase with the addition of the reversing chambers.

Although these types of turbine are generally used in small steam plants, they frequently form the first stage of very large compound, or multi-stage, units. Stage in an impulse type turbine is a term which refers to that part of the machine in which a pressure drop occurs in conjunction with an increase in velocity (nozzle), together with such passages where no further pressure drop occurs (bucket). A stage, therefore, includes nozzle, moving bucket and reversing chamber.

The multi-stage impulse turbine consists of a series of simple impulse turbines in the same casing on the same shaft. Each of these forms a stage. It is designed so that the steam expands through a series of pressure drops between inlet and exhaust, a pressure drop occurring at each stage.

On leaving the buckets of the first wheel, steam enters the second stage nozzles and expands through a further pressure drop. The jet then impinges on a second row of moving blades. This is repeated at each stage throughout the turbine cycle until the steam is exhausted. Extremely high efficiencies are possible with this turbine which is generally called a "Pressure Compounded" turbine, (see Figure 7.6).

Pressure - velocity compounding is a combination of both of the previous methods, and has the advantage of allowing a larger pressure drop across each stage, thus making fewer stages necessary. Hence for any given pressure drop, the turbine will be shorter. The diameter of the blade ring is increased at each stage to allow for the increased steam volume.
7.3 REACTION TURBINES

In the reaction type machine, the principle of operation is different. Work is still done by the impulse due to the reversal of direction of the high velocity steam jet on the moving blades. However, both the fixed and moving blades are so designed that the steam expands as it passes through each of them thus giving, in addition, a reactive thrust due to the expansion of the steam through the moving blades.

A rotary garden sprinkler is a perfect example of the reactive effect. As the water leaves the sprinkler, the reactive thrust causes rotation of the nozzles so a large area of lawn is watered.

![Diagram of a Parsons Turbine](image)

Fig. 7.7 Steam Path in a Parsons Turbine

Since, in a reaction type turbine, a pressure drop also occurs across the moving blades, it is necessary to provide effective sealing at the blade tips. This must be done to prevent leakage of steam past the shrouding of the wheel, and loss of efficiency, especially at the high pressure end of the machine. This is shown in Figure 7.7.
Modern steam turbines employ both impulse and reaction principles. Impulse is used at the high pressure end, where sealing is of greater importance, and reaction blading at the low pressure end where, because of the smaller pressure drop across the blading, sealing is not so important.

A typical turbine is shown in Figure 7.8.

![Fig. 7.8 Turbine Construction](image)

### 7.4 GAS TURBINES

Development of the gas turbine has been accelerated by the requirements of increased industrial activity. At first, research was focussed almost entirely on the needs of the aircraft propulsion but now have moved into the industrial and marine fields, even being used for car and truck propulsion.

Unlike the steam turbine in which a steam supply comes from a separate boiler, the "gas generator" of a gas turbine is an integral part of the unit.

The engine basically consists of:

- an air compressor
- flame tubes (in the combustion area)
- drive turbine.

Power output is via the compressor shaft or via an output reduction gearbox.

A typical gas turbine layout is shown in Figure 7.9.
Chapter Seven

7.7

Fig. 7.9  Gas Turbine

The gas turbine will not start by simply turning on the burner. The turbine must be run up to a minimum speed by a starter motor. When this speed has been reached the fuel is turned on, ignited and the turbine will then pick up speed. One of the best qualities of a gas turbine is its multi-fuel capacity. It will operate on practically any liquid hydrocarbon, such as petrol, diesel fuel, kerosene or jet fuel.

AIR CYCLES IN TURBINES

There are two main cycles or types of air circulation used in gas turbines:

1. Open Cycle
   In this open cycle the exhaust gases pass from the turbine and exhausts directly into the atmosphere, with a continuous fresh supply of air being drawn into the compressor. This is shown in Figure 7.10.

Fig. 7.10  Open Cycle
The efficiency may be improved in this cycle by using a heat exchanger (regenerator) to preheat the compressed air by using the exhaust gases from the turbine, shown in Figure 7.11.

Fig. 7.11 Turbine with Regenerator

2. Closed Cycle
In the closed-cycle turbine the air does not come in contact with the burning fuel. The turbine exhaust is cooled and then it re-enters the compressor as in Figure 7.12. The compressed air is heated externally by the use of oil, gas or solid fuel.

Fig. 7.12 Closed Cycle Turbine

In some gas turbine installations both open and closed cycle principles are employed.

FUNCTION OF TURBINE COMPONENTS

Pre-Cooler
The pre-cooler is a heat exchanger which removes heat from the incoming air prior to compensation.

Compressor
The function of the compressor is to deliver the air to the combustion chamber in sufficient quantity and at the required pressure.
**Combustor**

The combustor provides the space for burning the fuel and air mixture. This is normally a flame tube with the fuel being sprayed into the air flow.

**Turbine**

The turbine wheel and blades convert the energy in the expanding gases from the combustor into mechanical energy.

**After-cooler**

In a closed air system, the after-cooler is used to cool the exhaust gas from the turbine before re-entry to the compressor.

**TURBO-JET ENGINES**

The turbo-jet engine consists of the typical gas turbine with a standard compressor, combustion chamber and turbine as shown in Figure 7.13. The engine also has a tail piece containing a propulsion nozzle.

![Fig. 7.13 Turbo-Jet Engine](image)

**METHOD OF OPERATION**

The turbo-jet engine must be started in a similar manner to the straight gas turbine by using a starter motor to bring the rotor up to sufficient speed to induce air into the combustion chamber by the compressor. This compressed air at a ratio of 4:1 is directed into the combustion chamber where fuel is pumped in at approximately 5 MPa and burnt continuously, in a manner similar to a gas torch.

In flight, the work of compressing the air is greatly assisted by the forward motion of the aircraft. The lower temperatures encountered at high altitudes also improves the efficiency of the compression.

In this engine only a small quantity of air is used for combustion and the remainder is used to reduce the combustion chamber temperature before the gas enters the turbine. The heat released by combustion expands the air which increases in velocity as it moves through the turbine and out through the propulsion nozzle.
A diagrammatic representation of an aircraft's turbo-jet engine is shown in Figure 7.14.

Fig. 7.14  Aircraft Turbo-Jet Engine

7.5 TURBINE OPERATION

Turbines have gained in popularity because of their essential differences from other engines. These differences are:

1. Fewer Moving Parts
   The turbine rotor and associated blading are the only moving parts, apart from external control gear.

2. Ease of Control
   The speed and thus power of a turbine is controlled by the supply of operating gas (steam or fuel). This flow may be rapidly varied and the turbine response is equally rapid.

3. Smooth Operation
   The single direction rotation gives smooth "pulse less" power drive at the output shaft. However, the high operating speeds mean that the turbine rotors must be well-balanced. Even minor imbalance can cause vibration and ultimately major damage.

4. Power/Mass Ratio
   The power/mass ratio of turbines is very good for the power developed. Reciprocating engines have a mass of approximately 1.75 kg for each kW of power developed, while turbines require approximately 0.85 kg for the same power output.

5. Noise Level
   The main noises associated with turbines comes from either the mechanical attachments such as gearing or the flow of the gas. Correct design of the first and muffling of the other gives the turbine an even and relatively quiet operation. Vibration noise is kept to a minimum due to precise balancing of the turbine rotor.
COMPARISON OF STEAM AND GAS TURBINES

Steam Turbines

- The steam turbine has direct rotary motion and needs no internal lubrication as there is no mechanical friction.

- Steam turbine nozzles are designed to give "set" turbine speeds by the volume of steam passing through them.

- High thermal efficiency and operating economy are features of these turbines.

- Steam turbines attain maximum efficiency only when run at very high speeds.

- These units can be built in larger sizes than reciprocating engines, some more than 200 MW.

Gas Turbines

- Gas turbines have far better power-to-weight ratio than steam turbines.

- These turbines are less critical in their fuel requirements and are more cool to run.

- Gas turbines are more suitable for mobile applications as the heat combustor is part of the turbine.

- The units are compact and easily adaptable for land or marine applications.

- Because of the smaller rotors and the absence of a boiler, coupled with the use of anti-friction bearings, the turbine may be left inactive for a long time but can be quickly brought into service.

TURBINE APPLICATIONS

1. Power Generation.

Stationary plants used for electrical generation may be either steam or gas turbine driven. Steam is used in large units which carry the "base" or constant load and gas turbine driven units are used to carry the "peak" load. (See Figure 7.9.)

Often the gas turbine driven unit is located in an area where the electrical power is required - it does not have to be close to a major fuel supply (such as coal-fired power stations).

A diagrammatic representation of a two-stage steam turbine power generation system is shown in Figure 7.15.

![Two-Stage Steam Turbine Generator](image)

Fig. 7.15 Two-Stage Steam Turbine Generator
The two-stage turbine shown in Figure 7.15 is further explained in Figure 7.16. The H.P. (high pressure) and L.P. (low pressure) stages refer to the dual use of the steam. Initially, super-heated steam is fed to the H.P. turbine. Then the exhaust from the H.P. stage is returned to the boiler and re-heated before being fed to the L.P. stage of the turbine.

Fig. 7.16 Two-Stage Turbine Schematic

2. Marine Propulsion
Marine turbines generally use steam as their power source. Gas turbines are used in smaller vessels such as Naval pursuit-type ships. Marine turbines are normally a compound of two or three stages coupled through a gear reduction box, as shown in Figure 7.17.

Because a ship has to have the ability to go astern, the L.P. turbine contains an additional in-built turbine which provides reverse rotation. Also on ships are turbines that drive boiler feed pumps, forced-draught fans and power generators.

3. Aircraft Propulsion
Aircraft use gas turbines working on the jet or ram jet principle. Here the fuel is burnt with air that has been supplied by a natural forward travel of the engine, or force fed by a compressor. The burning of the fuel and compressed air provides the working gas for the turbine.

A diagrammatic version is shown in Figure 7.14 and another type is shown in Figure 7.18. The latter includes an indication of the pressures, temperatures and gas velocities occurring in the engine.

The aircraft engine uses either the direct thrust of the hot gases or uses, on a turbo-prop system, a drive through a reduction gearbox to a propeller, plus some thrust from the exhausting gas.
Fig. 7.17 Marine Turbine System

Fig. 7.18 Aircraft Engine
Chapter Eight

MECHANICAL DRIVE COMPONENTS

8.1 KEYS AND KEYWAYS

The main function of a key is to transmit torque between a shaft and the machine part assembled with it. In most cases, keys prevent relative motion, both rotary and axial.

Many types of keys exist, but in an attempt to standardise, a selection of the most popular types has been documented by standard-preparing groups, such as British Standards or the Standards Association of Australia. Thus, the more important key types will be indicated by the inclusion of the appropriate Standard number.

Square and rectangular parallel keys are designated in British Standard BS4235. The major dimensions are the breadth and height, as shown in Figure 8.1.

Note from Figure 8.1 that in the assembled condition, clearance exists between the top of the key and the slot in the hub. This means that there is a possibility that the hub could move along the key.

Square and rectangular tapered keys are tapered on the upper surface only (Figure 8.2). The surface projects into the hub and a corresponding taper is cut into the hub key-seat. When the key is driven in, hub and shaft are drawn together and this tends to prevent axial movement of the hub on the shaft. The cost of manufacturing such a system is, of course, more expensive than the parallel key system.

Fig. 8.1

Fig. 8.2 Gib Head Key
Mechanical Drive Components

The Gib Head version of the taper key (also shown in Figure 8.2) is designed for ease of removal. A Woodruff key is a type that may be used in a parallel or tapered situation, but it is cheap to produce and requires the minimum of hand fitting.

As shown in Figure 8.3, the key is a thin disc that has a sector removed and the key seat is a matching circular slot.

![Woodruff Key](image)

Fig. 8.3 Woodruff Key

Where only small amounts of power are to be transmitted and it is impracticable or inadvisable to use a sunk key, a key driving on a flat or a saddle key may be used (Figure 8.4). These keys drive mainly by friction.

The flat key usually has a slight taper, in the region of 1 in 100, so that the key is retained without using a set screw.

The saddle key relies completely on friction for drive purposes and for this reason is only used as a temporary key or in situations where slipping under excess load is a requirement. Again, the key needs to be tapered to achieve retention.

The pin key is another light duty key. It is commonly used on low cost applications where torque transmission is low and the assembly does not have to be dismantled. The pin may be tapered but a roll pin is normally used - this is a pin made of rolled spring steel that fits tightly into a drilled hole.

![Low Power Key Types](image)

Fig. 8.4 Low Power Key Types

Feather keys are used when the hub must slide axially along the shaft but at the same time rotate with it. The feather key is a sunk key. It may be dovetailed into the shaft or fixed by screws in which case it is called a
fixed feather. When the feather key is attached to the hub and moves with the hub, it is called a floating feather. Splined shafts are now largely superseding feather keys. Figure 8.5 illustrates both types of feather keys.

Fig. 8.5 Feather Keys

Where sliding is to occur with a feather, the key and hub are made of dissimilar materials to assure long life.

The method of producing the key-way or key-seat depends on a number of factors, including:

- the equipment available
- function of the key
- the need for avoiding stress concentration.

Figure 8.6 shows two methods of providing a slot for the key to fit.

Fig. 8.6 Keyway Cutting
The left-hand method in Figure 8.6, using a disc type cutter, is fast and provides a precise slot. Its radial ends reduce stress concentration but the key must be retained in place with screws.

The right-hand method using an end mill (or key-seat tool) retains the key axially. In the hub, the key-way is cut by a reciprocating shaped cutter, or for large-scale production, a suitable broach is used.

Note that the purpose of a key has been described as a means of coupling a shaft and hub to transmit torque. A key may also be considered as a torque-limiting device that will shear if excess torque is applied to the system. An example of such usage is the key in the flywheel of a small engine, such as is used to drive a rotary grass mower. An aluminium key is fitted and if the mower blade strikes a solid object, the flywheel key will shear and prevent internal damage in the engine owing to high inertia forces.

8.2 SPLINES

The development of the automobile required a connection between circular shafts and hubs that would be strong and light, would permit axial travel and would be suitable for high volume production.

To transmit high torques with a single key requires a large key and results in high loadings at one point. If the number of keys is increased, their individual dimensions can be reduced, and this is the principle of a splined shaft.

Originally, as in Figure 8.7, spline shapes were made to represent multiple keys.

One major disadvantage of this method of parallel-sided splines is that the abrupt corners cause stress concentration points. This means the spline and shaft dimensions must be larger for a given torque rating.

A later development is the use of the involute form for splines. This shape is similar to that used on gear teeth, the actual mating shapes being dependent on the intended use of the spline. Figure 8.8 illustrates the involute spline forms for various applications.

Note that involute form splines have become increasingly popular because:

- As previously stated, the involute form increases the torque capacity owing to reduction in stress concentration points.
- The splines may be produced on the same equipment and by the same techniques as those used for involute gearing.
- The involute form has a self-centring action under load even when there is backlash (wear) between the mating members.
Also note that care must be taken, when designing or selecting splined fittings, that the special tooling is available for any machining required. In an effort to avoid this problem, companies that market an item that has a splined shaft also sell a mating bush that can be used to convert the shaft to a more conventional drive if a mating splined input is not available.

Spline dimensions and supplementary information are available in Australian Standard B212 for straight-sided splines and B213 for involute splines.

8.3 RIGID COUPLINGS

A coupling is used to connect two shafts permanently - it is disconnected only for repairs or to make a change in the installation.

One other function of a coupling is to provide a form of torque limiting. An example is the coupling that transmits power to the saddle on a lathe. The coupling normally contains a shear pin which will break if the machine is overloaded.
As the name implies, rigid couplings will simply join two shafts rigidly. This means that the shafts must be aligned correctly before coupling, as any misalignment will cause a continuous reversal of bending stresses which will eventually lead to a failure through progressive fracture. In addition, there is always excessive friction and wear on any adjacent bearings.

A flanged rigid coupling, shown in Figure 8.9, is keyed onto each shaft and the two halves are bolted together. Recesses on the mating faces provide alignment for the two halves. Alternatively, one shaft may protrude through to be located in the flange on the other side. Types are available for transmitting up to 2000 kW at 100 r/min.

A split sleeve coupling is shown in Figure 8.10. It is alternatively called a clamp coupling because the two halves clamp onto the shafts. During manufacture the two halves are separated by a thin piece of metal (a shim) and when the unit is bolted into the shafts it clamps securely. In small sizes the coupling is made in cast iron. Cast steel is used for larger sizes where greater strength is important.

Sizes up to 150 kW at 150 r/min.

Fig. 8.10 Split Sleeve Coupling

Compression couplings (Figure 8.11) are based on the pressure created by the wedge action produced by a hollow cone. As the two halves are bolted together, the inner split-sleeve clamps onto the shafts. The other version shown in Figure 8.11 is similar in principle but cheaper because the sleeve compression is by two "hammer-on rings".

Fig. 8.11 Compression Couplings
The Sellers coupling (Figure 8.12) is another type of compression coupling but tends to be more compact than the types previously considered. This is an important factor when considering complex machinery in close proximity.

The central hole is provided for inspection purposes, to check that the two inner cones are not touching.

![Fig. 8.12 Sellers Coupling](image)

### 8.4 FLEXIBLE COUPLINGS

These couplings are used where power is to be transmitted from one shaft to another but there is a possibility that the shafts may become misaligned in service.

Some types of flexible coupling will also smooth out shock caused by a sudden change in transmitted torque or a sudden change in speed. Some types of flexible coupling will allow for relative changes in shaft length, as occurs when the shafts are subjected to varying temperatures with subsequent linear expansion or contraction.

The varying types of flexible coupling make the choice of a suitable coupling difficult because each has different characteristics and thus different applications. As an example of the variations in alignment that can occur between two shafts, Figure 8.13 shows three forms of misalignment.

Part (a) shows axial misalignment, where the shafts are still parallel but not in line. An example would be where the transmitted torque causes one shaft to "roll away" from another.

Part (b) shows angular misalignment, as occurs when a machine deflects or a machine's foundation subsides.

Part (c) shows a combination of alignment errors.

![Fig. 8.13 Possible Shaft Misalignments](image)
The *Spider flexible coupling* (Figure 8.14) transmits torque through an oil-resistant rubber spider assembled between two metal half-bodies.

It is used in the range of 1 kW at 11 000 r/min to 10 kW at 4000 r/min.

![Fig. 8.14 Spider Flexible Coupling](image1)

The *Disc flexible coupling* (Figure 8.15) uses steel pins fixed to metal half-bodies to transmit torque through a flexible disc. The disc is composed of layers of a rubber-impregnated fabric which must not be contaminated with oil.

It will transmit from 30 kW at 6000 r/min to 500 kW at 2000 r/min.

![Fig. 8.15 Disc Flexible Coupling](image2)

Another form of pin coupling is the *Crown Pin coupling* (Figure 8.16) which has pins on one half that locate in rubber bushes in the other half. Crown pin couplings which will transmit up to 7000 kW at 400 r/min are available.

![Fig. 8.16 Crown Pin Coupling](image3)
The *Chain coupling* (Figure 8.17). This coupling is basically two pinions joined by a duplex chain; a cover is often fitted to protect the unit. Some of this type will transmit up to 2000 kW at 750 r/min.

![Fig. 8.17 Chain Coupling](image)

*Internal Gear coupling* (Figure 8.18). Internal gear couplings consist of a pair of gears that fit into an internal gear, but the gear teeth are radiused so that angular misalignment can be accommodated. There are types of this gear coupling available which will transmit up to 70 MW at 250 r/min.

![Fig. 8.18 Internal Gear Couplings](image)
**Laminated Spring or Standage Coupling** (Figure 8.19). This type consists of a spring which fits into slots cut in the two half-bodies. The springs are oil lubricated and the oil is retained by a bolted cover made in two halves with appropriate provisions for sealing.

Although this coupling will withstand a small amount of shaft angular and parallel misalignment, the main function is to take up shock loads that arise from the type of machinery driven, e.g. paper milling machinery, dredges and crushers. Types available will transmit from 120 kW at 2500 r/min to 6500 kW at 300 r/min.

![Fig. 8.19 Spring Coupling](image)

**An Oldham coupling** (Figure 8.20). This type is sometimes known as the spider-type coupling. It has two hubs having jaw-type flanges positioned at right-angles to each other. These jaw flanges engage opposite parallel surfaces of a centre floating member.

This type is best suited where it is necessary to maintain correct angular velocities when the axial lines vary due to positional change of bearings or supports. There are a number of variations to the basic type, for example, a central member with radial flat springs, or a central element of steel with self lubricating bronze inserts. Maximum angular misalignment 1° and out of parallel 0.4 mm.

![Fig. 8.20 Oldham or Slider Coupling](image)

**Rubber Tyre Type.** (Figure 8.21). It can accommodate limited axial and angular alignment and provides cushioning against shock loads.

The type shown in Figure 8.21 couples directly onto the shaft via a tapered locking sleeve.
Typical performance figures for a rubber-type coupling are:

<table>
<thead>
<tr>
<th>Max Speed (r/min)</th>
<th>Max Torque (N m)</th>
<th>Parallel Misalignment (mm)</th>
<th>End Float (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4500</td>
<td>64</td>
<td>1</td>
<td>± 1.3</td>
</tr>
<tr>
<td>960</td>
<td>42 000</td>
<td>6.5</td>
<td>± 8</td>
</tr>
</tbody>
</table>

The *Disc-type coupling* (Figure 8.22), uses a fabric-reinforced neoprene disc to which is fastened, straddling each other, the fingers of the tubular shaft ends.

Up to 5° angular misalignment is possible with this configuration and if a greater angle is required only two fingers are used instead of three, the coupling then becoming more like a universal joint.

*Universal Joint* (Figure 8.23). This type of joint is used where the shaft coupling angle is up to 30° but low torque and speed are required - normally a maximum of 100 r/min.

As the joint operating angle increases it has a related drop in efficiency, to about 90% efficiency at an angle of 30°. Also this type of coupling does not transmit a constant angular velocity in its basic form as shown.
FLUID COUPLINGS

A fluid coupling has two basic parts, as shown in Figure 8.24, a pump (or impeller) and a turbine (or runner). There is no mechanical connection between the two halves, the power being transmitted by kinetic energy in the operating fluid.

As the input shaft rotates, oil flows from the pump section and strikes the vanes in the turbine section, causing the output shaft to rotate.

NOTE: As the speed of the output shaft increases, centrifugal force acts on the oil in the turbine and it becomes less effective. For this reason, a fluid coupling can never give 100% power transfer.

The output must always rotate slightly slower than the input (about 5% less) to maintain oil flow.

Fluid couplings are used where the prime mover needs to be started before the load is engaged. An example is an electric motor driving a conveyor belt. The motor can be started and obtain full speed before the belt starts to move.

An additional advantage is that if the conveyor belt should become overloaded, the fluid coupling will slip and avoid overloading the motor.
TORQUE CONVERTER

A torque converter operates on the same basic principle as a fluid coupling, but an additional component, called the stator, is added. The stator has curved blades which the oil strikes as it leaves the turbine. These blades turn the oil back in the direction of the pump rotation. Now the stream of oil is moving at a higher velocity which further increases through the pump. When the oil re-enters the turbine, its high velocity produces a greater torque.

Figure 8.25 shows the similarity between a fluid coupling and a torque converter.

![Fluid Coupling and Torque Converter Diagram](image)

Fig. 8.25

Note that the output torque of a torque converter is higher than the input torque, but the output speed is lower. Thus, a torque converter will provide twice the output torque of a fluid coupling but at half the output speed. As the need for high output torque decreases, the output speed increases and the stator becomes less effective. For this reason, the stator is mounted on a one-way clutch, which allows the stator to free wheel.

The graph in Figure 8.26 shows the relationship between torque multiplication and the ratio of input/output speeds. When starting, up to 2.5 torque multiplication is available. As the output speed increases, the torque multiplication decreases.

At a speed ratio of approximately 0.9, there is no torque multiplication and the torque converter functions as a fluid coupling.

NOTE: At high torque multiplication, system efficiency can be as low as 30%, with 70% of the input power changed into HEAT.

Maximum efficiency occurs when the input/output speed ratio is almost unity.
8.5 COUPLING SELECTION

The selection of the right kind of shaft coupling must depend on an accurate assessment of the duties which it must be capable of performing, followed by the necessary calculations to establish the correct size.

The information which is required before a specific size and type of coupling can be chosen includes factors such as:

- The power to be transmitted and the speed of the input shaft.
- The kind of duty. Does it include impact loads, sudden load changes?
- The amount of shaft misalignment present and its kind.
- Are there likely to be considerable shaft overloads?
- Must the coupling run in either direction?
- Must the coupling be removable without disturbing the driver and driven shafts?
- Is the coupling to resist heat, dust, chemicals etc?
- Can the coupling be fitted to a shaft or are there flanges present?
- Will the coupling be required to accommodate end float (or even end thrust)?

Table 8.1 displays some of the characteristics of a range of couplings, in terms of maximum misalignment and allowable end float.

<table>
<thead>
<tr>
<th>Coupling</th>
<th>Maximum misalignment</th>
<th>End float</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Axial (mm)</td>
<td>Angular (degrees)</td>
</tr>
<tr>
<td>Rubber Spider</td>
<td>0.25</td>
<td>1</td>
</tr>
<tr>
<td>Flexible Disc</td>
<td>0.5</td>
<td>1</td>
</tr>
<tr>
<td>Rubber Disc</td>
<td>0.125</td>
<td>0.2</td>
</tr>
<tr>
<td>Crown Pin</td>
<td>0.125</td>
<td>0.2</td>
</tr>
<tr>
<td>Chain</td>
<td>0.2 - 0.7</td>
<td>1</td>
</tr>
<tr>
<td>Gear</td>
<td>0.2 - 2</td>
<td>0.5</td>
</tr>
<tr>
<td>Spline</td>
<td>1 - 3</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Table 8.1
Ideally, you should obtain your own copies of coupling catalogues (consult "Power Transmission Equipment" in the Yellow Pages telephone directory). As an example of the extent of information available in catalogue form, the following pages contain extracts giving data on several types of coupling.

The Fenaflex extract that follows provides a worked example of the procedure for selecting an appropriate coupling, including allowances for shock loading. Note, however, the statement that where substantial shock exists, the manufacturers should be consulted so that a torsional analysis of the drive system may be undertaken.
Mechanical Drive Components

Fenaflex Selection

Details required for coupling selection are:

1. Type of driven machine and operating hours per day,
2. Speed and power absorbed by driven machine (if absorbed power is not known calculate on power rating of prime mover),
3. Diameter of shafts to be connected.

PROCEDURE
(a) Service Factor
Determine the required service factor from table 1.
(b) Design Power
Multiply the normal running power by the service factor. This gives the Design Power which is used as a basis for selecting the coupling.
(c) Coupling Size
By reading across from 1440 rev/min in table 3 the first power figure to exceed the required 63kW in step (b) is 76.1kW. The size of coupling is P100 Fenaflex.
(d) Bore Size
By referring to table 2 (page 2) it can be seen that both shaft diameters fall within the bore range available.

EXAMPLE
A Fenaflex High Speed coupling is required to connect a 75kW 2880 rev/min electric motor to a hammer mill which operates up to 8 hours per day. Both motor and mill shafts are 60mm diameter.
(a) Service Factor
From table 1, the service factor is 1.8.
(b) Design Power
Design Power = 75 × 1.8 = 135kW.
(c) Coupling Size
By reading across from 2880 rev/min in table 5 the first power figure to exceed the required 135kW in step (b) is 175kW. The size of High Speed coupling is 116X.
(d) Bore Size
By referring to table 4 (page 4) it can be seen that both shaft diameters fall within the bore range available.

TABLE 1: SERVICE FACTORS

<table>
<thead>
<tr>
<th>SPECIAL CASES</th>
<th>Type of Driving Unit</th>
<th>Operational hours per day</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Electric Motors</td>
<td>Steam Turbines</td>
</tr>
<tr>
<td></td>
<td></td>
<td>10 and under</td>
</tr>
<tr>
<td>CLASS 1</td>
<td>Agitators, Brewing machinery, Centrifugal compressors and pumps, Belt conveyors, Dynamos, Line shafts, Fans up to 7.5 kW, Blowers and exhausters (except positive displacement), Generators.</td>
<td>0.8</td>
</tr>
<tr>
<td>CLASS 2</td>
<td>Clay working machinery, General Machine tools, Paper mill beaters and winders, Rotary pumps, Rubber extruders, Rotary screens, Textile machinery, Marine propellers and Fans over 7.5 kW.</td>
<td>1.3</td>
</tr>
<tr>
<td>CLASS 3</td>
<td>Bucket elevators, Cooling tower fans, Piston compressors and pumps, Foundry machinery, Metal presses, Paper mill calenders, Hammer mills, Presses and pulp grinders, Rubber calendars, Pulverisers and Positive displacement blowers.</td>
<td>1.8</td>
</tr>
<tr>
<td>CLASS 4</td>
<td>Reciprocating conveyors, Gyrotry crushers, Mills (ball, pebble and rod), Rubber machinery (Banbury mixers and mills) and Vibratory screens.</td>
<td>2.3</td>
</tr>
</tbody>
</table>

* It is recommended that keys with top clearance are fitted on applications where load fluctuation is expected.
† Couplings for use with internal combustion engines may require special consideration, refer to separate publication "Fenaflex Flywheels and Couplings for Industrial Engines".

Table 2: Physical Characteristics and Dimensions

<table>
<thead>
<tr>
<th>Size</th>
<th>Max Speed (rpm)</th>
<th>Torque (Nm)</th>
<th>Moment of Inertia (kg-m^2)</th>
<th>Torque Sensitivity (N-m/°)</th>
<th>Maximum Misalignment (°)</th>
<th>Bush No.</th>
<th>A</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>J</th>
<th>T</th>
<th>L</th>
<th>M</th>
<th>Approx. Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>F40</td>
<td>4500</td>
<td>20</td>
<td>1.77</td>
<td>1208</td>
<td>8 - 25</td>
<td>105</td>
<td>82</td>
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<td>23</td>
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<tr>
<td>F80</td>
<td>3120</td>
<td>250</td>
<td>1.35</td>
<td>1820</td>
<td>15 - 20</td>
<td>161</td>
<td>144</td>
<td>32</td>
<td>32</td>
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<td>F90</td>
<td>3120</td>
<td>106</td>
<td>1.95</td>
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<td>161</td>
<td>144</td>
<td>32</td>
<td>32</td>
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<td>F100</td>
<td>2610</td>
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<td>2.33</td>
<td>2100</td>
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<td>230</td>
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<td>1.80</td>
<td>3430.02</td>
<td>25 - 75</td>
<td>116</td>
<td>2.18</td>
<td>0.30</td>
<td>30</td>
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<td>1881</td>
<td>6.94</td>
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<td>30</td>
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<td>3131</td>
<td>7.53</td>
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<td>25 - 75</td>
<td>116</td>
<td>2.18</td>
<td>0.30</td>
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<td>7839</td>
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<td>25 - 75</td>
<td>116</td>
<td>2.18</td>
<td>0.30</td>
<td>30</td>
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<td>F220</td>
<td>1100</td>
<td>11042</td>
<td>5.32</td>
<td>3430.02</td>
<td>25 - 75</td>
<td>116</td>
<td>2.18</td>
<td>0.30</td>
<td>30</td>
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<td>6.60</td>
<td>3430.02</td>
<td>25 - 75</td>
<td>116</td>
<td>2.18</td>
<td>0.30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
</tr>
</tbody>
</table>

Dimensions in Millimeters

* G is the amount by which clamping screws need to be withdrawn to release the latch.

J is the length of the threaded portion of the shaft.

T is the length of the threaded portion of the shaft.

For speeds below 100 rpm and intermediate speeds use normal torque ratings.
Fenaflex—Shaft to Shaft

To enable designers and engineers to gain maximum advantage from this versatile range of couplings it is presented as individual flanges. Fenaflex couplings embody all the desirable features of an ideal flexible coupling, including Taper-Lock fixing. They accommodate parallel misalignment up to 6mm, angular misalignment up to 4° and end float up to 8mm.

The excellent shock-absorbing properties of the rubber component reduce vibration and torsional oscillations.

Fenaflex tyres and elements are available in natural rubber compounds for use in ambient temperatures between minus 50°C and plus 50°C. Neoprene rubber compounds are used for temperatures of minus 15°C to plus 70°C or under adverse operating conditions—e.g. oil or grease contamination.

As contact between the metal flanges is eliminated Fenaflex couplings provide a certain degree of electrical isolation. However, for applications where fire-resistance and anti-static properties are required, neoprene (FRAS) tyres or elements should be used.

Any two flanges shown above may be combined with Flexible Tyre to provide a coupling most suited to the application (1) H Flange—Taper-Lock, (2) F Flange—Taper-Lock, (3) B Flange—Bored to size, (4) E Flange—Extended Bored, (5) M Flange—Mill Motor.

**SPACER**

S Flange – Spacer

To a FLEXIBLE TYRE coupling add the appropriate SPACER flange, these couplings are designed to accommodate standard distances between shaft ends and facilitate pump maintenance.

**HIGH SPEED**

X Flange (reversible) High Speed

Combining a F, H, B, E or M Flange with a FLEXIBLE ELEMENT and an X Flange enables couplings to be used up to speeds of 4800 rev/min.

**FLYWHEEL**

A range of Flexible Elements has been specially developed which fit directly to S.A.E. flywheels. For full details refer to separate booklet — "Fenaflex Flywheel Couplings for Industrial Engines."

### Chapter Eight

**Fenaflex—High Speed**

**X FLANGES (Steel) — REVERSIBLE TO PROVIDE H or F TAPER-LOCK® BUSH FITTINGS**

#### TABLE 4: PHYSICAL CHARACTERISTICS AND DIMENSIONS

<table>
<thead>
<tr>
<th>Size</th>
<th>Max Speed (rev/min)</th>
<th>Torque (Nm)</th>
<th>Moment of Inertia (kgm²)</th>
<th>Torsional Stiffness (Nm/°)</th>
<th>Max mis-alignment</th>
<th>Bush No.</th>
<th>A</th>
<th>E</th>
<th>H</th>
<th>J</th>
<th>S</th>
<th>T</th>
<th>U</th>
<th>V</th>
<th>Approx. Mass (kg)</th>
<th>X Flange Element</th>
</tr>
</thead>
<tbody>
<tr>
<td>87X</td>
<td>4800</td>
<td>239</td>
<td>716</td>
<td>0.04607</td>
<td>60</td>
<td>0.5</td>
<td>0.4</td>
<td>2012</td>
<td>50</td>
<td>240</td>
<td>32</td>
<td>26</td>
<td>42</td>
<td>96</td>
<td>89</td>
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† Mass is for X Flange with mid-range bore Taper-Lock® Bush. All couplings have an angular mis-alignment capacity up to 1 degree. Maximum torque figures should be regarded as short duration overload ratings for use in such circumstances as direct-on-line starting.

#### TABLE 5: POWER RATINGS (kW)

<table>
<thead>
<tr>
<th>Speed (rev/min)</th>
<th>87</th>
<th>96</th>
<th>116</th>
<th>131</th>
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For speeds below 100 rev/min and intermediate speeds use normal torque ratings.

Fire-resistant discs comply with the requirements of NCB 168

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<th>Weight (lbs)</th>
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Couplings in bold type are immediately available from
stockist stockists.

Also available with taper bush, details on request.

For power ratings, keyways and shafts of spares available, see website.

## Chapter Eight

### Disc Flexible Couplings

#### TABLE OF SERVICE FACTORS

<table>
<thead>
<tr>
<th>Power at 100 rev/min</th>
<th>Torque at 100 rev/min</th>
<th>Normal maximum speed</th>
<th>Maximum permissible speed</th>
<th>Malalignment</th>
<th>End float</th>
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<td>rev/min</td>
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<td>Axial</td>
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* Normal maximum speeds with 1° max angular misalignment, above these speeds a maximum of 0.5° is recommended.

** For ratings above these speeds, consult our Sets Technical Staff!

#### STANDARD KEYWORDS - for parallel keys

**BS 4225 Part 1:1972**

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<th>Pin core with nut and washer</th>
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### Mechanical Drive Components

**Chain Couplings**

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<th>Setting width (mm)</th>
<th>Weight (kg)</th>
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*M = Moulded cover  
A = Aluminium cover

Couplings in bold type are immediately available from factory stocks.

For power ratings, keyways and range of flanges available, see opposite.

### Crown Pin Flexible Couplings

#### Table of Service Factors

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<th>Machinery characteristics</th>
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#### Ratings

The ratings given below are based upon steady uniform load conditions. When the load is not steady, multiply the normal power absorbed in a driven machine by the relevant Service Factor.

Normal power at 100 rev/min = \( P \times 100 \) rev/min

#### Standard Keyways - for parallel keys

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<td>50+</td>
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<td>58+</td>
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<td>65+</td>
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<td>75+</td>
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<td>230+</td>
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<tr>
<td>260+</td>
<td>290</td>
</tr>
<tr>
<td>290+</td>
<td>330</td>
</tr>
</tbody>
</table>

#### Spare Parts

When ordering Spare Parts, suffixes indicated below is added to catalogue nos.

- e.g. Half-Body for half section of pt. PMQ3-10/P1

<table>
<thead>
<tr>
<th>No.</th>
<th>Part Description</th>
<th>Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Pin half-body</td>
<td>P3</td>
</tr>
<tr>
<td>2</td>
<td>Buffer half-body</td>
<td>P5</td>
</tr>
<tr>
<td>3</td>
<td>Buffer</td>
<td>P6</td>
</tr>
<tr>
<td>4</td>
<td>Washer*</td>
<td>P7</td>
</tr>
<tr>
<td>5</td>
<td>Pin, complete with nut and washer</td>
<td>P9</td>
</tr>
</tbody>
</table>

* Washer: Supplied with Crowning No. P2400 and above.

\(1\) Knewton = 1.34 HP
\(1\) Newton = 0.10197 kg

BRAKES AND CLUTCHES

9.1 BRAKE AND CLUTCH PRINCIPLES

The principle of operation of a friction clutch during engagement and of a friction brake during braking is to bring two members having relative motion to a state of no relative motion.

The operation of a clutch is therefore essentially the same as that of a brake; however, there are structural differences in the two units because of control requirements and the necessity for heat absorption or dissipation in brakes.

The function of a brake is to regulate the speed of a mechanism by transforming, through friction, the energy of a moving body into heat. For the brake to be effective, the rotating body must not only be retarded but the heat produced must be effectively dissipated. If the heat is not dissipated quickly enough, the brake operation can be affected and it will rapidly lose effectiveness.

As an indication of the magnitude of heat dissipation, consider a very common example:

A car engine develops perhaps 75 kW while it is accelerating from 0 to 60 km/h in 10 seconds. The car is then braked to bring it from 60 km/h to rest in 5 seconds. The power that must be absorbed by the brakes is in the order of 150 kW, or equivalent to 150 one-bar electric radiators. A hot situation!

For this reason a brake friction material is utilised that retains its characteristics at elevated temperatures, and the brake assembly is designed to provide maximum heat dissipation. Also the fluid used for hydraulically operating the brakes is chosen for its high chemical stability and high boiling point, because the brake temperature can exceed 250°C during rapid braking.

The function of a clutch is to connect a stationary machine part to a rotating part, to bring it up to a common speed and to transmit the design power with the minimum of slippage.

In certain cases, a clutch may serve as a safety device by slipping when the torque transmitted exceeds a certain value, thus preventing the breakage of parts in the transmission train or the overloading of the power source.

Friction clutches are based on the same principles as brakes, and their construction is also similar to that of a brake. In fact, every time a brake is locked it becomes a clutch.

Certain specialised clutch types exist. These clutches will be detailed later in this chapter but basically a clutch may be required to:

- operate in one direction of rotation only and disengage shafts in the reverse direction;
- operate when the input shaft has achieved a certain rotational speed - below the design speed the clutch disengages;
- possess variable drive characteristics remotely (electrically) controlled;
- provide a positive drive with mechanical locking, such that no slip can occur even in situations of excess torque transmission. This type of clutch cannot normally be engaged while the shafts are rotating.
9.2 BRAKE AND CLUTCH SPECIFICATIONS

When a brake or clutch is specified, consideration must be given to:

(a) adequate torque ability. If the torque rating is too low the unit will slip and heat and excessive wear will be caused;

(b) engagement and acceleration without shock. It is usually desirable to utilise a unit which will start or stop a machine element rotating with an acceptable acceleration/deceleration rate, to avoid shock loading. Every rotating body contains kinetic energy which cannot be obtained or lost instantaneously;

(c) provision for the unit to operate automatically. A brake used in a safety stopping situation (as on the cable drum of a lift) or a clutch that engages when the input shaft achieves a certain speed (as on a chain saw) are examples of automatically applied units;

(d) a unit for high-speed operation. It should contain low mass and be easy to balance if rotating. Out-of-balance components cause shaft deflection, excess bearing load and vibration, all of which can cause malfunction of a clutch or brake;

(e) automatic compensation for in-service wear, or at least the facility to adjust for wear without major disassembly of the unit;

(f) available space for the unit. This may be limited. Some units, for a given torque/ speed rating are much more compact than others;

(g) frequency of operation which has an effect on several factors. Units in continuous operation need a small travel to operate the engaging mechanism, a simple construction that assures long life without maintenance and adequate heat dissipation facilities;

(h) the ambient conditions affecting the choice of a unit. If the system is subjected to contamination with water or oil the clutch or brake must be composed of materials that are resistant. This is particularly the case where a friction material interface is used, in which the coefficient of friction can change remarkably with contamination.

Because the majority of brakes and clutches operate on the principle of a force applied between two materials to obtain a friction force, it is worth considering the types of materials used and their respective coefficients of friction.

Values of the coefficient of friction ($\mu$) for various common materials are given in Table 9.1. This table indicates the difference in coefficient between dry and oil-lubricated materials, and also the maximum allowable pressure by the friction material. Note that precise figures are not available because of the variance in the materials.

<table>
<thead>
<tr>
<th>MATERIALS IN CONTACT</th>
<th>$\mu$ DRY</th>
<th>$\mu$ WET</th>
<th>MAX PRESSURE (MPa)</th>
<th>MAX TEMP ºC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast iron on cast iron</td>
<td>0.2-0.15</td>
<td>0.1-0.05</td>
<td>1.5</td>
<td>300</td>
</tr>
<tr>
<td>Steel on cast iron</td>
<td>0.3-0.2</td>
<td>0.1-0.06</td>
<td>1.2</td>
<td>250</td>
</tr>
<tr>
<td>Cork on metal</td>
<td>0.35</td>
<td>0.25</td>
<td>0.08</td>
<td>100</td>
</tr>
<tr>
<td>Leather on metal</td>
<td>0.5-0.3</td>
<td>0.15-0.12</td>
<td>0.2</td>
<td>100</td>
</tr>
<tr>
<td>Moulded Asbestos on steel</td>
<td>0.5-0.2</td>
<td>0.15-0.08</td>
<td>0.5</td>
<td>250</td>
</tr>
</tbody>
</table>

Table 9.1

It should be noted that although the wet (lubricated) friction material has a lower coefficient than the same material in a dry state, the use of a lubricant assists in transferring heat away from the friction areas. Lubrication also provides a smoother action and reduces wear, and because the materials are lubricated, a higher force per given area (pressure) may be applied.
Choice of friction material depends on the application but preliminary specifications are determined by considering the brake/clutch rubbing speed and the allowable friction material pressures. Normally, the product of the friction pressure (MPa) and the rubbing speed (m/s) is kept between 1 and 3.

Values higher than $p \times v = 3$ can be used if the load is not applied continuously or if the heat dissipation characteristics are good.

It should be emphasised that with friction material usage, some of the laws of friction do not apply. One law states that friction force is independent of surface area, but the larger the surface area of a friction drive, the more area is available to dissipate the heat created.

Another law says that the coefficient of friction is independent of rubbing speed. With a friction material however, at higher rubbing speeds more heat is generated, which generally lowers the coefficient of friction. You have no doubt heard of the term brake fade, which is reduction in vehicle braking owing to the brake friction materials becoming hot.

Figure 9.1 shows the effect of an increase in rubbing speed at two loading pressures for a cork-on-steel clutch (automotive) running in oil.

Figure 9.1  Friction versus Rubbing Speed

Note that even though the friction material loading (pressure) is increased five times from 0.2 MPa to 1 MPa, the coefficient of friction only drops from 0.14 to 0.1.

As friction force = applied force x coefficient of friction, there are obvious advantages to using a higher applied pressure.

**9.3 BRAKE TYPES**

As stated in Part 9.1, there is a close similarity between many designs of brakes and clutches because both have basically the same function: to increase or decrease the rotation of one shaft relative to another. For this reason, in this part only, units which function solely as brakes will be included.

The block brake, shown in Figure 9.2, is the simplest form of braking device.
A block of friction material is forced against a rotating drum and the resultant friction force provides a retarding torque. The brake will operate if the drum is rotating in either direction and the braking force is relative to the force applied to the hand lever, within the limitations of the relationship between coefficient of friction and friction material pressure as detailed in Part 9.2.

One immediately apparent fault of the block brake is that it has a small surface area in contact and this causes localised heating during use.

To achieve a reasonable force onto the friction material the lever's velocity ratio needs to be large, which in turn means that as the friction material wears, the lever position or linkage must be adjusted.

To increase the friction force without excessive side loading, the friction material and drum may be shaped in the form of a tapered groove, shown in Figure 9.3.

Because of the wedging action of the friction block, the coefficient of friction of the system is effectively increased, where:

\[
\text{apparent value of } \mu = \frac{\mu}{\sin\theta + \mu\sin\theta}
\]

where \(\mu\) is the actual coefficient of friction and \(\theta\) is half the vee angle.

Another disadvantage of the single-block brake is that the force applied to the friction block is transferred directly to the bearings of the drum, producing an undesirable side loading.

The side loading problem is eliminated and the friction area increased in a commercial version of the block brake, shown in Figure 9.4 and called a double block brake.

In the unit shown in Figure 9.4, the brake blocks are pushed onto the drum by a spring. To release the brake, the lever is pulled down which then forces apart the two arms holding the brake blocks.
9.5

Fig. 9.4 Double Block Brake

If such a unit were used as a brake on the cable winding drum of a crane, the lever would be operated hydraulically or by an electric solenoid.

Note that the unit is fail-safe; if the lever is not operated, the brake is applied.

An *internal block brake* shown in Figure 9.5 is more commonly recognized as the system used on automotive drum brakes.

Fig. 9.5 Internal Block Brake

The two friction blocks (or brake shoes) are expanded by a hydraulic cylinder to be pushed against the inside of a brake drum. The brake shoes are retracted by springs and adjustment is provided to compensate for brake lining wear. In practice, the brakes are automatically adjusted during use by a mechanism which checks the lining-to-drum clearance each time the brakes are applied while the vehicle is moving in reverse.
A disc brake is another type of brake normally associated with cars, but the disc brake is also widely utilised in industry.

As shown in Figure 9.6, a cast steel disc is attached to the shaft to be braked and friction blocks are applied to either side. To ensure that both side loads are equal, the forces are applied to both blocks by the same mechanism, usually a hydraulic cylinder.

The major advantage of a disc brake is its ability to dissipate heat quickly, and it is self-adjusting for wear.

For high-power applications or where the brake unit must be kept relatively small in diameter, a multi-disc unit is available. It consists of a series of discs located on a shaft but free to move axially, and a series of interleaved friction blocks. A force applied to the end block is transmitted through the unit and the braking effect is shared by all the friction blocks.

A band brake shown in Figure 9.7 consists of a strip of flexible friction material which is wrapped around a drum. Braking action is achieved by pulling the band tight onto the drum, the applied force to the friction area being the difference between the tensions in either side of the band.

The band may be made of leather, canvas or reinforced fibre or of a friction material bonded onto a flexible steel backing. The band brake has many industrial applications but the most common is the system used in a
vehicle automatic gearbox. In this case, by braking rotating gear members with a series of band brakes the different gears are selected.

The simple band brake in Figure 9.7 has a low velocity ratio, unless the lever is made very long.

To improve the operating characteristics, a differential band brake (Figure 9.8) has the band anchored either side of the lever pivot. The differential action achieved greatly reduces the force that needs to be applied on the lever. If the drum rotates in reverse (anti-clockwise) the brake will tend to apply itself. This feature may be used to prevent a drive system running backwards.

An example of such use could be the drive to an elevating conveyor. If the drive should fail, it is essential to stop the conveyor running backwards as a result of its load. In this application the brake would be fitted at the drive motor where the torque is least and therefore requires a smaller brake than if it were placed on a low-speed high-torque shaft of the conveyor drive.

Special devices exist which are basically differential band brakes but called differential back stops. The devices allow free rotation in one direction and limited or zero rotation in the other direction.

As a comparison between the varying types of brakes, the following table (Table 9.2), contains several assumptions. The force ratio is the relationship between the input or operating force and the output friction force. A coefficient of friction of 0.3 has been used and the band brake used as a datum for comparison. The multi disc included has seven discs.

Lever end travel is for a normal lever operation with 5:1 linkage ratio. The travel distance is between operating the brake and the brake being clear of the rotating member. The comparison table is based on a range of braking systems applied to a hoist, such as on a lift or elevating device.

<table>
<thead>
<tr>
<th></th>
<th>DOUBLE BLOCK BRAKE</th>
<th>VEE BLOCK BRAKE</th>
<th>BAND</th>
<th>MULTI DISC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force ratio</td>
<td>20</td>
<td>9</td>
<td>1</td>
<td>3</td>
</tr>
<tr>
<td>Lever travel</td>
<td>8</td>
<td>18</td>
<td>70</td>
<td>5</td>
</tr>
<tr>
<td>Maximum power (kW)</td>
<td>2000</td>
<td>25</td>
<td>300</td>
<td>120</td>
</tr>
</tbody>
</table>

Table 9.2
9.4 CLUTCH TYPES

A large variety of clutch designs exist but it is only intended in this text to cover the more commonly occurring types.

Clutches can be classified into several distinct categories, namely:

- Positive
- Friction
- Fluid
- Electrical
- Overrun (or Sprag).

Some examples of positive clutches are shown in Figure 9.9. A positive clutch is designed to transmit torque with an absolute mechanical lock occurring so that no slip can occur.

![Fig. 9.9 Positive Clutches](image)

A positive clutch may be employed when a sudden starting action is not objectionable and when the inertia of the driven parts is small. In practice the clutch engagement speed should be limited to 10 r/min for square jaws and 150 r/min for angled jaws. Ideally, the angled jaw is used to minimise engagement shock but if the clutch has to drive in both directions the square jaw type is used.

![Fig. 9.10 Band Type Clutch](image)

One half of the clutch is keyed to one shaft and the other half of the clutch is on a sliding key fitted to the second shaft.

Friction clutches operate on a similar principle to a friction brake and many of the names used for friction clutches also apply to friction brakes.

The Band-type clutch (Figure 9.10) has one fixed end to the band and the other end is operated by a lever. The example shown in Figure 9.10 has adjustment on one end of the band.
The expanding shoe clutch (Figure 9.11) works in the opposite direction to a band clutch. Friction material is pushed out inside a ring (or drum) by some form of actuating mechanism.

![Fig. 9.11 Expanding Shoe Clutch](image)

The centrifugal clutch (Figure 9.12) is similar to the expanding shoe clutch but the friction linings are forced against the drum by centrifugal force. In the design shown in Figure 9.12, friction material is bonded to the shoe and as the driver increases in speed, the shoe is forced against the driven element and transmission occurs.

![Fig. 9.12 Centrifugal Clutch](image)

On some types of clutch a spring is added between the shoe and driver so that the clutch will not engage until a relatively high driver speed is obtained.
Generally a centrifugal clutch needs to rotate in excess of 300 r/min before it will engage. It requires an even higher speed before obtaining a positive (non-slip) drive.

The cone clutch (Figure 9.13) is often a metal-to-metal friction drive but friction material lining is added for high duty clutches. The angled cone means that for a given friction force the clutch operating force can be proportionally lower.

The cone clutch was the type used on early motor cars. It is still widely used in industrial applications.

A plate clutch is perhaps the most well-known because it is the type used in a motor vehicle as a link between the engine and the gearbox.

An industrial version is shown in Figure 9.14. It has two friction plates or discs which are squeezed between rotating members which are keyed to the shaft. To disengage the clutch, the collar (actuator) is moved to the right.

This type of clutch may be run dry but where high power is to be transmitted or where the clutch couples a high speed input shaft, the friction discs may be immersed in oil. This smoothes the clutch action but also assists to transfer heat from the friction material.

The automotive version of a plate clutch is shown in Figure 9.15. A thin steel disc is faced on either side with a friction material. The clutch is kept engaged by a series of springs and is released by pushing the release bearing towards the friction disc. The release bearing acts on a series of levers which effectively pull back the pressure springs.

Although clutches are normally designed to provide a positive coupling, situations do occur where a drive system must not exceed a specified torque.

The friction clutch is very adaptable to being used as a torque-limiting device, providing adequate heat dissipation is available. Versions of the disc type clutch are available which may be adjusted to transmit no more than a specified torque. Note, however, that the torque-limiting characteristics should only be used for short periods, as perhaps a safety aspect, because continual power wastage by a slipping friction clutch is very wasteful of energy.
Fig. 9.14 Plate Clutch

Fig. 9.15 Automotive Plate Clutch
Fluid couplings have already been discussed in Chapter 8 under the heading of "Fluid Couplings". It is worth re-stating that this type of device will couple two shafts and will do so without shock loading but can never provide a positive drive. Some slip always occurs.

The fluid coupling may be utilised as a clutch that is engaged or disengaged at will, by controlling the amount of oil within the body of the device. Provision can be made to circulate the oil through a fluid coupling to feed, for example, an external oil cooler.

If the oil level in the device is allowed to fall too low it will no longer transmit torque and will effectively disengage. The unit may even be used as a torque limiting device by only partially filling the coupling with oil. In practice, a fluid coupling is mainly used as a device which allows a drive motor to start up on low load and then the driven machine picks up speed via the coupling.

Fluid-operated clutches exist but these are normally friction type clutches which are activated by either oil or air under pressure. The clutch on a manual gearbox car, for instance, is normally hydraulically operated.

An electric clutch relies on the magnetisation of metallic particles. The type shown in Figure 9.16 consists of an inner rotating member which is contained by an outer rotatable member that is coupled to the output shaft. The gap between the two members contains ferro-magnetic particles.

When the coil is energised the particles tend to "solidify" and the unit transmits torque. Shear resistance of the particles depends on the coil current intensity. The unit shown mounts directly onto the end of an electric motor.

**Fig. 9.16 Electric Clutch**

The eddy-current clutch employs a magnetic field to couple the input drive to the output drive.
There is no mechanical connection between input and output and only the strength of the magnetic field determines the torque transmitted. Because there is a constant slip, similar to a fluid coupling, the energy loss is transformed into heat.

An over-run clutch is used to transmit torque in one direction of rotation but if the direction of rotation is reversed, the clutch slips with minimal reverse torque transmission.

The free-wheel mechanism on a bicycle drive could be classified as an over-run clutch but its operation is based on a ratchet drive which has a series of defined steps. Thus, in the drive mode, a small amount of free travel occurs before positive drive occurs.

Two types of over-run clutch are shown in Figure 9.17. Both types rely on a wedging action to provide a friction drive. In over-run operation the ball rotates but the sprag lifts slightly off the inner race to cause minimal frictional drag.

![Over-run Clutch Diagram](image)

(a) Roller type  
(b) Cam or Sprag type

Fig. 9.17 Over-run Clutch

A simple type of one-directional clutch can be made by using a spring as shown in Figure 9.18.

A tension spring is locked to the input shaft and the other end of the spring has an internal diameter that is a slide fit on the output shaft. When rotated in one direction the spring slips on the output shaft. If the direction of rotation is reversed, the spring tightens on the output shaft and a non-slip drive occurs.

This method is only acceptable for low speed, low torque applications where the relatively high friction losses are acceptable. For example, office machines or mechanisms that only operate occasionally.
Fig. 9.18 Uni-directional Clutch
BEARINGS

Wherever a gear, wheel or shaft turns within its supports, provision must be made to reduce friction and wear. A bearing must:

• support the moving part by resisting axial and radial loads and holding the moving part in alignment;

• reduce friction by providing either sliding contact (plain bearings) or rolling contact (anti-friction bearings). Usually bearings reduce friction by means of a lubricant in addition to their basic design;

• reduce wear or, if wear is to take place, have the facility to be easily replaced by a new bearing that will re-align the moving part.

A bearing is that part of a mechanical assembly which has to support or sustain another part which moves in sliding contact with it. The sliding may be linear or rotary.

There is no essential difference between a bearing and a bush. Conventionally, the latter term is reserved for bearings used as complete hollow cylinders and the former for those split into two parts for convenience of manufacture or ease of assembly. In the normal way of things, two parts rubbing together tend to wear each other away and, in so doing, generate heat.

Bearings may be classified into two main types - plain bearings (bushings) and anti-friction bearings.

10.1 PLAIN BEARINGS

Various types of plain bearing are shown in Figure 10.1.

A  is a solid bearing or sleeve, usually made of a bronze or sintered material and intended to support radial loads only.

B  is a split or wrapped bearing that is produced by rolling a strip of coated steel or bronze into a tube. It acts in the same way as a solid bearing for radial loads only, but is cheaper to produce than a solid bearing.

C  is a headed bearing that is used for radial and axial loads. It is usually made from a bronze or sintered material.

D  is a composite bearing. It has a steel outer casing and is lined with a bearing material such as a bronze or a plastic.

E  is a split bearing that may be constructed of wood, plastic, rubber or sintered metal. It may be made of cast iron. It is used where, for ease of assembly, a bearing carrier is split to insert a shaft.

F  is familiar as a split bearing used in an automotive engine on the big ends and main bearings. It is usually bronze or steel backed with a softer bearing surface of a high tin content alloy. Variations of this bearing will support both radial and axial loads.

G  is a solid bearing made for situations where oil lubrication is not possible. The bearing would be made of rubber, a fibre or plastic and the grooves allow the lubricant, normally water, to flow through the bearing.
Figure 10.2 shows various types of plain bearing that could be found in an automobile engine.

Bearing number 1, in the shape of a washer, is used for resisting thrust only.

Bearing 2 is of the wrapped type but the material join is on an angle to preserve the circular bore.
Chapter Ten

10.3 PLAIN BEARING MOUNTINGS

Plain bearings are usually installed in their supports with a press fit. This prevents the bearing from rotating in its mounting and also assures good heat transfer, which is important for bearing life.

If a bearing is purchased as a finished-size component it will appear to have an over-size bore, but when pressed into position the bore reduces marginally to the correct size. For this reason it is important to correctly tolerance any holes which are to receive bushes.

10.2 PLAIN BEARING MATERIALS

Bearing materials have traditionally been of a copper or tin base alloy, but nowadays a wide range of plastics are also used for bearings.

METALLIC BEARINGS

Bearing metals are generally soft, malleable and ductile, and consist of hard crystals of alloy held in the mass of soft metal. These crystals resist wear and therefore tend to stand out as the softer metal wears away. This differential wearing creates shallow depressions which hold the lubricant.

The soft metal adapts itself to the shaft and the harder crystals prevent unduly rapid wear. If the source of lubrication fails, the bearing metal is too soft to tear the shaft and melts at a low temperature - hence damage from seizure is slight and the bearing can be replaced.

WHITE METALS - TIN BASE AND LEAD BASE

In their simplest forms these are solid diecast bushes or half bearings, usually sufficiently accurately cast to be bolted up directly into a housing but requiring boring, reaming or burnishing of the bore to the required size.

Such bearings are not supplied with pre-finished bores since, due to their softness, they deform slightly when being bolted into position. The tin-base metals are the most useful, partly because they can be more accurately cast and partly because the lead-base metals are weaker, especially at higher temperatures.

HIGH LEAD AND STANDARD COPPER LEAD

Where there is full fluid lubrication, copper-lead bearings will stand up to the very heaviest loading without danger of mechanical failure. Owing to their hardness, however, (compared with white metals) an oil pad between journal and bearing is essential, and great care is needed when machining to ensure correct clearance. This clearance is generally greater than for white metal bearings to ensure a thicker pad of oil which in turn must be filtered and cooled.

In some cases, a thin coat of lead can be superimposed onto the bearing surface. This soft surface can readily deform to correct machining alignment errors and also to ease the "running in" phase.

These copper-lead bearings are most suitable for hardened steel shafts. They generally have a steel back to provide the extra strength needed to maintain a tight fit in the housing.

PHOSPHOR-BRONZE

This is used when the loading is very heavy. Because of its hardness, it demands a fully hardened journal or shaft. The material will make no concessions to misalignment and free lubrication should be used. When used for the heaviest possible loading, this metal is ideal.
The above bearing metals have been described in an order of increasing hardness. In general, always use the softest material which is acceptable. The lead and tin-base white metals, particularly the latter, are nearly perfect bearing materials.

They tolerate faulty alignment and dirty oil to a remarkable degree; they are perfectly satisfactory with soft-steel shafts as well as the hard-steel and cast-iron; they also keep cool when running. Their limitation is mechanical strength, particularly at high temperature.

**SINTERED BEARINGS**

These are produced by powder metallurgy, that is, the heat treatment of compressed powdered metals in a controlled gas atmosphere. These bearings are usually of the porous type, and are capable of holding sufficient oil either to safeguard in case of fluid failure or in some cases to be considered "self lubricating".

**NON-METALLIC BEARINGS**

Where conventional oil or grease lubrication is not possible, or is inconvenient, and in conditions where a fluid film cannot be maintained, as in oscillatory motion, plastic-faced or carbon/graphite bearings are being used increasingly in a very wide range of applications.

Most notable are automotive applications, but they are also used in the food and textile industries, chemical engineering, mechanical handling, aerospace, in conditions of high vacuum or high levels of ionising radiation, in roller skates and in the Concorde airliner. As dry rubbing bearings, they are limited by frictional heat build-up and low rotational speeds.

As well as being able to operate with minimal or no lubrication, non-metallic bearings can tolerate many process fluids unacceptable to conventional metal bearings; they can often perform well by using the process fluid as a lubricant, thereby running at much higher loads and speeds than would be possible as a dry bearing.

Furthermore, if they are run beyond their load-speed capacity so that overheating and failure occur, only bearing replacement is required as the metal shaft is not usually damaged.

Plastic bearings are usually lighter in weight than metal bearings and in the case of very large units this can greatly ease handling and assembly. While the plastic-faced, metal-backed type of bearing is usually prefinished and should not be machined, many types of homogeneous plastics and graphite bearings are readily machinable; other plastics lend themselves to wrap-around constructions that simplify installation in inaccessible positions.

In unlubricated bearings, continuous wear takes place, and a hard metal shaft is usually preferred since this minimises the take-up of work shaft particles in the softer bearing surface. It is possible however, to run an acetal resin-nylon bearing/shaft combination satisfactorily. A good surface finish on the shaft is desirable to minimise wear.

Because there is no corrosion protection from lubricating oil, the shaft needs to be of stainless steel or to have a corrosion-protective coating, if moisture or other corrosive agents are present. Otherwise, corrosion products will increase surface roughness and wear.

A disadvantage of plastics, from the bearing viewpoint, is their low thermal conductivity as compared with metals, which limits the dissipation of heat and hence restricts their load-speed capacity. Because of this, and also because most unfilled plastics have a high coefficient of thermal expansion, it is usually recommended that the thickness of solid plastics bearings should be made as small as practically possible. In some
proprietary bearings, low thermal conductivity and high expansion have been offset partly by the use of suitable fillers, or by using the plastics material (possibly filled) as a thin surface layer on a metal backing.

High thermal expansion coefficients and the capacity of some plastics to absorb water and other liquids demand the provision of much larger running clearances than are usual for metal bearings - probably of the order of 0.005 mm/mm of shaft dimensions.

NYLONS AND ACETALS

Nylons, acetal resins and PTFE (polytetrafluoroethylene) with or without fillers are the most commonly used thermo-plastics in dry bearing applications. Solid unfilled nylon and acetal mouldings can be produced in quantity at low cost for machining into bushes, thrust washers, etc. and are widely used for lightly loaded applications. Their load-carrying capacities are similar.

Unfilled nylons are preferred for high abrasion resistance in dirty conditions, and for silent running.

The acetals have better anti-friction properties, in particular, low static friction. Unlike nylon, they are not significantly absorbent.

Glass-filled nylons have lower thermal expansions and higher creep strength and can be used at higher loads and working temperatures, but with some loss of resilience.

An unusual application for glass-filled nylon is in Dowty Seals' Drylon bearing - a simple snap-together assembly of glass-filled nylon inner and outer races co-operating with stainless steel balls. Originally developed for roller skates wheels, Drylon bearings are now being applied to conveyor rollers and as steering column bearings.

POLYURETHANE BUSHES

These are being used for slow speed oscillatory and rotational movements in dirty environments where exceptional resistance to abrasion is required - for instance, Prescollan bushes are employed on the paddle arms of concrete mixers. As well as high abrasion resistance, polyurethane offers low friction coupled with sufficient resilience to accommodate misalignment.

PTFE BEARINGS

The primary attractions of polytetrafluoroethylene (PTFE) are its anti-friction properties, and particularly its freedom from stick-slip motion, and its outstanding chemical resistance over a very wide temperature range, from about -200°C to over 250°C. However, unfilled PTFE has poor mechanical strength and wear resistance, and is generally only suited to light duties, although it has been successfully applied in very low speed sliding bearings carrying heavy loads.

Possessing the anti-friction characteristics and wide temperature range of PTFE, Glacier DU bearings perhaps do not fall strictly within the category of non-metallic bearings, as they are made from a composite strip comprising a backing layer of steel, a middle layer of porous bronze solidly impregnated with a mixture of PTFE and lead, and a surface layer of the same PTFE-lead mixture.

Because of this construction, they have high compressive strength, good heat conductivity and excellent resistance to wear at a high pressure-velocity rating extending to higher rubbing speeds than most other dry rubbing bearings will tolerate.
They work well in mildly corrosive environments, but are not recommended for seriously corrosive conditions. Standard Glacier Dualign spherical bearings and rod-end joints, cadmium plated, operate satisfactorily in synthetic hydraulic fluids such as Skydrol, lubricating oils and antifreeze compounds. These are widely used in control linkages, automobile suspensions, and machinery of various sorts.

**CARBON/GRAPHITE BEARINGS**

For high temperature applications beyond the capacity of plastics, carbon/graphite and carbon/metal bearings are available, several capable of operating as high as 500°C.

In contrast to PTFE bearings, the dry friction coefficient decreases with load. Although static load capacity is not high, for a given wear rate, unlubricated carbon bearings can operate at higher speeds than most dry bearings, other than the metal-backed PTFE/lead type. As with the latter high performance bearings, good alignment is essential. Because carbon is highly inert, bearings of carbon/graphite can be lubricated by water, petrol or solvents.

Typical applications for carbon dry bearings include furnace and boiler plant bearings, food-conveyor bearings where free oil or grease is prohibited, veneer and paper drying machines; and in wet operations, pump rotor bearings immersed in circulating fuel, bearings in bottle cleansing plants subjected to cleansing fluid, and immersed impeller bearings in automatic washing machines.

**FLEXIBLE RUBBER BEARINGS**

A totally different non-metallic bearing for oscillatory movements, most widely used in the automotive industries but also in industrial machinery, is the flexible rubber bush such as the Metalastik family of rubber-bonded-to-metal units; such a bearing accommodates angular movements by the torsional distortion of rubber.

The virtue of this type of bearing is that it can be designed to provide a specified degree of flexibility in all directions, and it suppresses noise and vibration.

**THERMOSET-FABRIC BEARINGS**

Laminated cotton or asbestos fabric impregnated with a thermo-setting resin can be moulded or machined for use as dry bearings, and they are also widely used in lubricated applications, both with normal types of oil or grease lubricant, or using the process fluid or water as coolant and lubricant.

They are resistant to petrol, oil and grease, and to most dilute acids and alkalis. In some of these materials, swelling due to moisture absorption is considerable. They need full support, are fairly sensitive to misalignment and have low thermal conductivity.

<table>
<thead>
<tr>
<th></th>
<th>COEFFICIENT OF FRICTION (ON STEEL)</th>
<th>MAXIMUM BEARING PRESSURE (MPa)</th>
<th>MAXIMUM BEARING TEMP (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>White Metal</td>
<td>0.2</td>
<td>15</td>
<td>150</td>
</tr>
<tr>
<td>Bronze</td>
<td>0.35</td>
<td>40</td>
<td>300</td>
</tr>
<tr>
<td>Rubber</td>
<td>0.4</td>
<td>15</td>
<td>150</td>
</tr>
<tr>
<td>Laminated Fabric</td>
<td>0.3</td>
<td>70</td>
<td>100</td>
</tr>
<tr>
<td>Nylon</td>
<td>0.2</td>
<td>15</td>
<td>100</td>
</tr>
<tr>
<td>PTFE</td>
<td>0.04</td>
<td>8</td>
<td>270</td>
</tr>
<tr>
<td>Sintered Metal</td>
<td>0.15</td>
<td>60</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 10.1 Summary of Plain Bearing Materials
10.3 ANTI-FRICTION BEARINGS

Anti-friction bearings operate on the principle of substituting rolling for sliding friction and thus offer considerable advantages, particularly in power-saving through reduced bearing torques.

The rolling bearing usually comprises four main components, the inner ring, outer ring, rolling elements and separating cage. The inner and outer rings have suitably formed running tracks to support and guide the particular rolling elements which may be either balls or rollers. The separating cage spaces the rolling elements and prevents them from rubbing against each other at high relative velocity.

TYPES

Many types of rolling bearings have been developed and in order for draftspeople to choose the correct type for any new application, it is helpful if they have some knowledge of the more popular varieties.

Correct selection both for bearing type and size at an early design stage is desirable to avoid possible costly modifications at a later date. In this respect it is worth remembering that all bearing suppliers offer a free technical advisory service.

Fig. 10.3 Bearing Classification

Basically a bearing may be required to accommodate either radial or axial (thrust) loadings or both together.
Figure 10.3 provides a general classification of bearings in terms of their application, and indicates whether the bearing consists of balls or rollers.

**THE SINGLE ROW RIGID BALL JOURNAL**

This is the most popular type of anti-friction bearing due to its relatively low cost and versatile operating characteristics. While primarily designed for radial loading, the ball bearing also has some axial load capacity and is often used for shaft location duty.

Figure 10.4 shows a simple method of mounting a rotating shaft on two ball journal bearings.

![Fig. 10.4 Bearing Mounting](image)

Note that the shaft is located endways in relation to the fixed housing by the bearing which has both rings clamped. The other bearing has its outer ring left free endways and this is made a sliding fit to accommodate axial differences in thermal expansion of shaft and housing and allow correct positioning without the danger of axially preloading the two bearings.

It is necessary to make the rings which rotate relative to the applied load an interference fit on their seatings to avoid creep and wear, but rings which are stationary with respect to the load can be made a sliding fit on their seatings since in this case there should not be any tendency for creep to occur.

Where a shaft is located by one ball journal bearing, some free end float in the mounted assembly is bound to exist due to the internal geometry and clearances which are necessary for correct functioning of the bearings during service.

The axial load capacity of a ball bearing varies with the amount of diametrical slackness between the tracks and balls. Thus, bearings having minimum clearance should be used where the axial loading is very light and excessive end movement is undesirable. Similarly, bearings having increased clearance are used for heavier axial loads where end-float is not critical.

*Angular contact ball-bearings* have a large initial diametrical slackness and are designed for heavier axial loading than equivalent size ball journal bearings. They can also deal with radial loads but will only take axial
loading in one direction. It is therefore usual to adjust one bearing against another in order to locate the shaft in both axial directions and ensure correct ball tracking.

The bearings may be adjusted by means of shims as shown in Figure 10.5, which shows a rotating shaft application where it is necessary to make the outer rings a sliding fit. When adjusting these bearings it is essential to allow sufficient end float in the assembly to allow for temperature differences, particularly if high speeds are involved.

Single angular contact bearing mountings are only permissible where the axial load is constantly applied in one direction and cannot be exceeded by the radial load.

The *duplex bearing* (Figure 10.6) has four ball tracks and one split ring. It is thus designed to take axial loads in either direction while maintaining the same load capacity as an angular contact ball-bearing.

The radial load should never exceed the axial load and this type of bearing should not run either unloaded or at very high speeds, otherwise ball contact on more than two tracks will cause skidding and wear.

Figure 10.7 shows a *ball thrust bearing* which can deal with heavy axial loads where slow or occasional rotation is involved, such as in turntable or crane hook applications.
Bearings

This bearing is not suitable for high speeds and cannot take radial loading. Its popularity is, therefore, declining somewhat in favour of the angular contact bearing.

The spherical seating ring shown at the base is optional and allows for mounting errors in alignment.

ROLLER-BEARINGS

Roller-bearings have a greater load capacity than equivalent size ball-bearings because of the increased load area obtained at the points of contact between rolling elements and rings.

The standard pattern roller journal deals with heavy radial loads; Figure 10.8 shows a ball and roller bearing mounting in which the standard roller journal is used at the heavily loaded position, while the ball journal deals with the lighter radial load and locates the shaft axially.

This scheme can be used where either the inner or outer rings rotate in relation to the load, since both rings can be made interference fits on their seatings to avoid creep.

Radial roller bearings are available with various lip patterns to suit differing mounting requirements, and Figure 10.9 shows two lipped pattern bearings supporting and locating a heavily loaded shaft.

The guiding and locating lips are capable of withstanding light and intermittent axial loads on normal location duty, but heavy and continuous axial loading would soon result in lubrication breakdown and failure at the roller end-lip facing rubbing contact area. For this reason it is important to leave a small clearance between the lip faces and rollers during assembly to avoid binding and allow for shaft expansion.
The needle roller bearing shown in Figure 10.10 is fitted with a full complement of rollers and is particularly suited for heavy radial loading and slow or oscillating motions.

Speed is somewhat limited by lubricant space restriction and inter-roller rubbing, but increased speeds can be obtained by introducing a separating cage which provides improved roller guidance and increased lubricant space at the expense of reducing the radial load capacity.

Bearings with long rollers are more susceptible to mounting alignment errors and shaft deflection. Needle bearings should therefore only be used where alignment can be maintained, otherwise early failures owing to roller end-loading and skewing may result.

Tapered roller bearings (Figure 10.11) have radial and axial load capacity, and they are similar to angular contact ball-bearings in that one must be adjusted against another to eliminate unwanted shaft end-movement and provide axial location in both directions.

Mounting procedure is similar to that for the angular contact ball-bearings, but lubrication under heavy loads and high speeds is more critical owing to the sliding which occurs between the inner ring lip and the roller end contact points under load.

The barrel roller bearing (Figure 10.12) is relatively costly to produce because of the more complicated internal shapes involved, but it does offer the advantage of being a self-aligning bearing while being capable of dealing with radial and axial loads.
Bearings

It is predominantly a radial type of bearing and is used for dealing with both shaft deflection and initial housing misalignment on heavily loaded applications.

![Barrel Roller Bearing](image)

**Fig. 10.12 Barrel Roller Bearing**

**ALIGNMENT**

When accurate alignment cannot be guaranteed or maintained, bearings having an aligning feature are essential to avoid unfair, heavy stresses being set up between the rolling elements and tracks.

![Double Row Self Aligning](image)

The *double row self-aligning ball-bearing* (Figure 10.13) can accommodate both housing misalignment and shaft deflection.

However, it has only moderate radial load capacity and will only support very light axial loads due to the outer track geometry.

![Double Row Self Aligning](image)

**Fig. 10.13 Double Row Self Aligning**

Single-row ball and roller journal bearings and also ball thrust bearings can be obtained with spherically seated shell housings for alignment purposes. The externally aligned bearings will not align themselves satisfactorily, particularly under load, and when using these to accommodate initial housing misalignment, it is necessary to check the squareness of the rings with the axis of rotation during assembly.

Because of their inability to align themselves under load they are not generally recommended for dealing with heavy shaft deflections.

**BEARING SELECTION**

When selecting bearings for a new application, it is essential to calculate or estimate the maximum working loads and speeds. Then, knowing these and the degree of reliability or minimum life required, the bearing size can be determined by making reference to the manufacturer's catalogue.
Chapter Ten

Having determined that the selected bearing is suitable for both load and speed, the next question which arises is whether standard or extra precision limits are required. This will depend on the application, and it is for the user to decide whether increased accuracy justifies the extra cost.

Diametrical clearance between rolling elements and tracks has already been mentioned in connection with the axial load capacity of ball-bearings, but it is essential for both ball and roller journal type bearings that a small clearance is provided to allow for inner ring expansion and outer ring contraction owing to seating limits and/or temperature effects.

Bearings having insufficient diametrical clearance are liable to premature failure due to tightness, and it is therefore important to choose the correct clearance from the three or four standard grades which are generally available.

Bearings having the minimum amount of slackness are used where minimum shake is a prime requirement and there is not likely to be any loss of clearance due to fitting or temperature conditions, whereas bearings with greater than normal slackness are used where both rings are made interference fits and/or where temperature must be considered.

A rational bearing selection can be made by considering the following points:

- Load and direction of load.
- Speed.
- Operating temperature.
- Tolerances.
- Axial movement of floating bearings.
- Compensation of misalignment.
- Radial clearance.
- Separability to simplify mounting.
- Tapered bore and sleeve mounting.

The various bearing types described are available in numerous sizes and differing series to suit various applications. Selection is made from manufacturers' catalogues and bearings are standardised as follows:

- BS 292  Ball and roller bearing
- BS 1131  Plain bearings
- BS 4480  Plain metric bearings of metal and plastic
- BS 5512  Calculation of rolling bearings

10.4 BEARING LUBRICATION

In recent years a great deal of research has been undertaken into effective lubrication - in fact, the name tribologists is now applied to lubrication engineers. Lubrication of bearings is used to reduce friction and wear and in some cases to carry away the heat produced owing to bearing friction.

Four states of friction or lubrication may exist in a bearing:

1. **Dry friction** is when no lubricant is present between the contacting surfaces. Under conditions of light loading, the bearing will operate but with a high coefficient of friction. As the bearing load increases, a point is reached where metal-to-metal molecular contact causes the adjacent surfaces to "tear" each other, the process being caused by a temporary welding action.
2. **Boundary lubrication** is when the lubricant is in a very thin film between the sliding surfaces. It occurs in a machine bearing where the supply of forced lubricant fails, when a machine is started after standing idle for some hours, or in machines where lubrication of plain bearings is carried out on an intermittent basis, such as giving a squirt of oil from an oil can once a month. In such a lubrication situation the most important property of the oil must be the ability to attach itself to the two surfaces to provide a protective coating.

3. **Full-film lubrication** separates the two surfaces by a lubricant under pressure. The lubricant pressure is formed by a combination of adequate oil viscosity and the relative movement of the surfaces which forms a "hydrodynamic" pad under the shaft which ensures a constant supporting film of lubricant. For this type of lubrication, adequate oil must be available and oil type is important because, although a thick oil forms a good hydrodynamic pad, it also consumes more power.

4. **Rolling friction** occurs with anti-friction bearings, where bearing operation is not based in a sliding action but on the ease with which a ball or roller rolls along a smooth surface. Under load, a ball or roller indents slightly into its rolling surface and force is required to cause it to roll.

Figure 10.14 indicates the four types of friction situations previously discussed.

![Fig. 10.14 Bearing Friction Types](image)

Note that the position of the rotating shaft varies between dry friction, boundary lubrication and full-film lubrication. The shaft is loaded vertically downwards in each case.

As the shaft begins to rotate, as shown in boundary lubrication, it tries to climb the side of the bearing and at low speeds achieves the position shown.

At higher speeds, and with an adequate supply of lubricant, the rotating shaft causes an increase in the hydrodynamic pressure which tends to force the shaft into the position shown as full-film lubrication.

For anti-friction bearings, all ball and roller bearings (except spherical roller thrust bearings) can be lubricated with either oil or grease. Grease is, in fact, an oil which contains "scraps" to act as a thickener. Grease assists in forming an efficient seal to a bearing and oil is therefore only used where some special requirements make it desirable, for instance, when rotational speed is too high for grease to be effective.

If adjacent machine components have to be lubricated, such as gears and bearings, oil is used. Spherical thrust bearings must only be lubricated with oil. Grease should only be used at very low speeds.
10.5 LUBRICATION METHODS

Lubricating devices vary from a simple oil hole to a comprehensive circulating system incorporating a pump, filter, oil cooler and over-temperature or under-pressure warning devices.

Oil lubricating systems can be broadly classified into:

1. Hand oiling.
2. Regulated low-pressure, non-mechanical total loss systems.
3. Mechanically regulated high-pressure total loss feeds.
4. Circulating systems.

The first three methods are all *total loss* - the oil is only used once.

Hand oiling is, at its best, a hit-and-miss business, depending on the memory and conscientiousness of the operator. The simplest form of hand oiler comprises a hole drilled in a bearing. The entry is often chamfered and the entry area painted red. Because an open hole also collects dirt, the usual practice is to fit some form of cap or closure, as shown in Figure 10.15.

![Fig. 10.15 Oiling Points](image1)

If grease is to be used, a range of grease nipples is available, a selection being shown in Figure 10.16.

![Fig. 10.16 Grease Nipples](image2)
Low-pressure, non-mechanical systems consist of an oil reservoir with some form of supply method to the bearing. These supply methods are:

- by absorbent wick;
- by a constant drip system.

Oil flow to the bearing is at an almost constant rate, whether the shaft is rotating or not, but the oil reservoir means that there is little likelihood of the bearing running dry.

Figure 10.17 shows several types of constant-feed oilers.

The sight feed oiler has an adjustment to control the oil drip rate, and a transparent body so that oil level can be checked. A toggle mechanism in the top enables the oil flow to be stopped manually during the periods when lubrication is not required.

The wick oiler has a similar action to the sight feed oiler but the former is adjusted by raising or lowering the wick. The wick acts as a siphon and the longer the wick length outside the reservoir, the greater the oil flow. The screw-down grease cup and the spring-loaded grease cup are two methods of supplying grease from a reservoir.

In the case of a machine with many bearings within close proximity, it is not only time-consuming to lubricate each bearing separately but also there is the risk of one lubrication point being left unattended. In such cases a centralised pump is used with lines feeding oil (or grease) to the individual bearings.

To lubricate the machine, the pump is operated by hand or by foot and a set volume of lubricant is dispensed to the machine, at a delivery pressure of 15 MPa and a displacement of 0.02 mL for each 1000 mm² of bearing surface area.
Mechanical high-pressure total loss feeds

For automatic operation, the pump is stroked at a low speed by a cam that is driven from the machine to be lubricated. In such cases, a large reservoir is coupled to the pump to avoid having to replenish the pump with lubricant every day.

On more precise systems, a separate flow control valve is fitted on each line from the pump so that the lubricant supplied to each bearing may be individually adjusted. Figure 10.18 shows one such system with drive to the pump via a lever and ratchet. Lubricant flow is adjusted at the base of each sight glass and the actual lubricant flow can be seen in the sight glass.

![Lubricating Pump](image)

Fig. 10.18

An oil circulating system is the only practicable method of lubrication for bearings of high performance, for this method alone provides an adequate supply of lubricant to ensure that the hydrodynamic supply does not break down owing to oil starvation caused by end leakage from the bearings.

Only this system can provide the necessary surplus flow of oil to carry away excess heat. Also, the constant flow of oil ensures that any loose metallic particles are flushed from the bearing. For these reasons the inclusion of a filter and possibly a heat exchanger is desirable in a circulating system.

Probably the most familiar example of a circulating system is the method of lubricating a four-stroke car, where the oil not only lubricates but assists with heat transfer, sealing and flushing away the products of combustion.

**10.6 TEMPERATURE RISE IN A PLAIN BEARING**

The temperature rise in a journal (plain bearing) is the product of the friction force and the peripheral speed:

$$H_g = \frac{\mu W \pi DN}{60}$$

Where:

- $H_g$ = heat generated (watts)
- $\mu$ = journal coefficient of friction
- $W$ = total bearing radial load (newtons)
- $D$ = journal diameter (metres)
- $N$ = journal speed (r/min)
The only variable that cannot be easily measured is the coefficient of friction, since it varies widely for differing conditions. However, experimentation has shown that there is a relationship between the coefficient of friction and the oil viscosity, the bearing speed and the bearing pressure.

The coefficient of friction may be estimated from the McKee formula:

\[
\mu = 0.326 \times \left[ \frac{ZN}{p} \right] \frac{D}{C} + k
\]

Where:
- \( Z \) = absolute viscosity of the lubricant (kg/ms)
- \( N \) = journal speed (r/min)
- \( p \) = bearing pressure (N/m\(^2\))
- \( D \) = journal diameter (metres)
- \( C \) = clearance between the journal and the bearing (m)
- \( k \) = a constant, based on the bearing length/diameter ratio

The \( D/C \) ratio is typically in the region of 1000.

The \( k \) constant is in the region of 0.002 for most applications, but may be determined more precisely from the graph shown in Figure 10.19. \( L \) and \( D \) represent the bearing length and diameter.

![Fig. 10.19 Determination of \( k \) Factor](image)

Practical operating values for the \( \frac{ZN}{p} \) ratio are shown in Table 10.2.
### EXAMPLE

A 75 mm long bearing of 75 mm diameter supports a load of 12 kN on a journal rotating at 1800 r/min. Assuming a $D/C$ ratio of 1000 and using an oil of viscosity 0.01 kg/ms, determine the coefficient of friction and the heat generated.

From Figure 10.19, using $L/D = 1$, then $k = 0.002$

Bearing pressure $p = \frac{\text{bearing load}}{\text{bearing area}}$

$$p = \frac{12000}{0.075 \times 0.075} = 2.13 \text{ MPa}$$

Using the McKee equation:

$$\mu = \left(0.326 \times \frac{0.01 \times 1800}{2.13 \times 10^6} \times 1000\right) + 0.002$$

$$\mu = 0.005$$

Using the coefficient of friction in the original formula:

$$H_d = \frac{\mu W \pi DN}{60}$$

$$H_d = \frac{0.005 \times 12000 \times \pi \times 0.075 \times 1800}{60}$$

$$H_d = 424 \text{ watts}$$

### HEAT DISSIPATED FROM A BEARING

The heat generated in a plain bearing due to friction may be dissipated by a flow of lubricant through the bearing - as in a car engine. Where heat loss is only through the bearing material into the surrounding air, the Lasche formula may be used to estimate heat dissipation.

$$H_d = \frac{(\Delta T + 18)^2}{K} \times (L D)$$

Where:

- $H_d$ = heat dissipated (watts)
- $\Delta T$ = the difference between the bearing surface temperature $T_B$ and the temperature $T_A$ of the surrounding air, or $(T_B - T_A)$
- $K$ = a constant (from 0.273°C m²/W for a heavy bearing that is well ventilated to 0.484°C m²/W for a light bearing in still air)
For example, let’s use the parameters from the previous bearing example to determine the probable surface temperature. For temperature equilibrium, the heat gain must equal the heat dissipated.

\[ H_g = H_d = 424 = \frac{(\Delta T + 18)^2}{0.484}(0.075\times0.075) \]
\[ \Delta T = 173^\circ C \]
\[ T_B = T + T_A = 173 + 20 = 193^\circ C \]

**10.7 BEARING SELECTION FROM A CATALOGUE**

Factors to be considered in the choice of a bearing are:

- axial load conditions
- the combination of axial and radial loads, called the equivalent load
- bearing load may be variable so there is the need to determine the mean or representative load
- an important factor is bearing life, in terms of hours of operation or number of revolutions, which in itself is an indication of the fatigue rating of the bearing.

Load ratings are considered on a *static* basis (non rotating) or a *dynamic* basis. Both values are provided in bearing catalogues, where:

\[ C_o = \text{static load rating} \]
\[ C = \text{dynamic load rating} \]

**BEARING LOAD RATING**

A bearing may be subjected to axial or radial loads, or a combination of axial and radial.

The *equivalent load* is defined as:

"the constant static load which, if applied to a bearing with a rotating inner ring and a stationary outer ring, would give the same life as that which the bearing would attain under the actual conditions of load and rotation"

The expression used to define the equivalent load, which is applicable to radial and angular contact bearings, and roller bearings, is shown here:

\[ P = xF_r + yF_a \]

Where:

\[ P = \text{equivalent load} \]
\[ F_r = \text{radial load} \]
\[ F_a = \text{axial load} \]
\[ x = \text{radial load factor (0.35 - 1). Where the outer ring rotates, the radial factor is multiplied by a factor of 1.2} \]
\[ y = \text{axial load factor (0.25 - 0.8)} \]

The values of \( x \) and \( y \) are available from bearing catalogues.
Because the $x$ and $y$ factors are generally less than unity, it is possible to obtain a combination where, if the axial component is much smaller than the radial component, then the radial component without the radial factor $x$ is the larger of the two radial loads $P$.

So, the value of the equivalent load is taken as $xF_r + yF_a$ or just $F_r$, whichever is the greater.

$P$ is always greater than (or equal to) $F_r$

**BEARING LIFE**

Extensive testing of rolling bearings and subsequent statistical analysis have shown that the life and load of a bearing are related in the expression:

$$L = \left( \frac{C}{P} \right)^n$$

Where:

- $L$ = life in millions or revolutions
- $C$ = basic dynamic load rating
- $P$ = equivalent bearing load
- $n$ = 3 for ball bearings, 10/3 for roller bearings

$\frac{C}{P}$ is known as the *loading ratio*

To express the bearing life in hours, the bearing speed must be incorporated:

$$L_H = \frac{10^6 \left( \frac{C}{P} \right)^n}{60N}$$

Where:

- $L_H$ = life in hours
- $N$ = revolutions per minute

**NOTE:** Whereas the *static capacity* of a bearing is based on the permissible magnitude of permanent deformation, the *dynamic capacity* is based on the fatigue life of the material. When a bearing rotates, it will eventually fail by surface fatigue due to repeated stresses at the contact surfaces between balls and races.

The life of a bearing is actually the life that 90% of a group of bearings would reach, or exceed.

At the end of this chapter are some extracts from a bearing catalogue, to be used for the following examples.

**EXAMPLE 1**

A DGBB (deep groove ball bearing) is required to carry a radial load of 2182 N and an axial load of 445 N at 1600 r/min. If the nominal working life of $10^4$ hours is required, investigate a suitable bearing load rating.

$$L_H = \frac{10^6 \left( \frac{C}{P} \right)^n}{60N}$$
and \( P = xF_r + yF_a \) where \( x = 1 \) and \( y = 0 \)

First, substitute in the above equation to determine the equivalent load:

\[
P = (1 \times 2182) + (0 \times 445) = 2.182 \text{ kN}
\]

Now use the bearing life formula:

\[
L_H = \frac{10^6 (C)^{y}}{60N}
\]

\[
C = 21.52 \text{ kN}
\]

**EXAMPLE 2**

Determine the nominal life of a DGBB No. 6308 which carries a radial load of 2750 N at 800 r/min.

Referring to Figure 10.21, the basic dynamic load rating for a 6308 bearing is 31.5 kN. Substituting in the bearing formula:

\[
L_H = \frac{10^6 (C)^{y}}{60N}
\]

\[
= \frac{10^6 (31500)^{3}}{60 \times 800}
\]

\[
= 31310 \text{ hours}
\]

**EXAMPLE 3**

If, in addition to the radial load in Example 2, an axial load of 1700 N is applied, what life could be expected from the bearing?

Again, referring to the bearing catalogue (Figure 10.21). To determine \( P \) requires a value for \( x \) and \( y \) but these factors depend on the relationship between \( F_a, C_o \) and \( e \).

\[
\frac{F_a}{C_o} = \frac{1700}{22400} = 0.076
\]

Taking the next highest value from Figure 10.21, the value of \( e = 0.31 \)

\[
\frac{F_a}{F_r} = \frac{1700}{2750} = 0.61
\]

So \( \frac{F_a}{F_r} > e \)

Again, from Figure 10.21

\( x = 0.56 \) \( y = 1.4 \)
EXAMPLE 4

Select a DGBB for a 55 mm shaft to carry a radial force of 4 kN and an axial force of 2.2 kN at 1000 r/min. Duty hours required are 10 000.

Try bearing 6211 - refer to Figure 10.22

\[ \frac{F_a}{C_o} = \frac{2.2 \times 10^3}{25 000} = 0.088 \]

\[ \frac{F_a}{F_r} = \frac{2.2}{4} = 0.55 \]

So, use \( e = 0.31 \)

Thus \( \frac{F_a}{F_r} > e \)

which means that \( x = 0.56 \) and \( y = 1.4 \)

\[ P = (0.56 \times 4 \times 10^3) + (1.4 \times 2.2 \times 10^3) = 5.32 \text{ kN} \]

This life is too low, so try a 6311 bearing, which has higher ratings.

Again, using values from Figure 10.22:

\[ \frac{F_a}{C_o} = \frac{2.2 \times 10^3}{41 500} = 0.05 \]

\[ \frac{F_a}{F_r} = \frac{2.2}{4} = 0.55 \]

So, use \( e = 0.27 \)

Thus \( \frac{F_a}{F_r} > e \)

which means that \( x = 0.56 \) and \( y = 1.6 \)
Bearings

\[ P = (0.56 \times 4 \times 10^3) + (1.6 \times 2.2 \times 10^3) = 5.76 \text{ kN} \]

\[
L_H = \frac{10^6 \left( \frac{55000}{5760} \right)^3}{60 \times 1000}
= 14510 \text{ hours}
\]

So, bearing 6311 is suitable.
### Deep Groove Ball Bearings

#### 3-15 mm

<table>
<thead>
<tr>
<th>Boundary dimensions</th>
<th>Basic load ratings dynamic static</th>
<th>Limiting speeds Lubrication</th>
<th>Mass kg</th>
<th>Designation</th>
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#### Equivalent bearing load dynamic

\[ P = X_{U} + Y_{U} \]

\[ F_{D} = 0.5F_{U} + 0.5F_{L} \]

When \( F_{D} < F_{U} \), use \( F_{D} = F_{U} \)

### Calculation factors

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### Dimensions

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<th>( r )</th>
<th>( d_{2} )</th>
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### Abnormal and fillet dimensions

#### Fig. 10.20
## Deep groove ball bearings

**Dimensions**

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<th>d₁</th>
<th>d₂</th>
<th>f</th>
<th>d₃</th>
<th>d₄</th>
<th>t</th>
<th>tₐ</th>
<th>tₐ max</th>
<th>tₐ max</th>
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</table>

## Calculation factors

- **Equivalent bearing load**
  \[ P = 60P_e + 0.5P_t \]
- **Static dynamic**
  \[ P_{01} = 0.9F_c + 0.5F_t \]
- When \( P_c < P_t \), use \( P_c = P_t \).

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<th>( P_{01} )</th>
<th>( F_c )</th>
<th>( F_t )</th>
<th>( e )</th>
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<td>1</td>
<td>0.56</td>
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**Boundary dimensions**

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<th>N (N = 6.25D³)</th>
<th>( r/N )</th>
<th>( h/N )</th>
<th>( H/N )</th>
<th>( e/N )</th>
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<td>0.5</td>
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</table>

**Basic load ratings**

- **Static**
  \[ P_c = 60P_e + 0.5P_t \]
- **Dynamic**
  \[ P_t = 0.9F_c + 0.5F_t \]

### Notes

- **Fig. 10.21**
- **SKF**
- **Equivalent bearing load**
  \[ P = 60P_e + 0.5P_t \]
- **Static dynamic**
  \[ P_{01} = 0.9F_c + 0.5F_t \]
- When \( P_c < P_t \), use \( P_c = P_t \).

**Equivalent bearing load**

\[ P = 60P_e + 0.5P_t \]

**Static dynamic**

\[ P_{01} = 0.9F_c + 0.5F_t \]

**When \( P_c < P_t \), use \( P_c = P_t \).**

---

**Boundary dimensions**

<table>
<thead>
<tr>
<th>d</th>
<th>N (N = 6.25D³)</th>
<th>( r/N )</th>
<th>( h/N )</th>
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**Basic load ratings**

- **Static**
  \[ P_c = 60P_e + 0.5P_t \]
- **Dynamic**
  \[ P_t = 0.9F_c + 0.5F_t \]

### Notes

- **Fig. 10.21**
- **SKF**
- **Equivalent bearing load**
  \[ P = 60P_e + 0.5P_t \]
- **Static dynamic**
  \[ P_{01} = 0.9F_c + 0.5F_t \]
- When \( P_c < P_t \), use \( P_c = P_t \).
Deep groove ball bearings

### Boundary dimensions

<table>
<thead>
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<th>D</th>
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<th>d4</th>
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<td>5000</td>
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### Lubrication

- *d* = Calculated diameter
- *D* = Outer diameter
- *D1* = Inner diameter
- *r* = Ball radius
- *d2* = Bore diameter
- *d3* = Outer raceway diameter
- *d4* = Inner raceway diameter
- *Grease oil* = Oil quantity

### Equivalent bearing load

- Dynamic: *P = KF + Fp* = KF + Fp = 0.6 F + 0.5 F
- Static: *P = 0.5 F + 0.5 F*

### Calculation factors

- Dynamic: *P = FwFf + 0.5 F*
- Static: *P = 0.5 F + 0.5 F*

### Dimensional Abolishment and lift dimensions

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<tr>
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### Diagram

- Fig. 10.22

---

**Chapter Ten**
BELT AND CHAIN DRIVES

This Chapter deals with the methods of coupling two shafts together that are not axially in line but are parallel, and are coupled by a belt drive or a chain drive.

11.1 BELT DRIVES

In the past, the flat belt and pulley system was commonly used to transmit power from a prime mover, such as an electric motor or an internal combustion engine.

Suitable pulleys (one fixed and one idler) were placed along the drive or “line” shaft to convey power to small countershafts. By means of a forked device, the belts were moved sideways from one pulley to the other. When the belt was on the flat pulley, the shaft revolved. When the belt was on the adjacent idler pulley the drive stopped.

This simple method was used to stop and start various machines without stopping the common drive shaft. With the wider use of individual electric motors, machine drives became self-contained and the use of the common line shaft has virtually disappeared. However, applications still occur where the flat belt is used. It still remains as a means of transmitting power in agricultural machinery. The familiar power take-off on most tractors is the starting point for many flat belt drives driving external equipment such as welding plants, pumps and saw benches. Flat belts are used on high-speed grinding machine applications and likewise on some wood-working machines where high shaft speed is a requirement.

Flat belts:

• may be used on very long centre distances
• are relatively inexpensive
• may be used to transmit power with variations in shaft location, such as right-angled (quarter turn) and even crossed belt to provide reverse rotation
• may be used in situations where there are dust and abrasive particles in the air which would rapidly wear other drive systems
• may be used where the drive centre distances are changing slightly during running.

However, flat belts:

• require careful belt tensioning to keep slip down to a reasonable amount (less than 20%) and to ensure that the belt stays on the pulley
• tend to be noisy
• cannot transmit as much power per unit width as a V-belt. Thus flat belt drives tend to be space consuming.

FLAT BELT MATERIALS

The most popular belting materials include leather, impregnated leather, rubber, canvas and more recently, synthetic materials with nylon reinforcing. Leather was traditionally popular because of its durability and ability to retain end fasteners.

Rubber belting exhibits a lower coefficient of friction than leather so it requires greater tension.

Solid woven cotton belting is available in multiple plies and used for low-power transmission as well as conveyor application. Combinations of cotton, nylon, rayon and polyester are used in belts, where the one material is used in stranded form as a ply for strength and another between the plies to provide flexibility.
BELT FASTENERS

Wherever possible, endless belts are used, produced by moulding a complete belt or forming a scarfed joint - a long angled joint.

When belt fasteners have to be used they are metallic hinges, plates or lacing. Generally, lacing and hinges are used on light to medium loads, and plates on slow speed, heavy load applications such as conveyor belts.

Flat belt pulleys generally have a slight "crown" - a raised position in the centre of the pulley face which assists in keeping the flat belt on the pulley. The crown height is based on 0.015 times the pulley width.

V-BELTS

A comparatively recent development, V-belts are made in a variety of types, varying from the cheaper natural rubber versions, reinforced with rayon fibres, to the super service belts in which the reinforcing cords are made of terylene, nylon and steel.

The cheaper kind of belt is to be found in the fan drive of motor vehicles while the more expensive versions, having power ratings as much as 40% higher than standard belts, cost proportionally more.

Advantages of a V-belt (compared to a flat belt):
- The increased grip, owing to the wedging action of the belt, makes short centre drives possible.
- V-belt drives are quiet at high speeds.
- Belt slippage does not cause the belts to come out of the pulleys; this reduces belt wear.
- Belts and pulleys have been standardised, thus making replacements easy.
- Belt tension is low so load on shafts and bearings is low.
- V-belt drives require no lubrication and are suitable for belt speeds in the range 500 to 1500 m/min. They can be used for small domestic appliances such as sewing machines as well as for machinery using thousands of kilowatts.
- V-belts can transmit pulsating loads and also minimise shock when a machine starts.

Disadvantages:
- They are mainly applicable to short centre drives. Although versions that can be jointed are available, the centre distances are not, as a rule, as great as for flat belts.
- The enemies of the V-belt are heat, oil and dust. Dust also damages the pulleys over which they pass.
- The V-belt is best suited to drives where maximum speed ratio for single reduction drives is 6:1. In fractional power drives, however, this can be as much as 8:1.

Fig. 11.1 How the Vee Belt Operates
The principle of a V-belt drive is shown in Figure 11.1. As the belt bends it also bulges slightly - the bulge causes the belt sides to grip into the vee of the pulley. Furthermore, the angled side has the effect of increasing the coefficient of friction approximately three times for a given belt tension.

Figure 11.2 illustrates a general view of differing types of vee belts.

**TOOTHED BELTS** *(Figure 11.3)*

The toothed belt (or timing belt) was originally developed for an improved transmission on a high speed machine drive, where shock loads were to be encountered and there was need for synchronization of the shafts. It comprises a neoprene body reinforced with a number of steel or glass-fibre cords and having a woven nylon protection for the teeth.
Toothed belts are available in many sizes, for duties that range from the lightest of drives on sewing machines to the transmission of many kilowatts.

Advantages (compared to other drive systems):
- The speed range is wide, from practically zero to over 5000 m/min.
- It provides positive drive without backlash.
- It does not stretch or elongate in service unless overloaded.
- Unlike a chain drive, it provides a close approximation to constant velocity.
- It permits the use of small pulleys and thus makes compact drives possible.
- Since initial tension in the belt is not necessary, bearing sizes can be reduced.
- It has a long service life.
- It has a very low noise level.
- In addition to its function as a means of transmitting power, it can also be used as a functional part of a machine, for example, as a conveyor. Open ended versions can be used as a rack, as is sometimes done with chain.

Disadvantages:
- The toothed wheels must be made with great accuracy.
- Like V-belts, although made of oil-resistant neoprene, toothed belts are not yet proof against hot oil.
- The toothed belt is still a fairly expensive product.

11.2 BELT CALCULATIONS

The power transmitted by a belt (other than a toothed belt) is related to the force with which the belt is wrapped around the drive pulley. This force is itself dependent on the tensions in the belt - to drive, one side of the belt has a higher tension than the other side of the belt.

The other factors, as shown in Figure 11.4, are the angle of belt wrap and the coefficient of friction for the belt/pulley.

![Fig. 11.4 Belt Angle of Wrap](image)

It can be shown that:

$$\frac{T_1}{T_2} = e^{\mu \theta}$$
Where

\[ e = 2.718 \text{ (base of natural logs)} \]
\[ \mu = \text{coefficient of friction between belt and pulley} \]
\[ \theta = \text{arc of belt wrap in radians} \]
\[ T_1 = \text{tension in tight side of belt (N)} \]
\[ T_2 = \text{tension in slack side of belt (N)} \]

The arc of belt wrap is measured on the smaller of two pulleys in a drive system because that is where slip will first occur.

The arc of belt wrap can be improved by:

- maintaining the maximum distance between pulleys
- keeping pulley diameters as equal as possible
- installing a jockey pulley to "push" the belt together
- crossing the belt.

Ideally, the tension ratio is \( e^{\mu \theta} \) but because of the difficulty of maintaining belt tensions, the tension ratio is generally between \( e^{\mu \theta} \) and \( 0.5 e^{\mu \theta} \) when transmitting full power.

The power that can be transmitted by a V-belt depends on several factors:

(a) The belt cross-section. As shown in Figure 11.5 standardised belts fall into four main categories, designated A, B, C and D. Larger and smaller belt sizes exist than those shown but are not in such common use.

(b) Belt speed. As a result of centrifugal force tending to lift the belt out of the groove at high speed, the efficiency of a V-belt drive peaks as the belt speed reaches about 25 m/s. Above 25 m/s any increase in belt speed is accompanied by a decrease in transmitted power.

(c) Pulley diameter. The smaller the pulley, the more severe is the bending stresses because of belt wrap. Thus, the permissible belt tension is dependent on pulley sizes.

(d) Belt length. The severity of service (the service factor) of a V-belt drive depends on belt length because the shorter the belt, the more often it flexes around a pulley. The length factor is important on very short belt drives.
(e) Number of belts. To reduce the loading per belt, several belts may be used on a multiple groove pulley. However, the wider pulley increases the loading distance from the bearing. Also, the belts have to be matched in terms of their length and characteristics, otherwise all belts do not share the same loading. For the same reason, if one belt is damaged, the whole set must be replaced.

(f) Arc of contact. As the angle of belt wrap decreases, there is a greater tendency for the belt to slip. For example, if the arc of contact for a given belt reduces from 180° to 90° the transmittable power is reduced to three-quarters of the original value. Thus, large speed reductions are not desirable, particularly for short centre distance drives.

(g) Service conditions. If the belt system is subjected to shock loadings, its power rating is decreased. Table 1 from the Dunlop catalogue following shows service factors for various drive situations.

11.3 VEE BELT SELECTION FROM A CATALOGUE

Normally, a vee belt catalogue provides enough information to enable a correct vee belt system to be chosen for a given installation.

As an example, the following 12 pages are taken from a vee belt catalogue. The first page provides a worked example of vee belt selection.
# Designing V-Belt Drives

<table>
<thead>
<tr>
<th>Step</th>
<th>Procedure</th>
<th>Example</th>
</tr>
</thead>
</table>
| 1    | List the known quantities  
   a) Power to be transmitted  
   b) Speed of input and output shafts  
   c) Centre distance of shafts  
   d) Type of driving and driven machines  
   e) Running time per day  
   f) Driven Pulley—limit outside diam. | 11 kW squirrel cage motor  
1,440 r.p.m. and 720 r.p.m.  
750 mm  
Squirrel cage motor/fan  
18 hours per day  
355 mm |
| 2    | Select service factor and calculate the “belt design power” (Table 1) | 1.3 (service factor)  
11 x 1.3 = 14.3 Design power |
| 3    | Select the vee belt section to be used. (Table 2) | “B” section |
| 4    | Calculate the speed ratio and select pulley diameters. (Table 3) | 1,440  
720 = 2.1 speed ratio |
| 5    | Calculate the belt length mm  
\[ L = \frac{2c + 1.57(Dp - dp)}{(Dp - dp)^2} \] | Calculated length  
2214 mm |
| 6    | Choose the nearest standard belt length. (Table 5) | Select next smaller standard length  
= 2180 mm pitch (890 belt) |
| 7    | Calculate the correct centre distance mm  
\[ C = A + \sqrt{(A^2 - B)} \]  
where \( A = \frac{1}{2} - \pi \frac{(Dp + dp)}{8} \)  
and \( B = \frac{(Dp - dp)^2}{8} \) | 733 mm centres |
| 8    | Determine the power per belt. (Table 10) | \( 5.11 \times 0.54 = 5.65 \) kW |
| 9    | Determine the ‘arc of contact’ factor. (Table 6) | 0.97 |
| 10   | Determine the ‘belt length’ factor. (Table 7) | 0.99 |
| 11   | Calculate the number of belts required where the number of belts  
\[ = \frac{14.3}{5.17 \times 0.97 \times 0.99} = 2.63 \]  
Use 7 belts. |

**Metric (SI) — Imperial conversion factors**  
Horsepower x 0.746 = kilowatts  
Inches x 25.4 = millimeters

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Table 1

<table>
<thead>
<tr>
<th>TYPES OF DRIVEN MACHINES</th>
<th>TYPES OF DRIVING UNITS</th>
<th>OPERATIONAL HOURS PER DAY</th>
</tr>
</thead>
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<tr>
<td></td>
<td></td>
<td>10 and under</td>
</tr>
<tr>
<td>Special</td>
<td>A.C. Motors</td>
<td>over 600 rev. min.</td>
</tr>
<tr>
<td></td>
<td>D.C. Motors</td>
<td>over 600 rev. min.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Light Duty</td>
<td>Centrifugal pumps and compressors</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>Light duty conveyors</td>
<td>1.0</td>
</tr>
<tr>
<td>Medium Duty</td>
<td>Belt conveyors for sand, grain, etc</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Dough mixers</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Fans over 7.5 kw</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Generators</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Line shafts</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Laundry machinery</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Machine tools</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Pumps, press, shears</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Printing machinery</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Positive displacement pumps</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Resin and vibrators</td>
<td>1.1</td>
</tr>
<tr>
<td>Heavy Duty</td>
<td>Bricklaying equipment</td>
<td>1.1</td>
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<tr>
<td></td>
<td>Conveyors (drag, screw)</td>
<td>1.1</td>
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<tr>
<td></td>
<td>Hammer mixers</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Paper mill beaters</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Paper pulp mill</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Positive displacement pumps</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Pumps, press, shears</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Printing machinery</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Positive displacement pumps</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>Sawmill and woodworking machinery</td>
<td>1.1</td>
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<tr>
<td></td>
<td>Textile machinery</td>
<td>1.1</td>
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<td>Crushers (granity, raw, oil)</td>
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<td>Miscellaneous (Load)</td>
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<td></td>
<td>Miscellaneous (醫療)</td>
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<td></td>
<td>Miscellaneous (medical)</td>
<td>1.3</td>
</tr>
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</table>

Note: Statistical authorities should be consulted for the appropriate factor for such equipment as cranes, hoists, lifts, elevators and conveyors not specified above.

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# Chapter Eleven

## Table 2

<table>
<thead>
<tr>
<th>Speed of faster shaft (rev/min)</th>
<th>Design power in kW</th>
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<td>0.751</td>
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</tr>
<tr>
<td>2</td>
<td>3</td>
</tr>
<tr>
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<td>4</td>
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<td>400</td>
<td>500</td>
</tr>
<tr>
<td>500</td>
<td>750</td>
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</tbody>
</table>

## Table 3

### RECOMMENDED STANDARD PULLEY OUTSIDE AND PITCH DIAMETERS (Old Imperial sizes in brackets)

<table>
<thead>
<tr>
<th>&quot;A&quot; Section (inch)</th>
<th>Pitch diameter</th>
<th>&quot;B&quot; Section (inch)</th>
<th>Pitch diameter</th>
<th>&quot;C&quot; Section (inch)</th>
<th>Pitch diameter</th>
<th>&quot;D&quot; Section (inch)</th>
<th>Pitch diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter</td>
<td></td>
<td>Outside diameter</td>
<td></td>
<td>Outside diameter</td>
<td></td>
<td>Outside diameter</td>
<td></td>
</tr>
<tr>
<td>mm</td>
<td></td>
<td>mm</td>
<td></td>
<td>mm</td>
<td></td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>54.2</td>
<td>50</td>
<td>2.0</td>
<td>85.0</td>
<td>10.2</td>
<td>133.4</td>
<td>125</td>
<td>211.4</td>
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<td>57.2</td>
<td>53</td>
<td>2.1</td>
<td>88.6</td>
<td>10.2</td>
<td>140.4</td>
<td>132</td>
<td>223.4</td>
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<tr>
<td>60.2</td>
<td>56</td>
<td>2.2</td>
<td>91.5</td>
<td>10.2</td>
<td>146.4</td>
<td>139</td>
<td>235.4</td>
</tr>
<tr>
<td>64.2</td>
<td>60</td>
<td>2.4</td>
<td>95.5</td>
<td>10.6</td>
<td>159.4</td>
<td>150</td>
<td>247.4</td>
</tr>
<tr>
<td>67.2</td>
<td>63</td>
<td>2.5</td>
<td>99.9</td>
<td>10.8</td>
<td>168.4</td>
<td>160</td>
<td>254.4</td>
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<td>71.2</td>
<td>67</td>
<td>2.6</td>
<td>106.6</td>
<td>11.0</td>
<td>178.4</td>
<td>170</td>
<td>276.4</td>
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<td>75.2</td>
<td>71</td>
<td>2.8</td>
<td>112.6</td>
<td>11.2</td>
<td>188.4</td>
<td>180</td>
<td>291.4</td>
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<td>79.2</td>
<td>75</td>
<td>3.0</td>
<td>118.6</td>
<td>11.4</td>
<td>196.4</td>
<td>191</td>
<td>311.4</td>
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<tr>
<td>84.2</td>
<td>80</td>
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<td>124.5</td>
<td>11.6</td>
<td>209.4</td>
<td>202</td>
<td>326.4</td>
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<td>94.2</td>
<td>90</td>
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<td>131.6</td>
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<td>396.4</td>
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<td>12.5</td>
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<td>146.6</td>
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<td>289</td>
<td>411.4</td>
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<td>125</td>
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<td>155.6</td>
<td>14.0</td>
<td>326.4</td>
<td>320</td>
<td>451.4</td>
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<td>140</td>
<td>5.5</td>
<td>166.6</td>
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<td>514.4</td>
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<td>160</td>
<td>6.0</td>
<td>176.6</td>
<td>16.0</td>
<td>393.4</td>
<td>393</td>
<td>541.4</td>
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<tr>
<td>184.2</td>
<td>180</td>
<td>6.5</td>
<td>186.6</td>
<td>17.0</td>
<td>416.4</td>
<td>416</td>
<td>571.4</td>
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<td>204.2</td>
<td>200</td>
<td>7.0</td>
<td>196.6</td>
<td>18.0</td>
<td>456.4</td>
<td>456</td>
<td>611.4</td>
</tr>
<tr>
<td>228.2</td>
<td>224</td>
<td>7.5</td>
<td>206.6</td>
<td>19.0</td>
<td>496.4</td>
<td>496</td>
<td>641.4</td>
</tr>
<tr>
<td>254.2</td>
<td>250</td>
<td>8.0</td>
<td>216.6</td>
<td>20.0</td>
<td>536.4</td>
<td>536</td>
<td>671.4</td>
</tr>
</tbody>
</table>

**Note:** Aust. Standard Tolerance on the metric pitch diameter = 0 ± 1.6%, Imperial Standard Tolerance = 0 ± 1.6% Approximate Imperial sizes in brackets.
### Table 4
DIMENSIONS OF STANDARD V-GROOVED PULLEYS
(See Table 3 for recommended standard pulley outside and pitch diameters).

The maximum distance between the outside edges of the pulley, i.e., the face width, is equal to \( w = n \times 2 \) where \( n \) is the number of grooves.

#### Multi-groove pulley cross section

<table>
<thead>
<tr>
<th>Groove cross section symbol</th>
<th>Pitch diameter of the pulley (dp)</th>
<th>Groove angle (a)</th>
<th>Minimum top width of groove (g)</th>
<th>Minimum groove depth below outside diameter (d)</th>
<th>Centre-to-centre of grooves (see Note 2) (e)</th>
<th>Edge of pulley to first groove (f)</th>
<th>Minimum distance from outside diameter to pitch diameter (b)</th>
<th>Groove pitch width (p)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A 75 (Recommended minimum)</td>
<td>34 ( \pm 0.5 )</td>
<td>13.0</td>
<td>12</td>
<td>15 ( \pm 0.3 )</td>
<td>10 - 2</td>
<td>3.3</td>
<td>11</td>
<td></td>
</tr>
<tr>
<td>and under 125</td>
<td>36 ( \pm 0.5 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>B 125 (Recommended minimum)</td>
<td>34 ( \pm 0.5 )</td>
<td>16.6</td>
<td>15</td>
<td>19 ( \pm 0.4 )</td>
<td>12.5 ( \pm 2 )</td>
<td>4.2</td>
<td>14</td>
<td></td>
</tr>
<tr>
<td>and under 200</td>
<td>36 ( \pm 0.5 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C 200 (Recommended minimum)</td>
<td>36 ( \pm 0.5 )</td>
<td>22.7</td>
<td>20</td>
<td>25.5 ( \pm 0.5 )</td>
<td>17 - 2</td>
<td>5.7</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>and under 300</td>
<td>36 ( \pm 0.5 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>D 355 (Recommended minimum)</td>
<td>36 ( \pm 0.5 )</td>
<td>32.3</td>
<td>28</td>
<td>37 ( \pm 0.6 )</td>
<td>24 - 3</td>
<td>8.1</td>
<td>27</td>
<td></td>
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<tr>
<td>and under 500</td>
<td>38 ( \pm 0.5 )</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>500 and over</td>
<td>38 ( \pm 0.5 )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Note 1:** The tolerances on dimension "w" apply to the distance between the centres of any two grooves whether consecutive or not.

**Note 2:** It is recommended that the tolerance on dimension "t" should be taken into account in the alignment of pulleys.

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# Table 5

**CROSS SECTION DIMENSIONS & STANDARD LENGTHS**

<table>
<thead>
<tr>
<th></th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
</tr>
</thead>
<tbody>
<tr>
<td>Z</td>
<td><img src="image1" alt="Illustration A" /></td>
<td><img src="image2" alt="Illustration B" /></td>
<td><img src="image3" alt="Illustration C" /></td>
<td><img src="image4" alt="Illustration D" /></td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Nominal pitch length</th>
<th>Inside length inches</th>
<th>Nominal pitch length</th>
<th>Equivalent designation in Table 3 BS1440</th>
<th>1968 (inside length)</th>
<th>Nominal pitch length</th>
<th>Equivalent designation in Table 3 BS1440</th>
<th>1968 (inside length)</th>
<th>Nominal pitch length</th>
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<td>in</td>
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<td>in</td>
<td>mm</td>
<td>mm</td>
<td>in</td>
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<td>mm</td>
<td>in</td>
<td>mm</td>
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<td>530</td>
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<td>590</td>
<td>35</td>
<td>1160</td>
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<td>105</td>
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**Designation**

For ordering purposes belts are designated by the inside length in inches (Sept. 75). E.g. 897 belt is 17 mm wide and has an inside length of 97 inches and a pitch length of 2200 mm.

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### Table 6
**CONTACT SHOWING PROPORTION OF POWER RATINGS AT 180° ARC OF CONTACT FOR V- AND V-FLAT DRIVES**

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<thead>
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<th>D-d</th>
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<th>Correction factor i.e., proportion of 180° rating</th>
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<th>V-Flat</th>
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### Table 7
**POWER CORRECTION FACTORS FOR BELT LENGTH**

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**Notes:**
1. Arcs of contact below 120° should not be used unless full drive details are first submitted to the V-drive manufacturer concerned for confirmation.
2. It should be noted that the correction factors for V-flat drives diminish progressively for arcs of contact greater than 133°, and the use of such drives is usually not found to be economical for arcs of contact greater than 151°.

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RECOMMENDED PRACTICE FOR INSTALLATION TENSION IN V-BELT DRIVES AND CALCULATION OF RESULTANT FORCE IMPOSED ON THE SHAFT BY V-BELTS

V-belts work satisfactorily over a wide range of belt tensions so that rough and ready methods of deciding on belt tensions have not, as yet, resulted in serious troubles with the drive. However, it is desirable to be able to measure tensions with sufficient accuracy to avoid bearing trouble or belt slip or to meet particularly arduous conditions. The following procedure is recommended for drives coming within the normal range for each belt section as defined in this specification.

Measure the length of the span in millimetres. At the centre of the span apply a force with a spring scale in a direction perpendicular to the span, until the belt is deflected from the normal by an amount equal to 0.016 mm for every millimetre of span length (see Fig. 1).

For example, the deflection for a span of 1 m would be $1000 \times 0.016$ or 16 mm. Note the force and compare it with the value of $P$ given in Table 8.

In all cases it is essential that the pulley centres be fixed and that the larger pulley is then rotated at least four times before making the measurement. On a multiple V-belt drive it is essential that a matched set of belts be used and the above procedure carried out on each belt. The average values of these forces being compared with the specified values of $P$ in Table 8.

The belt tension should be satisfactory if its value is between that for normal and 1.5 times normal tension. However, when starting up a drive, with new belts a tension of 2.0 times normal is acceptable, since the tension falls rapidly in the early stages of running-in. Some difficult drives may need a tension as high as 2.0 times normal tension to be maintained, therefore retensioning is necessary after an initial running-in period. Difficult drives are usually those with one or more of the following properties:

- high belt speed.
- low belt speed.
- small arc of contact.
- high overload on start-up.

Fig. 1. Belt deflection measurement

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### Table 8

**DEFLECTION FORCE REQUIRED FOR MEASURING INSTALLATION TENSION IN V-BELT DRIVES**

<table>
<thead>
<tr>
<th>Belt cross section</th>
<th>Required force $P$ at centre of span for normal tension</th>
<th>Required force $P$ at centre of span for 1.5 times normal tension</th>
<th>Required force $P$ at centre of span for 2.2 times normal tension</th>
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The total static hub load $W$, imposed by the belts on the shaft is the vector sum of the tensions in the belts and it can be calculated with sufficient accuracy by the following formula:

$$W = 32nP \sin \frac{n}{2} \text{ (newtons).}$$

where $n = \text{number of belts,}$

$n = \text{arc of contact on smaller pulley,}$

$P = \text{force at centre of span (newtons).}$

To determine the dynamic hub load $W$, a correction needs to be made to the static tension to account for the effect of centrifugal force before the vectorial summation, i.e.

$$W = 32n(P - K) \sin \frac{n}{2} \text{ (newtons).}$$

where $K = \text{correction factor for centrifugal tension (see Table 9).}$

### Table 9

**VALUES OF K TO CORRECT FOR THE EFFECT OF CENTRIFUGAL TENSION**

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Reproduced by courtesy of Dunlop Industrial, Australia
Table 10A

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<td>Solder pitch diameter (mm)</td>
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Table 10A

| R.P.M. of |
|----------|--------------------------------------------------|
| 65       | Additional power per belt for speed ratio       |
| 75       | R.P.M. of                                    |
| 85       | 1.02                          | 1.15                          | 1.23                          | 1.31                          | 1.40                          | 1.46                          | 1.52                          | 1.60                          | 1.73                          |
| 95       | 0.12                          | 0.14                          | 0.17                          | 0.20                          | 0.23                          | 0.27                          | 0.32                          | 0.39                          | 0.49                          |
| 105      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 112      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 125      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 135      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 1440      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 1600      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 1800      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 2000      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 2200      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 2400      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 2600      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 2800      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
| 3000      | 0.02                          | 0.02                          | 0.03                          | 0.04                          | 0.05                          | 0.06                          | 0.08                          | 0.10                          | 0.12                          |
### Table 10B

**KW POWER RATINGS FOR "B" SECTION DUNLOP V-BELTS**

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### Table 10C

**KW POWER RATINGS FOR "C" SECTION DUNLOP V-BELTS**

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<th>Speed of faster shaft</th>
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<td>kW</td>
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<td>kW</td>
<td>kW</td>
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<td>4.05</td>
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<th>Additional power per belt for speed ratio 1.19 to 1.24</th>
<th>Additional power per belt for speed ratio 1.25 to 1.30</th>
<th>Additional power per belt for speed ratio 1.31 to 1.36</th>
<th>Additional power per belt for speed ratio 1.37 to 1.42</th>
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**Table 10C Reproduced by courtesy of Dunlop Industrial, Australia**
## Table 10D

**KW POWER RATINGS FOR “D” SECTION DUNLOP V-BELTS**

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<tr>
<th>Speed of faster shaft (rev/min)</th>
<th>Small pulley pitch diameter (mm)</th>
<th>Speed of faster shaft (rev/min)</th>
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## Table 10D

**KW POWER RATINGS FOR “D” SECTION DUNLOP V-BELTS**

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<th>1.13 to 1.19</th>
<th>1.19 to 1.25</th>
<th>1.25 to 1.35</th>
<th>1.35 to 1.51</th>
<th>1.51 to 1.99 &amp; over (kW)</th>
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<td></td>
<td>Speed of faster shaft (rev/min)</td>
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<td>kW</td>
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<td>0.43</td>
<td>0.57</td>
<td>0.71</td>
<td>0.85</td>
<td>0.99</td>
<td>1.13</td>
</tr>
<tr>
<td>600</td>
<td>0.17</td>
<td>0.34</td>
<td>0.51</td>
<td>0.68</td>
<td>0.85</td>
<td>1.02</td>
<td>1.19</td>
<td>1.36</td>
</tr>
<tr>
<td>700</td>
<td>0.20</td>
<td>0.40</td>
<td>0.60</td>
<td>0.79</td>
<td>0.99</td>
<td>1.19</td>
<td>1.39</td>
<td>1.58</td>
</tr>
<tr>
<td>800</td>
<td>0.22</td>
<td>0.46</td>
<td>0.68</td>
<td>0.90</td>
<td>1.13</td>
<td>1.36</td>
<td>1.58</td>
<td>1.81</td>
</tr>
<tr>
<td>900</td>
<td>0.25</td>
<td>0.51</td>
<td>0.77</td>
<td>1.01</td>
<td>1.28</td>
<td>1.53</td>
<td>1.78</td>
<td>2.04</td>
</tr>
<tr>
<td>1000</td>
<td>0.28</td>
<td>0.57</td>
<td>0.85</td>
<td>1.13</td>
<td>1.42</td>
<td>1.70</td>
<td>1.98</td>
<td>2.27</td>
</tr>
<tr>
<td>1100</td>
<td>0.31</td>
<td>0.63</td>
<td>0.93</td>
<td>1.25</td>
<td>1.56</td>
<td>1.87</td>
<td>2.18</td>
<td>2.49</td>
</tr>
<tr>
<td>1200</td>
<td>0.34</td>
<td>0.68</td>
<td>1.02</td>
<td>1.36</td>
<td>1.70</td>
<td>2.04</td>
<td>2.37</td>
<td>2.72</td>
</tr>
<tr>
<td>1300</td>
<td>0.37</td>
<td>0.74</td>
<td>1.10</td>
<td>1.47</td>
<td>1.84</td>
<td>2.21</td>
<td>2.57</td>
<td>2.95</td>
</tr>
<tr>
<td>1400</td>
<td>0.40</td>
<td>0.79</td>
<td>1.19</td>
<td>1.58</td>
<td>1.98</td>
<td>2.38</td>
<td>2.77</td>
<td>3.17</td>
</tr>
<tr>
<td>1500</td>
<td>0.43</td>
<td>0.85</td>
<td>1.26</td>
<td>1.70</td>
<td>2.13</td>
<td>2.55</td>
<td>2.97</td>
<td>3.39</td>
</tr>
<tr>
<td>1600</td>
<td>0.46</td>
<td>0.91</td>
<td>1.36</td>
<td>1.81</td>
<td>2.27</td>
<td>2.72</td>
<td>3.16</td>
<td>3.63</td>
</tr>
</tbody>
</table>

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As another example of vee belt selection, the following pages show the system given in a Fenner catalogue.

NOTE: Inputting the design power and fastest shaft speed provides not only suitable belt sizes, but also suggests a small pulley pitch diameter.

The first chart (Figure 11.6) requires a certain amount of extrapolation. For instance, if the design power is 14.3 kW and the maximum speed is 1440 r/min, then two choices exist:

1. a small pulley of approximately 135 mm pitch diameter and six type-A belts
2. a small pulley of approximately 160 mm pitch diameter and five type-B belts.

To verify the number of belts needed, a "Power per belt" table should be consulted, such as Figure 11.7, which is for B-section Fenner vee belts.

Figure 11.8 is a catalogue page that aids selection of a toothed or timing belt. The three categories of belt refer to:

<table>
<thead>
<tr>
<th>TYPE</th>
<th>PITCH (mm)</th>
<th>WIDTH (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light</td>
<td>9.53</td>
<td>13 - 25</td>
</tr>
<tr>
<td>Heavy</td>
<td>12.7</td>
<td>19 - 76</td>
</tr>
<tr>
<td>Extra Heavy</td>
<td>22.23</td>
<td>51 - 102</td>
</tr>
</tbody>
</table>

Table 11.1 Toothed Belt Dimensions

Note that the "Design Power" is the transmitted power multiplied by a design factor which allows for loading conditions.

Once the belt pitch is chosen for a given application, a suitable pulley is selected. The belt width is determined by the power the pulley can transmit per unit width, compared to the total power to be transmitted.
Fig. 11.6

Reproduced with permission of Fenner Australia Pty Ltd
<table>
<thead>
<tr>
<th>R/MIN OF OFFSET SHAFT</th>
<th>RATED POWER (kW) PER BELT FOR SMALL PULLEY PITCH DIA. (mm)</th>
<th>BELT SPEED M/S</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>125</td>
<td>132</td>
</tr>
<tr>
<td>720</td>
<td>1.61</td>
<td>1.79</td>
</tr>
<tr>
<td>860</td>
<td>2.02</td>
<td>2.24</td>
</tr>
<tr>
<td>1440</td>
<td>2.72</td>
<td>3.03</td>
</tr>
<tr>
<td>2880</td>
<td>3.96</td>
<td>4.44</td>
</tr>
<tr>
<td>100</td>
<td>0.32</td>
<td>0.35</td>
</tr>
<tr>
<td>200</td>
<td>0.57</td>
<td>0.63</td>
</tr>
<tr>
<td>300</td>
<td>0.80</td>
<td>0.88</td>
</tr>
<tr>
<td>400</td>
<td>1.01</td>
<td>1.11</td>
</tr>
<tr>
<td>500</td>
<td>1.21</td>
<td>1.33</td>
</tr>
<tr>
<td>600</td>
<td>1.40</td>
<td>1.55</td>
</tr>
<tr>
<td>700</td>
<td>1.58</td>
<td>1.75</td>
</tr>
<tr>
<td>800</td>
<td>1.75</td>
<td>1.94</td>
</tr>
<tr>
<td>900</td>
<td>1.92</td>
<td>2.13</td>
</tr>
<tr>
<td>1000</td>
<td>2.08</td>
<td>2.31</td>
</tr>
<tr>
<td>1100</td>
<td>2.23</td>
<td>2.49</td>
</tr>
<tr>
<td>1200</td>
<td>2.38</td>
<td>2.66</td>
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<tr>
<td>1300</td>
<td>2.53</td>
<td>2.82</td>
</tr>
<tr>
<td>1400</td>
<td>2.66</td>
<td>2.97</td>
</tr>
<tr>
<td>1500</td>
<td>2.79</td>
<td>3.12</td>
</tr>
<tr>
<td>1600</td>
<td>2.92</td>
<td>3.26</td>
</tr>
<tr>
<td>1700</td>
<td>3.04</td>
<td>3.40</td>
</tr>
<tr>
<td>1800</td>
<td>3.15</td>
<td>3.52</td>
</tr>
<tr>
<td>1900</td>
<td>3.26</td>
<td>3.65</td>
</tr>
<tr>
<td>2000</td>
<td>3.36</td>
<td>3.76</td>
</tr>
<tr>
<td>2100</td>
<td>3.45</td>
<td>3.87</td>
</tr>
<tr>
<td>2200</td>
<td>3.54</td>
<td>4.00</td>
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<td>2300</td>
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<td>4.14</td>
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<tr>
<td>2500</td>
<td>3.77</td>
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<tr>
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<td>3.82</td>
<td>4.28</td>
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<tr>
<td>2700</td>
<td>3.88</td>
<td>4.35</td>
</tr>
<tr>
<td>2800</td>
<td>3.93</td>
<td>4.40</td>
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<tr>
<td>2900</td>
<td>3.97</td>
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<td>4.00</td>
<td>4.48</td>
</tr>
<tr>
<td>3100</td>
<td>4.02</td>
<td>4.50</td>
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<td>3200</td>
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<td>4.52</td>
</tr>
<tr>
<td>3300</td>
<td>4.06</td>
<td>4.52</td>
</tr>
<tr>
<td>3400</td>
<td>4.07</td>
<td>4.53</td>
</tr>
<tr>
<td>3500</td>
<td>4.04</td>
<td>4.50</td>
</tr>
<tr>
<td>3600</td>
<td>4.02</td>
<td>4.48</td>
</tr>
<tr>
<td>3700</td>
<td>3.99</td>
<td>4.45</td>
</tr>
<tr>
<td>3800</td>
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<td>4.40</td>
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<td>3900</td>
<td>3.91</td>
<td>4.34</td>
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<tr>
<td>4000</td>
<td>3.85</td>
<td>4.28</td>
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<tr>
<td>4100</td>
<td>3.78</td>
<td>4.20</td>
</tr>
<tr>
<td>4200</td>
<td>3.71</td>
<td>4.11</td>
</tr>
<tr>
<td>4300</td>
<td>3.62</td>
<td>4.00</td>
</tr>
<tr>
<td>4400</td>
<td>3.53</td>
<td></td>
</tr>
<tr>
<td>4500</td>
<td>3.42</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 11.7

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Fig. 11.8

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11.4 CHAIN DRIVES

GENERAL

Chains are available from the smallest types for instrument purposes, some of these being in plastics material, to multi-strand versions capable of transmitting thousands of kilowatts.

Advantages:
- No slippage, therefore high torques can be transmitted.
- Synchronous relationship between the driver and driven shafts. This means that there is no slip.
- No slack-side tension is required. Therefore there is no severe loading of the bearings; power being transmitted is independent of the initial tension in the chain and larger ratios of driver to driven sprockets can be achieved in a smaller space than with belt drives.
- Chain drives transmit the most power for width for a given diameter and speed, with least bearing overhang.
- Larger ratios can be driven with shorter centres than by belt, as the arc of contact is less important.
- Chain provides an easy means of driving several shafts from one power source, from either side of the chain.
- Chain drives can be used in conditions of heat which would render any belt drive useless. Special forms of malleable chain are used in furnace conveyors.

Disadvantages:
- At comparable speeds, chain drives are noisy unless a special form of silent chain is specified. The noise of chains is largely owing to pulsations in the drive caused by a polygon effect, i.e. slight accelerations and decelerations of the chain as it passes over the sprocket teeth. Silent forms of chain are claimed to go far to curing this problem. See Figs 11.13 and 11.11.
- For satisfactory performance, at medium to high speeds, chains should be enclosed and lubricated. For high speed applications this can be a costly business as the enclosures have to be oil-tight.
- Chains are somewhat inelastic and are therefore not always suitable for pulsating loads. Unless precautions are taken, the backlash can be a disadvantage.
- The effect of chain joint wear, which has the effect of elongating the chain, if carried too far, necessitates the replacement of the sprockets as well as the chain.

Beyond these very general comments it can be said of chains that, in addition to their special uses for transmitting power, they can also be used in a variety of ways for achieving mechanical movements.

PRECISION ROLLER CHAIN

The universally-accepted type of chain used for transmission of power is the precision roller chain. It is called precision roller chain because its components are made to fine tolerances but it is generally referred to as roller chain.

Roller chain (Figure 11.9) consists essentially of alternatively assembled inner and outer links. An inner link consists of two steel side plates held rigidly together with hardened steel bushes. A hardened steel roller mounted on each bush between the plates is free to rotate on the bush. The outer link consists of two steel side plates rigidly held together by two hardened steel pins.

On assembly, the pins are rivetted to one side plate and then passed through one bush on each side of two adjacent inner links. The other side plate is pressed over the ends of the pins and the pin ends are rivetted over.
Roller chains are classified by:

(a) Pitch - the centre distance of adjacent teeth. Based on inch sizes, standard chain pitches are 9.5, 12.7, 15.9, 19 and 25.4mm. Also available is 31.8, 38.1, 44.5, 50.8, 63.5 and 76.2 mm pitch chain.

(b) Width - Short pitch chains are manufactured in single, double and triple widths (known as single, duplex and triplex). (Figure11.10.) As would be expected, the load carried by a triplex chain is approximately three times the load carried by a single chain.
OTHER CHAIN TYPES (Figure 11.11)

*Cranked link chain* is a heavy duty roller chain made in large sizes. It is used for oil well and excavator drives and similar heavy rough machinery. Wheels for the chain normally have cast teeth.

*Malleable iron chain* is used under rough and exposed conditions.

*Inverted tooth chain* is a plate link chain with teeth on the inside of the chain. It usually has a central guide link in the chain to prevent it from running off the wheel while in motion. This type of chain is comparatively quiet and very efficient. It is often called silent chain. The most common application of this type of chain is as the timing chain on a vehicle.

*Conveyor chain* consists of a series of links with either extended side plates or flat top plates. Additional items can be fixed to the plates to provide a series of hangers or supports. Material may be conveyed on the chain or components may be suspended from the chain.

![Fig. 11.11 Chain Types](image)

**WHEELS FOR CHAIN DRIVES FROM A CATALOGUE**

Chain wheels are frequently called *sprockets* or sprocket wheels. Wheels for use with precision roller chain have machine-cut teeth and the standard teeth range is 17, 19, 21, 23, 25, 38, 57, 76, 95 and 114. However, wheels with any number of teeth can be produced for special applications. Wheels having up to 29 teeth are called *pinions* while those having 30 or more teeth are called simply wheels.

Note that the pitch of a wheel implies the chordal distance between teeth, identical to the chain pitch.

**11.5 CHAIN SELECTION FROM A CATALOGUE**

Table 11.2 displays the ultimate strength, in kilonewtons, of a typical roller chain.

<table>
<thead>
<tr>
<th>PITCH</th>
<th>9.5 mm</th>
<th>12.7 mm</th>
<th>15.9 mm</th>
<th>19.0 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simple</td>
<td>9.8</td>
<td>18.6</td>
<td>24.5</td>
<td>29.4</td>
</tr>
<tr>
<td>Duplex</td>
<td>18.6</td>
<td>35.3</td>
<td>44.9</td>
<td>58.9</td>
</tr>
<tr>
<td>Triplex</td>
<td>26.5</td>
<td>52.0</td>
<td>73.6</td>
<td>88.3</td>
</tr>
</tbody>
</table>

Table 11.2 Ultimate Tensile Strength of Roller Chain
For installations where chain life is not important, such as a drive that only operates occasionally, a suitable chain may be chosen by multiplying the maximum drive load by a suitable factor of safety (say 10) and using the value to select a chain.

For continuous use applications of roller chain, manufacturers' catalogues provide comprehensive information to aid the correct chain selection.

Figure 11.12 shows a chart which may be used to select the correct pitch chain for a given set of conditions.

Consider the situation where a rotary kiln is to be rotated at 80 r/min at a power consumption of 2.5 kW. The prime mover is a geared output electric motor running at 200 r/min. Because the type of drive is subjected to only moderate shock, the service factor is taken as 1.3. The service factor is found from the catalogue and represents a safety factor to suit the drive conditions.

Thus, the design power is 2.5 x 1.3 = 3.25 kW at a speed of 80 r/min.

Referring to Figure 11.12, the two design parameters indicate a chain of either 19 mm pitch or 25.4 mm pitch. The shorter the chain pitch, the smoother is the chain drive system, so a 19 mm pitch chain is chosen.

A shorter pitch provides a more constant drive velocity. If the chain is considered as a series of drive pins connected by straight links as in Figure 11.13, the chain velocity must vary continuously.
Considering the roller at point A, as it moves towards B its locus is radial but the distance AB must be constant. With shorter pitch chain (smaller AB) the chain tends to conform to the arc length AB instead of the chordal length AB.

![Fig. 11.13 Cause on Non-Constant Chain Velocity](image)

Because of the low power transmitted a single chain will suffice but for higher powers a duplex or triplex chain is chosen. Choose the correct sprockets to obtain the correct speed ratio but use standardised sprockets; again the catalogue provides a speed ratio table. (Table 11.3)

<table>
<thead>
<tr>
<th>RATIOS POSSIBLE WITH STOCK SPROCKETS</th>
</tr>
</thead>
<tbody>
<tr>
<td>NUMBER OF TEETH IN DRIVEN SPROCKET</td>
</tr>
<tr>
<td>13</td>
</tr>
<tr>
<td>13</td>
</tr>
<tr>
<td>15</td>
</tr>
<tr>
<td>17</td>
</tr>
<tr>
<td>19</td>
</tr>
<tr>
<td>21</td>
</tr>
<tr>
<td>23</td>
</tr>
<tr>
<td>25</td>
</tr>
<tr>
<td>27</td>
</tr>
</tbody>
</table>

Table 11.3 Sprocket Ratios

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The speed ratio required for the example given is:

\[ 200 \div 80 = 2.5:1 \]

From Table 11.3, the nearest ratio possible with standard (stock) sprockets is 2.48:1 which is a 23-tooth pinion driving a 57-tooth sprocket.

If a ratio closer to 2.5:1 is essential, special sprockets would have to be made with the correct number of teeth. This procedure should obviously be avoided if possible because not only is a custom-made sprocket more expensive, but should it need to be replaced at a later date, such a sprocket would not be available "ex-stock".

It should be noted that the highest ratio shown in Table 11.3 is 5.85:1.

Ratios in excess of 5:1 should be avoided because at higher ratios the number of teeth on the pinion in contact with the chain is inadequate to carry the load. At any sprocket ratio the power carrying capacity of the sprockets must be checked with the catalogue to ensure that their maximum rating is not exceeded.
Belt and Chain Drives

The chain length is calculated from:

\[ L = \frac{2C}{P} + \frac{T + t}{2} \]

Where:

- \( L \) = length (in chain pitches)
- \( C \) = shaft centre in distance (mm)
- \( T \) = number of teeth in large sprocket
- \( t \) = number of teeth in small sprocket
- \( P \) = chain pitch

Note that where a big difference exists between the number of teeth on the two sprockets, an additional factor is included which increases the chain length.

If the torque transmitted by a sprocket needs to be calculated, the catalogue also provides the pitch circle diameter of each standard sprocket for each pitch of chain.

As a further check on the suitability of the chain chosen in the previous example, the catalogue states that the pitch circle diameter of a 23 tooth pinion for 19 mm chain is 139.9 mm.

Power transmitted (including the service factor) was 3.25 kW at a speed of 200 r/min.

\[ \text{Power} = 2\pi NT \]

\[ \text{Torque} = \frac{\text{Power}}{2\pi N} \]

\[ \text{Torque} = \text{force} \times \text{radius} \]

So:

\[ \text{Chain force} = \frac{\text{Power}}{2\pi \times N \times \text{chain radius}} \]
\[ = \frac{3.25 \times 10^3}{2\pi \times \frac{200}{60} \times \frac{0.1399}{2}} \]
\[ = 2.22 \text{ kN} \]

11.6 CHAIN OPERATION

The effectiveness of a chain drive is dependent on correct installation, operation and maintenance. While chains are built to absorb a lot of punishment (consider the chain drive on a motor cycle) the factors that affect a chain's life and performance are:

- Correct choice of chain and chain wheel to transmit the required power and speed.
- Correct installation in terms of chain tension and alignment.
- Correct lubrication and long-term maintenance.
Chapter Eleven

11.29

Chain selection has already been discussed in Part 11.5 and in practice selection is guided by the chain supplier. Companies marketing chain have a wealth of experience in specifying the most suitable chain drive for a given application and their advice should always be requested.

Chain installation is, however, open to a large degree of malpractice. The fact that a chain appears to couple two shafts successfully may hide the occurrence of the chain being too tight or too loose, causing over-stressing of the components, vibration, noise and fatigue failure.

The chain sprocket/wheel may not be aligned, again causing noise, excessive wear and premature chain failure.

CHAIN ALIGNMENT

Make sure that the shafts are properly supported in bearings. Shafting, bearings and foundations should be suitable to maintain the initial alignment and not deflect excessively under load. Wheels should be arranged close to bearings.

Accurate alignment of shafts ensures even load distribution across the chain width and contributes substantially to maximum chain life. Figure 11.14 illustrates an approved method of obtaining alignment.

![Fig. 11.14 Method of Chain Alignment](image)

The straight-edge ensures that the chain wheels are in the same plane and a spirit level check of the shafts ensures that the shafts are horizontally aligned. A visual check, once the chain is installed, also indicates misalignment if the chain path is twisted.

CHAIN TENSIONING

Correct chain tension assists in preventing the chain from becoming misaligned with the wheels and, for a reversing drive, reduces backlash. In use, a chain wears and elongates, so some form of tension adjustment is essential.

The preferred method of chain tensioning is to move the position of one of the shafts. (Figure 11.15)

If shaft movement is not possible, an adjustable idler wheel engaging with the unloaded side of the chain is recommended.
Belt and Chain Drives

Fig. 11.15 Chain Tensioning Methods

Generally:
- The idler should have the same number of teeth as the pinion.
- The idler should be mounted so that at least three teeth are engaged with the chain.
- The idler size and speed should comply with the catalogue selection criteria.

The chain should be adjusted so that, as shown in Figure 11.16, with one side of the chain drive tight, the other side can be moved a total distance A, where:

\[
\text{Total movement (A) mm} = \frac{\text{centre distance (B) mm}}{\text{a load factor}}
\]

The load factor is 25 for "smooth" drives and 50 for shock-loaded drives.

If the drive system is vertical the total movement A should be half the chain pitch.

Fig. 11.16 Chain Tension Setting
**CHAIN REPLACEMENT**

As the chain wears it becomes longer owing to increased clearance in the many bearing surfaces. The change in length in a chain is used as a means of determining if the chain is due for replacement. A good guide is to compare the worn chain length with what it should be, that is, the number of links multiplied by the pitch. If the worn chain has increased by 2% it should be replaced.

If no means are available to adjust the chain tension it should be replaced when its length has increased by 1%.

**CHAIN LUBRICATION**

There are four basic methods of chain drive lubrication: manual, dripfeed, bath lubricant and oil stream. The lubrication method used is dictated by the drive application but basically, as the transmitted speed or power increases, so the lubrication method must improve.

The relationship between transmitted power, speed and lubrication method is shown in Figure 11.17.

![Fig. 11.17 Chain Lubrication Methods](image)
Chapter Twelve

GEARS AND GEARING

12.1 INTRODUCTION

The toothed gear has long been a familiar element in mechanical engineering and its importance seems likely to increase as development proceeds. Early examples of gears were cumbersome, noisy and inefficient, owing to the lack of suitable materials, design technique and manufacturing facilities. Improvements in these respects have more than kept pace with advancing demands and toothed gearing is now confidently employed at speeds and power outputs that could not have been attempted a few decades ago.

Gears are also used to transmit power between rotating shafts which are not in the same straight line and which have, in general, different speeds. Other means are possible in some circumstances, but where a compact drive is necessary, or where “timing” of the shafts is essential, the toothed gear usually surpasses all competitors in simplicity, reliability and efficiency.

Examples of the use of gearing occur in the conventional automobile. The camshaft which operates the valves has to run at half the crankshaft speed and in many instances it is driven from the crankshaft through gearing. In order to permit the engine to run within a reasonable economical speed-range whilst driving the vehicle at different speeds, the gearbox contains several pairs of gears, the drive being taken through any particular pair at will, to suit the conditions under which the vehicle is running.

From the propeller shaft which runs parallel to the length of the vehicle, the drive is taken to the rear axle through another pair of gears, in which there is a reduction in speed besides a change in the direction of the drive. The rear axle gear assembly also contains the differential, which is a gearing assembly designed to divide the driving effort equally between the two rear wheels.

In contrast to the small gears used in automobiles, gearing in diameters up to six metres is used in ship propulsion, where the economical speed of the steam turbine may be about 3000 r/min and that of the propeller about 250 r/min.

In industrial service, gearing is very extensively used, largely because of the widespread application of electric motor drives. The economical running speeds of electric motors of moderate output range from about 750 r/min to 1500 r/min, whereas the greater proportion of industrial machinery runs at considerably lower speeds, usually from about 50 to 200 r/min. In such applications, toothed gearing forms a compact and efficient transmission, the availability of which makes it possible to have driving and driven machinery each running at its own most economical speed.

12.2 GEARING NOMENCLATURE

Several terms are used to describe components of a gear, particularly during calculation of gear sizes. Figure 12.1 illustrates the terms specified.

*Pitch Circles* are the outlines of the imaginary smooth rollers or friction discs that form the drive system. Most gear dimensions are taken on or from these circles.

*Pitch Diameter* of a gear is the diameter of its pitch circle.

*Addendum* is the height from the pitch circle to the top of the tooth.

*Dedendum* is the tooth depth below the pitch circle. It equals the addendum plus clearance.
**Root Diameter** of a gear is the diameter at the base of the teeth spaces. It equals the pitch diameter minus twice the dedendum.

**Circular Pitch** is the length of the arc of pitch circle between similar faces of successive teeth.

**Module** is the pitch diameter divided by the number of teeth. It is used as a basic classification of gear tooth size.
Bottom Clearance is the shortest distance between the top of a tooth on one gear and the bottom of the mating space in a second gear with which it engages.

Base Circle Diameter is the diameter of the circle from which the involute tooth form is generated.

Line of Action is the common tangent to the two base circles which passes through the pitch point of a pair of mating gears.

Pressure Angle is the acute angle formed between the line of action and the common tangent to the two pitch circles which pass through the pitch point. The pressure angle is normally 20°.

Backlash is the shortest distance between non-driving surfaces of adjacent teeth in mating gears.

Outside Diameter is the overall dimension of the gear. It equals the pitch diameter plus twice the addendum.

12.3 GEAR TYPES

Toothed gearing is described on the basis of relative shaft positions.

The simplest examples are those used for parallel shafts such as in the conventional clock or in an automotive gearbox. These gears are classified as SPUR gears if their teeth are parallel to the shaft centre line.

Figure 12.2 illustrate some types of gears.

In a gear system, the smaller gear is normally called the PINION and the larger gear is called the WHEEL.

Being of simple form, spur gears can be accurately produced and operate at high efficiency. Tooth contact, owing to the involute tooth form, provides a rolling drive action rather than a sliding motion. Tooth contact at any instant occurs along a line parallel to the axis of the gear and tooth load produces no axial thrust.

Spur gears give excellent results at moderate peripheral speeds but tend to be noisy at high speeds unless made to a high degree of accuracy - or if one of the gear pair is made from a suitable non-metallic material.

As well as conventional gear systems, a spur gear drive may exist as an internal gear system. It may be used to convert rotary motion to linear motion by using a spur gear driving a rack. An example of a rack and pinion is the method used on a lathe for moving the saddle along the lathe bed.
One disadvantage with a spur gear, as previously stated, is that at high speeds the gear system tends to be noisy. This is because the drive is transmitted by a series of sudden contacts between the gear teeth.

*Helical Gears* were developed to provide a smoother transmission of power. As shown in Figure 12.3, helical gears have teeth that twist uniformly throughout the length (or face) of the gear.

The angle, at the pitch line of the helix, is called the *helix angle*. The larger the helix angle, the greater the number of teeth in helical contact across the face of the gear. However, the greater the helix angle, the greater the end thrust transmitted through the gear shaft.

The end thrust on the shaft bearings may be avoided by combining two pairs of helical gears, one with a left-hand and the other with a right-hand lead of helix. When the two gears are cut on one single blank, they are called *herringbone* gears.

![Fig. 12.3 Helical Gears](image)

*Bevel Gearing* (Figure 12.4) is used to connect intersecting shafts. The teeth of bevel gears are formed about the frustums of cones whose common vertex is the point where the shaft's centre lines would meet if prolonged.

The cones are called pitch cones since they roll upon each other, and on them the teeth are pitched. When bevel gears are the same size the shafts lie at an angle of 90° and are called *mitre gears*.

![Fig. 12.4 Bevel Gears](image)
A *crown gear* is one in which the sides of the pitch cone make an angle of 180° with each other, i.e. the pitch cone has no altitude and is therefore a flat surface. Note that the tooth action of bevel gears is analogous to spur gears, the teeth making line contact parallel to the pitch line.

There is no longitudinal sliding between the teeth, but end thrust is developed under tooth load, acting away from the apex and tending to separate the gears. Thrust bearings are therefore used with bevel gears to keep the gears in the correct relative position.

Note that, as shown in Figure 12.4, a bevel gear may also have helical teeth to provide a smoother drive system.

*Spiral Bevel Gears* (Figure 12.5) were introduced to obtain a more gradual tooth engagement which is necessary for high-speed operation and also to improve the load carrying capacity. Straight tooth bevels tend to be noisy at high speeds.

Spiral bevel gears bear the same relationship to straight bevels that helical gears do to spur gears.

A variation of the spiral bevel gear is the *Hypoid Gear*, where the pinion shaft is offset from the centre line of the gear. The hypoid gear is mainly used in motor car back axle drives, giving a smooth engagement at high speeds and high load carrying capacity.

---

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---

Internal Gears provide a compact drive for transmitting motion between parallel shafts which rotate in the same direction. The tooth form is usually a straight spur tooth because an internal helical tooth is difficult to manufacture.

As indicated in Figure 12.6, a pair of internal gears comprises a spur pinion meshing with an internal wheel, which has the teeth cut on the inside of its rim.

The teeth of the wheel, which project inwards, represent the spaces of an external wheel of the same size, and equal internal and external gears have the same tooth contour.
Spiral Gears are used to connect shafts whose axes lie at an angle to one another but do not meet. A spiral gear (Figure 12.7) is really a helical gear mating with a second helical gear but the helix angle on each gear need not be the same angle.

Spiral gears operate on close to a point contact and always have longitudinal sliding between mating teeth. The result is that spiral gears have very poor wear-resisting properties which cannot be improved by increasing the tooth face width.

For this reason spiral gears are only used successfully for light duty drive systems.

Worm Gears are used to transmit motion between shafts at right angles, which do not lie in a common plane.

While worm gears serve the same purpose as spiral gears, the major difference is that a worm gearing system operates with a line contact being obtained (whereas a spiral gear drive has point contact only). Thus, a worm gear drive may carry higher loads.

A pair of worm gears consists of a cylindrical worm having helical teeth or thread similar to a helical gear, meshing with a wheel having a concave face (see Figure 12.8 and Figure 12.16).

In most cases worm drives are used for large speed reductions between shafts at right angles; however, they can be substituted for a spiral gear drive at any angle. The speed reduction ratio equals the number of teeth on the wheel divided by the number of threads (or starts) on the worm.
Speed ratios of 70:1 are possible, with a transmission efficiency of at least 80%.

Fig. 12.8  Worm Gear

12.4 GEARING CALCULATIONS

Gear terminology was discussed in Part 12.2 and reference should be made to Figure 12.1 to identify the gear segments. Dimensional relationships between the various gear parameters are given below, for reference only.

Remember that module is specified as the length, in millimetres, that each tooth would occupy if the teeth in the gear were spaced along the pitch diameter.

\[
\text{Module} = \frac{\text{pitch diameter}}{\text{number of teeth}}
\]

Also,

\[
\text{Addendum} = \text{module}
\]

\[
\text{Dedendum} = \text{module} \times 1.25
\]

Centre distance between two gears = \[
\frac{\text{pitch diameter of } A + \text{pitch diameter of } B}{2}
\]

\[
= \frac{(\text{teeth in } A + \text{teeth in } B) \times \text{module}}{2}
\]

Considering two gears, as shown in Figure 12.9. The speed ratio between the two gears is proportional to their pitch diameter which, in turn, is proportional to the number of teeth.

\[
\frac{\text{pitch diameter of } A}{\text{pitch diameter of } B} = \frac{\text{speed of } B}{\text{speed of } A}
\]

\[
\frac{\text{number of teeth on } A}{\text{number of teeth on } B} = \frac{\text{speed of } B}{\text{speed of } A}
\]
EXAMPLE 1

In the gearing system shown in Figure 12.10, the input speed is 800 r/min. The input shaft drives the intermediate shaft via an 18:130 tooth ratio and the intermediate shaft drives the output shaft via an 85:100 ratio. Calculate:

(a) The centre distance between the input and intermediate shaft if the gears have a module of 2.
(b) The output shaft speed.

\[
\text{module} = \frac{\text{pitch diameter}}{\text{number of teeth}}
\]

\[
\text{pitch diameter} = \text{module} \times \text{number of teeth}
\]
For the 18 tooth gear:

\[
\text{pitch diameter} = 2 \times 18 = 36 \text{ mm}
\]

For the 36 tooth gear:

\[
\text{pitch diameter} = 2 \times 130 = 260 \text{ mm}
\]

\[
\text{centre distance} = \frac{30 + 260}{2} = 148 \text{ mm}
\]

(b)

Input speed = 800 r/min

Intermediate shaft speed = \(800 \times \frac{18}{130}\) r/min

Output shaft speed = \(800 \times \frac{18}{130} \times \frac{85}{100} = 98.15\) r/min

EXAMPLE 2

The gearing system shown in Figure 12.11 represents a drive to a planing machine table. Input to the driving pulley is transmitted through a speed-reducing gear train to a bull wheel. The bull wheel drives a rack which is attached to the underside of the planing machine table. If the pitch of the gear rack teeth is 25 mm, calculate the planing table speed.

![Gearing System](image)

Speed of the bull wheel = \(300 \times \frac{25}{85} \times \frac{35}{130} \times \frac{30}{75} = 9.5\) r/min

For each revolution of the bull wheel, the rack would be driven a distance equal to the number of teeth on the bull wheel multiplied by the pitch of the rack teeth.
So,

\[
\text{rack linear speed} = \text{bull gear speed} \times \text{number of teeth on bull gear} \times \text{rack pitch}
\]

\[
\text{rack speed} = 9.5 \times 75 \times 25 = 27.81 \text{ mm/min}
\]
\[
= 0.297 \text{ m/s}
\]

**EXAMPLE 3**

Figure 12.12 shows a mechanism of a dial test indicator - a very small movement of the stem is enlarged by gearing and displayed on the scale. If the pitch of the rack teeth is 1.2 mm, calculate the angle through which the needle moves if the stem is lifted 0.02 mm.

![Fig. 12.12 Geared Mechanism](image)

If the 30-tooth pinion is to rotate one complete turn it would move the stem 30 teeth (or 30 x 1.2 mm). If the stem is lifted 0.02 mm, then the 30-tooth pinion is rotated:

\[
\frac{0.02}{30 \times 1.2} \text{ turns} = \frac{0.02 \times 360}{30 \times 1.2} \text{ degrees}
\]
\[
= 0.2^\circ
\]

due to the 100:20 gearing ratio:

Pointer rotates \(0.2 \times \frac{100}{20} = 1^\circ\)

**12.5 GEAR CONSTRUCTION**

**TOOTH SHAPE**

For various theoretical and practical reasons (mainly to provide a rolling action rather than a sliding action) the tooth shape normally used for gearing is the *involute*. This is the shape which is traced out by the end of a cord as it is unwound from a base circle.
Involute form teeth have a smooth rolling action which keeps noise and wear to a minimum and reduces the demand for lubrication. The involute tooth form allows the gear centre distance to be varied slightly without greatly affecting gear performance.

A cycloidal tooth form may be used for gearing where high power is to be transmitted.

The main requirements for suitable gear tooth form are:

- teeth must have a profile which ensures a constant velocity ratio.
- relative motion of one gear on another should be more of a rolling nature than a sliding one.
- the arc of engagement should be long enough so that at all times more than one pair of teeth is in mesh. In practice this is not always fulfilled.
- tooth profile should approach a cantilever beam of uniform strength.
- tooth profile must be capable of being generated by simple cutters.

GEAR MATERIALS

The material chosen for a gear depends on the gear's application and on the method of gear manufacture. If the gear is to be moulded, for instance, a cast iron, aluminium alloy or plastic is used.

Materials can be classified as follows:

- ferrous - iron and steel - used in castings, fabricated, forged, stamped or extruded bar form.
- non-ferrous - gunmetal, bronze, aluminium alloy - used mainly as a casting or in bar form.
- non-metallic - plastics, resin bonded paper or fabric - moulded or fabricated from sheets or bar.

GEAR PRODUCTION

Gears may be moulded in metallic or non-metallic materials. If the gear teeth are to be machine cut, for small quantities a suitable milling cutter is used to produce one tooth at a time, with the gear blank indexed around in a dividing head.

For large-scale gear production, the teeth are "generated" by forming the teeth on a gear shaper. This machine shapes the gears using a cutter that resembles either a gear wheel or a rack. This method can be used to produce internal spur gears as well as external spur gears.

After machining, gear teeth may be hardened. A popular method for automotive gears is to manufacture the gears from 0.4% carbon steel and then the teeth are flame hardened or induction hardened. This process means the gears are rapidly heated and then quenched to produce a hard-surfaced skin.

For precision gearing the teeth may be ground after hardening.

12.6 GEAR SYSTEMS

Figure 12.13 illustrates a very basic gearing system.

Figure 12.14 shows a heavy-duty vehicle gearbox, providing a selection of output gear ratios.
An in-line helical gear reduction gearbox is shown in plan view in Figure 12.15 with an associated view of the gears. As a reduction box, drive would be from C to D and then B to A. Note the differing shaft diameters for the high speed input and the low speed, high torque output.

Measure the gear pitch diameters and estimate the speed reduction of the system.
A worm reduction gearbox is shown in Figure 12.16. Input is to the smaller worm shaft. Note the use of a fan and cooling fins on the gearbox casing.
A simplified diagram of an automotive type gearbox is shown in Figure 12.17. The gearbox is called constant mesh or synchromesh because all the gears are running in mesh.

It is only necessary to move the sleeves B or E, which are splined to the output shaft, to obtain the gear change required.

- For top gear, sliding sleeve B will engage with A
- For second gear, sliding sleeve B will engage with C
- For low gear, sliding sleeve E will engage with D
- Reverse gear, sliding sleeve E will engage with F
It should be noted that the output gears C, D and F are NOT attached to the output shaft, they only revolve around it.

Fig. 12.17 Principle of an Automotive Gearbox

Fig. 12.18 Vehicle Transfer Case
Figure 12.18 shows another type of vehicle gearbox - called a *transfer case*. A transfer case would be used on a heavy-duty four-wheel drive vehicle and is situated between the main gear box and the wheel drives. The unit shown includes:

- A selectable low gear ratio.
- An output to the rear wheels and a selectable output to the front wheels.
- An additional output, a power take off, to drive external equipment.

### 12.7 GEAR SYSTEM SELECTION

There is such a wide range of different types of gearboxes to choose from that it is difficult to decide which will be best for a particular application. Often several types would be suitable, but on some applications the mechanical requirements and the working environment point to only one possible solution to the problem. The final choice is almost invariably a question of economics and here the selector should examine installation, maintenance and running costs, as well as the initial cost of the equipment.

It is assumed that the loading on the gearbox has been assessed and the correct application factor chosen to obtain the life and reliability required. Reliability is inherent in every type of gearbox, providing that it is well designed, manufactured to a good standard of quality, chosen with sufficient capacity for the duty involved and is correctly lubricated.

### SELECTION CONSIDERATIONS

The factors affecting the choice of a gearbox may be classified into three main groups as shown.

<table>
<thead>
<tr>
<th>Economic Aspects</th>
<th>Mechanical Requirements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Cost</td>
<td>Gear Ratio</td>
</tr>
<tr>
<td>Installation Costs</td>
<td>Speeds</td>
</tr>
<tr>
<td>Running Costs</td>
<td>Power</td>
</tr>
<tr>
<td>Maintenance Costs</td>
<td>Nature of Loading</td>
</tr>
<tr>
<td></td>
<td>Direction of Shaft Axes</td>
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<tr>
<td><strong>Working Environment</strong></td>
<td>Direction of Rotation</td>
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<tr>
<td>Allowable Noise Level</td>
<td>Axial Thrust Loads</td>
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<td>Alignment Variations</td>
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<td>Accessibility</td>
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<td>Restricted Space</td>
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<tr>
<td>Weight Limitations</td>
<td></td>
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<tr>
<td>Ambient Temperature Range</td>
<td></td>
</tr>
<tr>
<td>Rough Handling</td>
<td></td>
</tr>
<tr>
<td>Atmospheric Pollution</td>
<td></td>
</tr>
</tbody>
</table>

### 12.8 CAMS AND FOLLOWERS

A cam is a mechanical machine element that is used to drive a follower.

Cams are among the most useful and important machine elements, especially for control of movements of automatic machinery. The cam and follower mechanism is simple but is able to produce complex movements that could not be obtained with a linkage.

Cams are generally cylindrical but may also be of a sliding type.
Figure 12.19 shows cams that produce a translating motion and Figure 12.20 shows the same cams but producing an oscillating motion. Considering only the rotary type cams, as shown in Figure 12.19 and Figure 12.20, these cams may be classified into face cams and cylindrical cams.

An example of each is shown in Figure 12.21. The face cam shown is the familiar type used on an internal combustion engine to operate valve gear.

The cylindrical (or barrel) cam shown is, in this case, an adjustable type. An application of a cylindrical cam is to cause various movements on an automatic lathe.

The followers used to transmit the cam motion are classified by the type of contact face, as shown in Figure 12.22.
The choice of follower depends on the complexity of the cam profile.

The *knife-edge follower* will track the most complex cam but the end wear is high.

The *flat-faced follower* may only be used on cams where the slope change is small, but this type of follower will accept sudden face loads and has a large surface area to minimise wear.

The *roller follower* has low frictional losses but should not be subjected to impact forces.

The *mushroom follower* is used in applications between the flat and roller followers.

The output motion from a cam follower is generally in the form of recognisable geometric form, such as:

- uniform velocity
- uniform acceleration/deceleration
- simple harmonic motion.

The design of a suitable cam is reasonably straightforward in terms of cam profile but the overall system design is complex because of the impact stresses and inertia forces caused by a cam operating at high speeds. Very high point loads occur between the cam and follower which can cause breakdown of lubrication, overstressing of the material or fatigue failure owing to repeated impact.

### 12.9 FLYWHEELS

A flywheel is a rotating member that acts as a storage reservoir for energy when work is not "consumed" at as fast a rate as the power is supplied.

When the work being done is greater than the work input, the flywheel gives up some of its stored energy to supply the deficiency. Unlike other forms of energy storage, a flywheel stores and gives up the energy at 100% efficiency.

An example of flywheel use is an internal combustion engine. Reciprocating prime movers are characterised by power pulses; in diesel engines for instance most of the power is generated just after the fuel ignites, but a properly designed flywheel keeps the speed fluctuations within acceptable limits and makes it possible for the engine to deliver work at a constant rate.
Many driven machines require power in fluctuating amounts. The input to a compressor varies. Machines such as punches, shears and presses require a low power approach sector of the cycle with a high power work sector and then a low power retract sector.

In the case of a machine such as a punch, which has a peak power requirement for a fraction of a revolution, a smaller drive motor may be used - so small in fact that the flywheel may contribute the major part of the energy during the work part of the cycle. The prime mover then again stores energy during the remainder of the cycle to bring the flywheel back up to its original speed.

The amount of flywheel speed variation depends on the application, based on the maximum/minimum speed. For example:

<table>
<thead>
<tr>
<th>APPLICATION</th>
<th>MAX SPEED - MIN SPEED</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>AVERAGE SPEED</td>
</tr>
<tr>
<td>Generators</td>
<td>0.002</td>
</tr>
<tr>
<td>Geared Drive</td>
<td>0.02</td>
</tr>
<tr>
<td>Reciprocating Pumps</td>
<td>0.04</td>
</tr>
<tr>
<td>Punch, shears</td>
<td>0.05</td>
</tr>
<tr>
<td>Crushers</td>
<td>0.2</td>
</tr>
</tbody>
</table>

To achieve maximum energy storage in a flywheel requires a large rotating mass but as the mass increases the cost and bearing loads increase. The energy stored is in the form of kinetic energy, calculated from the expression:

\[
\text{kinetic energy of rotation} = \frac{1}{2} \times \text{moment of inertia} \times (\text{angular velocity})^2
\]

As the moment of inertia is a function of the flywheel mass and the radius at which that mass occurs, it can be seen that if the mass is located at maximum radius, the flywheel will store more energy. For this reason flywheels consist of a heavy rim supported by a thin centre web or spokes.

Perhaps one of the most interesting applications of a flywheel is an experimental bus developed in Europe that runs on stored flywheel energy. The horizontally-mounted flywheel is initially driven to a speed in excess of 30 000 r/min by an integral electric motor. The electricity supply is then disconnected and the bus has enough stored energy to complete its run. On downhill parts of the run the braking energy is fed back to the flywheel. At the end of the run the electric motor is reconnected to "charge up" the flywheel for the return journey.
Chapter Thirteen

BOILERS

13.1 INTRODUCTION

The knowledge that heat applied to water will produce steam is the basis of a process that has been in use in the world for many centuries. This knowledge, however, was not recognised as a source of power for engines until the 18th century. Reciprocating engines were among the first to use this source of power, but they were slow-moving, large and cumbersome, and quickly lost their popularity when alternative power sources were discovered.

Although steam lost popularity (with the reciprocating engine) it continued to be used widely in power generating plants. Today steam is used extensively as the medium for transferring heat energy from the boiler or reactor to the engine or turbine.

CHANGE IN STATE

Matter is defined as anything that occupies a space. Therefore all objects consist of matter.

Matter can exist in one of the three physical states: as a solid, a liquid or a gas. When a transfer of heat occurs, matter in one form may be changed to another. This change is called a phase change.

When water is heated there is a phase change from a liquid to a vapour (steam). This phase change of water to steam occurs at 100°C (at atmospheric pressure). With this change in phase there is an associated change in volume that increases the original volume 1600 times. When this expansion, owing to the increase in volume, is contained, such as steam formed in a boiler, then the continuous generation of the steam brings about an increase in pressure.

TYPES OF STEAM

1. Wet Steam

Wet steam is the term used to describe steam that has been contained in the same vessel as it was generated. This steam, which is formed at constant temperature, holds small droplets of water in suspension.

2. Dry Saturated Steam

Dry saturated steam occurs when the wet steam is taken away from where it was generated and further heated. During this stage all the water droplets in suspension turn to steam and the steam becomes transparent.

3. Superheated Steam

Superheated steam is the final condition of the steam. This condition is produced by further heating associated with a rise in temperature. Superheated steam is used in turbines and high speed engines.

STEAM GENERATION PLANT

The steam generation plant is a complex system that contains three basic sub-systems:

1. The boiler.
2. The engine supply and ancillaries.
3. The cooling and condensate.

A schematic layout for a steam generating plant showing these three systems is shown in Figure 13.1.
Boilers

Fig. 13.1 Steam Generating Plant

THE MAIN COMPONENTS IN THE SUB-SYSTEM

1. **Boiler**
   The design of the boiler relates to the two basic cycles that operate in conjunction with one another to produce steam. The first is the heat cycle through the boiler. Here pulverised coal, gas or fuel oil is ignited in the furnace and the resulting hot gases are channelled through the boiler so that maximum heat transfer can be obtained.

   The second is the water cycle through the boiler that produces steam. Water enters this cycle via the feed pump, travels through the economiser and into the boiler drum. The steam that emerges is passed through a superheater and then to the engine to do work.

   From the above cycles it should be appreciated that the greatest intensity of heat must be near to the point where the steam leaves the boiler.

2. **Feed Pump**
   The function of the feed pump is to deliver feed water into the boiler against the pressure of the water already in the drum. In modern installations a multi-stage centrifugal pump is used for this purpose. The feed water is pre-heated and de-aerated (oxygen removed) before entering the feed pump.

3. **Economiser (Figure 13.2)**
   Economisers are heat exchange units. Their function is to heat the water with either furnace gases or steam "bled" from the system. The main feature of the economiser is that it uses heat that would normally be lost to the system. This results in less heat having to be applied to the water at a later stage in order to produce steam.

4. **Superheater (Figure 13.2)**
   The superheater is a heat exchange unit and is therefore similar to the economiser. The superheater however works at a much higher temperature than the economiser. The wet steam from the boiler enters the
superheater and passes through tubes that extend into the main body of the furnace. Here the tubes are subjected to the intense heat of the furnace gases and any moisture in the steam is quickly removed.

When the steam leaves the superheater it possesses all the characteristics of a gas and is ready for work. The steam (gas) is high pressure, high temperature and transparent.

5. Engine or Turbine
The engine or turbine converts the kinetic (heat) energy in the generated steam (gas) into mechanical energy. This mechanical energy is obtained when the steam expands through the engine, giving up its stored power.

6. Condenser (Figure 13.3)
The condenser is another heat exchange unit but works in the reverse way to the superheater and economiser. In the condenser, heat is removed from the steam by the cooling water circulating through a series of tubes. Once the heat is removed from the steam, it condenses and condensate forms. The condensate formed in this way is used as feed water and is recycled through the boiler.

The condenser, besides being a heat exchange unit, is designed to reduce the back pressure on the engine.

7. Condensate Extraction Pump
The condensate extraction pump is designed to remove the condensate from the condenser and deliver it to the feed pump. This process is necessary because the condenser operates below atmospheric pressure and is therefore outside the range of the suction of the boiler feed pump.

8. Circulating Water Pump
The circulating water pump can be either a centrifugal pump or a propeller pump. Its function is to circulate the cooling water through the condenser.

In land-based steam-generating plants, cooling towers are used to cool the cooling water, but when the generating plant is situated on a river or ocean, this source is used to cool the water.

9. Cooling Tower
The cooling tower is used to remove the heat collected by the cooling water as it passes through the condenser. The cooling water is raised to the top of the tower, and allowed to cascade by gravity through a series of trays. At the bottom of the tower the water is collected and re-cycled through the condenser.

13.2 BOILER TYPES
There were two major problems that faced manufacturers when boilers were first designed. Initially they had to develop suitable materials to resist the high temperatures and pressures associated with steam generation.

Secondly, they had to produce a design so that sufficient steam could be generated to meet the increasing demands of larger engines.

The development and improvement in metal technology solved the first problem. The second problem was not solved until attention was focused on a change in the design of the boiler.

Early boilers were designed on the fire tube principle. These boilers, however, were limited in their rate of steam generation and were gradually replaced by water tube boilers when large steam flow rates are required.
1. FIRE-TUBE BOILERS (Figure 13.4)

Fire-tube boilers derive their name from the passage of the hot gases through the boiler. These hot gases are formed in the fire box and are exhausted to atmosphere through the smoke stack. In this boiler the tubes extend through the boiler drum and are surrounded by water. The hot gases are directed to pass through the tubes, hence the term "fire tube".
The steam rating of the fire tube boiler is determined by the size and the number of tubes in the boiler: the more tubes, the greater the surface contact with the water in the drum. This surface contact was further improved in the design of the Wet back boiler. In this boiler the water also surrounds the fire box.
2. WATER-TUBE BOILER (*Figure 13.5*)

Water-tube boilers take their name from the water being directed through the tubes while the hot gases surround the outside of the tubes. Early designs had better water circulating and steam raising properties than the fire tube boiler. Their design also allowed for an upper and lower water drum with connecting tubes. These connecting tubes greatly increased the surface area of the boiler available for heat transfer.

More advanced boiler designs increased the number of drums, and hence the number of tubes. This resulted in increased efficiency because of the extended path of the hot gases through the boiler. This in turn enabled greater volumes of steam to be generated in less time.

Practically all large modern boilers are of the water tube type. Among these boilers there is great variation in drum arrangement, and it is possible to have the walls of the furnace either air or water cooled.

---

**ADVANTAGES OF WATER-TUBE BOILERS (COMPARED TO FIRE-TUBE BOILERS)**

- The thin, strong tubes allow a rapid transfer of heat.
- Steam can be raised quickly owing to the comparatively small water capacity of the system, the greater number of tubes exposed to the hot gases, and the positive water circulation.
- The smaller diameter water and steam drums enable higher working pressures (up to 7000 kPa).
- Very high evaporative capacities (steam forming) are practical. They will take overloading more readily than fire-tube boilers.
- For the same evaporative capacity a water-tube boiler requires less floor space than the fire-tube boiler.
- Repairs can be made easily and rapidly.
- Flexibility for different fuels, firing methods and available space.
DISADVANTAGES OF WATER-TUBE BOILERS

- A failure in feed-water supply is serious because of the small amount of water contained in the system.
- It is extremely difficult to descale the inside surface of the boiler tubes.
- The feed water must be chemically treated to a high purity to prevent scale, corrosion and foaming.
- More care must be taken in the operation of these boilers.

13.3 BOILER DESIGNS

Although steam boilers are classified in many ways, the general procedure is to divide them into two principal groups:

*INTERNALLY FIRED* - the furnace is situated within the boiler.

*EXTERNALLY FIRED* - the furnace is outside the boiler.

The other means of classification, namely WATER TUBE or FIRE TUBE have already been discussed.

While the mode of boiler operation or the method of boiler construction may be used to describe a boiler type, specific design is referred to by either the manufacturer's name or by the name of the boiler's point of origin.

A Babcock and Wilcox Water-Tube Boiler is shown in Figure 13.6.

![Babcock and Wilcox Water-Tube Boiler](image)

The model shown is not a high-performance type, but a superheater may be added in the space between the water tubes and the steam drum to increase the energy content of the steam. Water circulation is by natural convection in the angled water tubes.
The mud leg is a collection and drain point for sediments that occur in the boiler water. The blow down cock is located on the side, external to the brickwork or casing lining.

Specially designed firebricks, secured between the inclined tubes and supported by a metal plate, baffle the hot gases and cause them to rise to the bottom of the steam and water drum. A second baffle, constructed similarly to the first, allows the gases to flow down between the tubes, then up again, before being liberated to the atmosphere. This arrangement enables the hottest water to receive the heat from the hottest gases, causing a rapid circulation. Consequently, steam is raised and maintained quickly and easily. The path of the flue gases is therefore from the furnace, up and around the inclined tubes between the front header and first baffle, to the bottom of the steam and water drum. Then down between the first and second brick-work baffles, and up between the second baffle and rear header, passing between the vertical tubes. Down to the flue, past the damper, into the stack, and away to the atmosphere.

The Under-Fired Multi-Tubular Boiler ([Figure 13.7](#)) is popular in Australia. It is a horizontal, externally fired, return tubular boiler, and is set in brickwork in which is incorporated a furnace, generally of good dimensions, designed to burn practically any type of fuel.

Approximately one-third of the boiler shell is subjected to the heat from the furnace. This restricts to a marked degree the permissible thickness of the boiler plate. A mudleg is fitted at the bottom of the boiler, towards the rear, to collect any sediment deposit. A pipe is connected to the mudleg at the bottom, and is led out through the side brickwork and plate of the boiler. To this pipe is connected the blow down cock.

Steam pressure for this boiler is up to 1100 kPa and it is claimed that it has greater evaporative power for its size than any other return tubular boiler.

![Under-Fired Multi-Tubular Boiler](image)

**Fig. 13.7** Under-Fired Multi-Tubular Boiler

The Economic Boiler ([Figure 13.8](#)) is familiar as a package-type boiler, providing a self-contained steam supply. It is considered to be an externally fired fire-tube design because its steel encased combustion chamber is not a pressure region of the boiler. The term "economic" implies its improved performance but it is the Scotch boiler principle applied to a Dry-back boiler.
The efficiency of the boiler is increased by directing the furnace gases through a series of tubes that not only increase the heat transfer area but improve the water circulation. The boiler is self-supporting in its special casing and this obviates the need for external brickwork. The flat surfaces on either side are supported by stay bolts.

The gases pass from the furnace tube to a rear combustion chamber, then through the lower nest of tubes to the smoke box at the front of the boiler. The gases return to the rear of the boiler through the upper nest of tubes to a connection to the exhaust stack.

13.4 BOILER MOUNTINGS AND ANCILLARIES

Boiler mountings are appliances fitted to maintain safe boiler working and to assist boiler operation. They are generally fitted to mounting blocks, made of mild steel in the form of short conical pipes, flanged at both ends. One end is riveted to the boiler shell and the other end is machined to receive the flange of a boiler mounting. This system permits easy removal of the mounting for repair or replacement.

Smaller mountings, such as pressure gauges, may be screwed directly into a strengthened portion of the boiler shell.

Typical boiler mountings are shown in Figure 13.9. Note that many boiler mountings are required to be fitted by law, because the mounting is a safety feature. In some case, more than one such mounting must be fitted, such as two water gauge glasses or two safety valves.

Regulations ensure that the safety-related mountings are tested at regular intervals, and that the boiler is operated by a suitably qualified person.
1. WATER GAUGE GLASS

The water gauge consists of two fittings supporting a toughened glass tube and mounted on the boiler drum. The function of the water gauge is to indicate the water level in the boiler. This function is especially important on water-tube boilers, where the amount of water entering the boiler drum is balanced against the steam being generated and drawn off.

2. PRESSURE GAUGE

A pressure gauge is fitted to the top portion of the boiler drum to indicate the pressure of steam being generated. The gauge has an isolator valve to allow the gauge to be removed while the boiler is in operation, and is connected by a U-tube so that a water barrier exists between the gauge and the steam - to reduce thermal effects on the gauge.
3. SAFETY VALVE  (*Figure 13.11*)

At least two safety valves, each of ample flow rate, are legally required. They are fitted to the highest part of the steam space and to a superheater if one is fitted. They allow any excess steam pressure above the safe working pressure to escape to the atmosphere.

Spring-loaded safety valves are the type fitted to all modern boilers. The ordinary spring-loaded valve fitted to the majority of medium pressure boilers is shown in Figure 13.11.

![Fig. 13.11 Spring Loaded Safety Valve](image)

4. MAIN STOP VALVE  (*Figure 13.12*)

The main stop valve is fitted to the highest part of the steam space on the boiler. One is fitted directly to the shell between the boiler and the main steam line and, if two or more boilers can be connected together, an additional intermediate stop valve or approved isolating valve is fitted between the main stop and the main steam range.

Its function is to control the passage of steam from the boiler to the main steam line supplying the various items of plant, each of which will also have its own stop or isolating valve.

![Fig. 13.12 Stop Valve](image)
5. MANHOLE

Manholes - or in smaller boilers, handholes - are essential openings in the boiler shell for construction, cleaning, repair and inspection of the boiler.

To allow the manhole to be fitted internally it is an elliptical shape and mates with the boiler shell via a gasket. As the boiler pressure increases, the manhole seal-loading increases.

6. BLOW-DOWN VALVE

When water containing dissolved solids is pumped into a boiler, the water is evaporated but the solid matter will remain in the boiler. Over a period of time, the quantity of dissolved solids in the boiler will rise, and the function of the blow-down valve is to maintain a reasonable level of density of the water in the boiler.

A secondary function is to allow the boiler to be emptied prior to overhaul or refit. The blow-down apparatus consists of cocks and pipes, connected so that the operator can allow all or part of the contents of the boiler to escape as required. It is divided into surface blow-off equipment (or scum line) and bottom blow-off equipment.

The boiler water is tested at regular intervals and the boiler "blown down" when the total dissolved solids (measured in parts per million) exceed an acceptable level.

7. FEED VALVE

The feed check valve, and a stop cock or valve, are always fitted as a pair. The stop valve is fitted to the shell between the boiler and the check valve, so that repairs can be carried out on the feed line or pumps while steam pressure is in the boiler.

The feed check valve is a non-return valve which retains the water in the boiler and controls the flow of water from the feed line. It may be fitted to the front, side or top of the boiler, and an internal feed pipe is always fitted in conjunction with this valve.

If the valve is fitted to the top of the shell, the internal pipe carries the cold feed through the steam space and distributes it in the water space.

If the valve is fitted on the front or the side, the internal feed pipe carries the cool feed away from the shell plate where it would cause contraction every time the feed rate is changed. The internal feed pipe also has the effect of heating the incoming feed before it is mixed with the water already in the boiler.

8. HIGH/LOW WATER LEVEL ALARM

The water level alarm is designed to indicate when the water level goes outside the limits set for the boiler drum. The level alarm is linked with the level controller, an alarm only being sounded if the water level starts to approach a dangerous condition.

If the water level is too low, damage could occur to the boiler. If the water level is too high, there is a carry-over of water with the steam which causes problems at the point where the steam is to be used.
BOILER ANCILLARIES

**Superheater**
When steam is in contact with the water from which it was generated, it still contains a small percentage of water particles in suspension. In order to increase the heat content of the steam and evaporate the suspended water particles, the steam must be removed from the water space and more heat added. The function of a superheater is to perform this act.

The superheater receives wet steam at saturation temperature, and as heat is absorbed the steam dries out and then rises in temperature with a slight reduction in pressure due to the resistance offered by the superheater tubes.

**Economiser**
The function of the economiser is to absorb heat from the flue gas and add it to the feed water, thereby reducing the amount of heat required in the boiler. This helps to maintain economical usage of the boiler fuel. During this heat transfer, dissolved oxygen in the feed water may be liberated, giving rise to severe corrosion problems. It is normal therefore, for the water to be de-aerated before passing to the economiser.

**Air Preheater**
The air heater in a boiler plant reclaims some of the heat still left in the flue gases after they have passed through the economiser. The heated air is used to improve and accelerate combustion thus improving the efficiency of the plant.

### 13.5 PULVERISED FUEL BURNING

Pulverised fuel has been used since 1920, and is the standard method of fuelling many power station boilers. Some of the advantages of pulverised fuel (compared to solid fuel) are:

- greater surface area means better combustion, because there is a high surface area for the fuel to combine with the oxygen.
- the efficient use of lower grades of fuel. Practically any dry fuel may be burnt in a pulverised fuel furnace, but coal with a volatile content of about 20 to 25 percent is most suitable. Moisture content of the coal may be as high as 15 percent.
- increased efficiency, especially with large boilers on continuous load.
- larger boilers are possible than with conventional stoker-fired plant.

The first step in the process is a **pulverising mill** that reduces the input coal to a powder. The pulverising mill may contain a series of hammers (an impact mill), or a series of balls that roll within the fuel (a ball mill).

With each type of mill, an air **flow** carries the coal dust through to a classifier, which checks the size of the pulverised fuel and returns oversize coal to the mill for further treatment. Typical pulverised fuel size is 75 micrometres.

**Burners**
There are various types of burners in use and different combustion chambers, but the aim of every system is to distribute the burning fuel and resulting gases so that the heat is dissipated to the correct parts of the furnace. Figure 13.13 shows some combustion chamber arrangements.

(a) Vertical firing
A number of fan-shaped nozzles are set across the width of the furnace. The pulverised fuel and air mixture is directed vertically downwards towards the bottom of the furnace and then made to turn up again to pass
Boilers

through the combustion chamber. This gives a long path to the flame and is particularly suitable for coals of low volatile content.

(b) Horizontal firing
The burners are set in the front or rear walls of the furnace. The primary air and fuel is mixed, then a stream of secondary air is introduced to cause a turbulent flow condition.

(c) Tangential or Corner firing
The burners are set at each corner of the furnace and are directed to strike an imaginary circle of about two metres diameter in the centre of the furnace. Because the streams of fuels strike each other, extremely good mixing results. The body of flame produced is given a rotating motion that leads to longer flame travel and the spread of hot gases to fill the combustion chamber. Corner burners may be arranged to tilt up and down, to provide some measure of steam temperature control.

It is usual to use oil-fired burners for igniting the pulverised fuel burners, and the oil burners are sometimes adapted for pressure raising and steaming at low loads.

Fig. 13.13 Burner Arrangements
PIPEWORK AND VALVES

14.1 PIPE AND TUBE

Various references provide different definitions of the words pipe and tube. One popular definition is that pipes are dimensionally classified by the inside diameter or bore while tube is classified by the outside diameter.

The word pipe is by common usage used for pipelines that convey water, oil or gas over long distances whereas tube or tubing is used for shorter flow paths such as the heat transfer tubes within a boiler or heat exchanger or in machines to convey hydraulic fluids.

Another form of classification appears to be on relative size. Generally, small diameters (up to say 10 mm) are called tube and larger diameters called pipe.

Australian Standard 1074-1989 on the dimensions and properties of steel tubing states in the definitions section that the term tube is synonymous with the term pipe. Hence, in this text both terms will be used to describe a circular sectioned material used for conveying fluids.

PIPE MATERIALS

In engineering the commonly-used pipe materials are aluminium, copper, asbestos-cement, concrete, steel and plastic. Other lesser-used pipe materials are glass, lead, fibreglass and rubber.

The original Goldfields water supply scheme in Western Australia was partially constructed using wooden pipes bound with iron. This may not seem a very durable material until it is realised that the wooden pipes were not completely replaced with steel until the pipeline had been in operation for sixty years.

Aluminium pipe is used for some structural work as well as conveying fluids because it is light and has reasonable strength. Differing grades of aluminium allow for welded construction or facilitate pipe bending.

Aluminium forms a natural oxide coating which retards corrosion, or the aluminium may be anodised, an electro-chemical process that forms an anodic protective surface coating. Because aluminium conducts heat readily, it has been used for heat-exchanger applications.

Fibre-cement pipe is manufactured from a mixture of synthetic fibres, cement and silica (or sand). During manufacture the pipe density is carefully controlled by varying wall thickness and composition. Fibre cement pipe is not used for pressure applications, but for drainage. Pipes of 100 to 200 mm diameter are available produced in four-metre lengths. Joints between the pipes are normally made with coupling sleeves and rubber sealing rings.

Concrete pipes are similar to fibre-cement pipes but steel reinforcement may be incorporated within the concrete to provide additional tensile strength. The pipes are cast in a stationary mould or are centrifugally cast by rotating the mould on a horizontal plane.

Pipe diameters range from 100 mm up to 3000 mm, with lengths up to 2.4 metres.

Concrete pipes are used for sewer lines, main drainage, water supplies and for channelling creeks. Normally, one end of a pipe length has a larger diameter neck so that pipes may be joined together by using a rubber seal or by filling the pipe joint with mortar.
Iron pipe is a general classification for several different forms of iron (rather than steel) pipes:

(a) Standard grade cast iron pipes were used mainly for water supplies, sewerage pipe and house drains. In cases where the liquid conveyed is corrosive, the iron may be cement-lined, or this process is done in situ if the iron pipes became excessively corroded in the bore during use.

Pipes of up to 750 mm diameter are available, manufactured by a centrifugal casting process, producing pipes of up to six metres in length.

(b) Ductile spun cast iron is similar to grey cast iron but is specially prepared and heat treated to provide high strength and ductility by changing the form of the graphite structure in the iron.

Sizes from 80 to 750 mm diameter are available and may be supplied with a cement lining for corrosion resistance. Working pressures range from 4 MPa for the small sizes to 2.5 MPa for 600 mm diameter. Standard pipe length is 5.5 metres. Pipe lengths are joined together with a stepped end and jointing ring or an external flange and gasket system is available.

Full details of ductile spun iron pipe are given in Australian Standard 2280 - 1995.

Steel pipe is probably the most common type of pipe available because of its versatility and high performance characteristics.

(a) A.S. 1074-1989 provides information on steel tube suitable for screwing to A.S. 1722-1975 Pipe Threads. As the name of the Standard implies, this pipe is primarily intended for use with threaded fittings, but may also be welded.

The pipe is manufactured in Nominal Bore sizes, thus the diameter of the pipe bore will be close to a nominal dimension such as 6, 8 or 10 mm etc (up to 150 mm).

Note that because the outside diameter is to receive a thread, the outside diameter must be carefully controlled to a specific size. This means that if the pipe is to withstand a higher pressure, the pipe wall thickness must be increased. To maintain the same outside diameter (suitable for a thread) thickening the walls can only be achieved by reducing the inside diameter or bore.

For instance, a 25 mm nominal bore steel pipe is available in three types:

<table>
<thead>
<tr>
<th>Outside Diameter</th>
<th>Wall Thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light</td>
<td>2.65</td>
</tr>
<tr>
<td>Medium</td>
<td>3.25</td>
</tr>
<tr>
<td>Heavy</td>
<td>4.05</td>
</tr>
</tbody>
</table>

Note that the outside diameter of steel pipe made to A.S.1074 is of a suitable size for forming a screw thread. The pipes are normally supplied "screwed and socketed" which means that the ends are threaded with a tapered Whitworth form called a "British Standard Pipe Taper" and each pipe is fitted with one socket joint. The thread used is based on Imperial dimensions so a 25 mm pipe would be threaded 1" BSPT.
This type of pipe is normally formed by rolling a strip of steel sheet into a tubular form by a series of rollers and then welding the join. The whole process is carried out on a continuous strip and the weld area is often only detectable by inspecting the inside of the pipe.

Such a pipe is pressure tested at 5 MPa during manufacture and thus is usable for the majority of liquid and gas conveying systems.

(b) Seamless or solid-drawn steel tube is made, as the name implies, from a solid billet of steel which is first formed into a very thick walled tube and then, by successive working, formed into tube of the correct dimensions.

This type of product may be used for systems that demand a higher pressure rating than that which can be withstood by the seamed or welded pipe.

Seamless tube is usually classified by the outside diameter with three grades of wall thickness to allow for differing working pressures.

Solid drawn tube is available from 5 mm outside diameter upwards, with working pressures in the smaller sizes in excess of 150 MPa.

Steel piping generally has many advantages over pipes of non-ferrous materials:

- Relatively cheap and available in a wide range of sizes.
- Good resistance to shock and vibration.
- High resistance to external or internal pressures.
- External corrosion protection is possible by coating the steel with paint, polythene sheathing, hot dip zinc galvanising or bituminous coating.
- Internal corrosion protection is possible by lining with a material such as bitumen, cement, epoxy resin, paint or plastic coating.

_Copper pipe_ is produced by drawing the material out through a die which contains an internal mandrel representing the pipe bore. A vast range of pipe sizes is available, from 5 mm diameter up to 150 mm diameter and in wall thickness of 0.7 to 2.6 mm.

The pipe is available in an “as drawn condition” (work hardened) in straight lengths or in an “annealed condition” (softened) in straight lengths or in coils.

_Plastic tube_ is available in a wide range of sizes and materials:

(a) PVC (Polyvinylchloride) is produced by an extrusion process and may be specified as rigid or flexible.

Rigid PVC is referred to as Unplasticised PVC (UPVC). A popular use of this material is for reticulation systems for water but its excellent resistance to chemicals also means it may be used to transport acids and similar corrosive chemicals.

UPVC piping is classified as class 4.5, 6, 9, 12, 15 or 18. The classification relates to the maximum working pressure at 20°C, from 0.45 MPa to 1.8 MPa.

Note that as the temperature increases, the working pressure must be reduced. Also note that the outside diameter of the pipe is constant (for a given size) but the wall thickness increases to provide the range of pressure ratings. (See Table 14.2.)
Pipework and Valves

As a means of identification, UPVC is coloured:

- **Dark grey**: soil waste and vent pipe
- **Light grey**: electrical conduit
- **Orange**: conduit for use underground
- **Yellow**: gas pipelines
- **Off white**: pressure pipe
- **Cream**: sewer and stormwater pipe

Flexible PVC is probably best known in its form as green garden hose. It is also available as clear or in various colours and is utilised for transporting a variety of fluids.

Some of the properties claimed for PVC pipes are:

- Lightness - it is much lighter than steel or concrete.
- Varying degrees of flexibility - even a so-called "rigid" plastic pipe does have some ability to bend.
- Low cost - usually cheaper than other types of pipe.
- Fast assembly - the pipe is easy to fabricate and join together.
- Low frictional loss owing to the very smooth bore.
- Good chemical resistance - it is a non-conductor and does not suffer from galvanic corrosion.
- Low maintenance - painting is not required.

Some disadvantages are:

- Exposure to sunlight can cause problems. Some plastic pipes become brittle if exposed to direct sunlight and for "above ground" installations pipe must be used which contain a thermal stabilizer.
- PVC has a high coefficient of expansion so that expansion joints may be required and care given to the selection of suitable pipe-anchoring methods.
- Limited working temperature - continuous operation above 70°C should be avoided and low ambient temperatures are also undesirable.
- Low strength - plastic pipes need more support than metallic pipes and have a lower working pressure, especially at elevated temperatures. Internal erosion occurs if an abrasive material is conveyed.

(b) Polyethylene pipe (Polythene) is popular as the black plastic pipe used for conveying water on domestic and farming properties, or the small diameter tubes used for "trickle" irrigation systems.

Polyethylene is classified in two ways:

1. Type 30, 40 or 50 specifies the type of material used in the extrusion process.
2. Class 3, 4.5, 6, 9, 12 and 15 relates to the maximum working pressure for the pipe (0.3 MPa to 1.5 MPa). The pressures are, of course, modified at higher temperatures.

The pipe is available from 8 mm nominal bore up to 700 mm nominal bore. It is supplied in 6 metre lengths or the smaller diameter types are also available in coil form.

The black colouring owes to the addition of carbon black as a protection against ultra-violet light degradation should the pipe be exposed to sunlight.
(c) Polypropylene pipe has similar properties to PVC and polythene. It has a slightly higher temperature resistance than PVC and better chemical resistance than polythene but at low temperatures, polypropylene becomes brittle. Thus, its major application is the conveying of corrosive materials at elevated temperatures.

(d) ABS (acrylonitrile butadiene styrene) has higher qualities than PVC and polythene but also a higher cost. ABS is mainly used where the component is subjected to rough usage.

(e) Nylon is another very stable plastic used for pipes but because it is difficult to extrude, its diameter is normally limited to 25 mm. Nylon tubing is commonly used for compressed-air installations.

(f) PTFE (polytetrafluoroethylene) is an inert thermo-plastic which is virtually immune to chemical attack. It neither absorbs nor is wetted by water - as a non-stick material it is used to line non-stick frying pans. Its working temperature is 100°C to 250°C.

Unfortunately, although PTFE is a thermo-plastic it does not have a separate melt state so it cannot be extruded except as a granular powder. PTFE tube is available but is usually a thin tube, externally reinforced with PVC or wire braid.

14.2 PIPE CONNECTING METHODS

Pipe may be joined together by methods ranging from the simple welding of two ends together to the use of a complex fitting which not only joins the pipes but also allows the pipes to be separated easily at a later date.

Before discussing the methods used, it is necessary to be introduced to the terminology of pipe jointing components.

The diagrams shown in Figures 14.1 to 14.7 inclusive are of fittings used on threaded steel pipe but the same basic shape would be used for a fitting assembled by welding or a plastic fitting assembled by using a solvent cement.
Consider Figure 14.1.

A **Socket** is used to join two pipes together in line.

A **Reducing Socket** joins two pipes of differing diameters.

A **Union** is used to join two pipes together but the union itself may be disassembled without having to remove the unit from the pipe ends. This allows pipes to be removed and replaced as required.

Fig. 14.1 Sockets, Caps and Unions
Consider Figure 14.2.

A Plug is used to block an internal or female thread. The plug shown has a tapered thread so that the plug will tighten within the female thread.

Bushes are used to adapt from one size of female thread to another.

Hexagon Nipples join two female threads in line and a Reducing Nipple joins two female threads of differing sizes.

Fig. 14.2 Plugs, Bushes and Nipples
Consider Figure 14.3. This figure illustrates a series of Elbows, to join pipes at either 90° or 45°. The F & F refers to female to female and M & F refers to male to female.

Fig. 14.3 Elbows
Consider Figure 14.4. This figure illustrates a series of Bends, which have a similar use to an elbow but cause less restriction to flow of the conveyed fluid.

Fig. 14.4 Bends
Consider Figures 14.5 and 14.6. These figures illustrate a series of fittings for joining three or more pipes, either of similar size or of differing sizes.

Fig. 14.5 Elbows and Crosses
PIPE JOINTING

Figures 14.1 to 14.6 inclusive are threaded pipe fittings and intended for either connecting to each other - for instance a hexagon nipple could be used to join two female thread fittings - or for joining pipes.

The threaded joint may be a tapered male thread fitted into a parallel female thread, or both male and female threads may be parallel.
Normally, a jointing compound is used to seal the helical space between the mating threads. The jointing compound used depends on:

- the chemical characteristics of the fluid conveyed by the pipe line.
- the working temperature.
- whether the joint is permanent or needs to be disassembled at a later date.

A popular jointing compound is PTFE (polytetrafluoroethylene) in either a thin tape form or as a liquid. Threaded connections are normally only used with steel pipe but also have limited usage with plastic pipe.

Pipes may be joined by fusion, either high temperature fusion such as welding of steel, brazing of copper or lower temperature hot inert gas welding of plastics. Fusion jointing may also be used with plastics by using a chemical solvent on the parts to be joined. The solvent forms a thin layer of soft plastic and has the effect of combining the two parts.

Fibre cement, concrete and cast iron pipes may be joined by fitting the pipe into a socket which contains a rubber seal. This method of jointing has the advantage that pipe assembly is fast and the rubber seal allows limited pipe movement when, for instance, the pipe support system is not stable.

Compression fittings are used on smaller diameter pipes (up to 35 mm diameter). The compression sleeve deforms on assembly to grip the pipe. For lower pressure application, a flexible sleeve may be used. A flared fitting is formed by initially deforming the pipe end to the correct angle - the mating parts of the fitting conform to the deformed angle.

A flanged joint relies on a gasket to form a seal and on bolts to maintain seal pressure. The flanges may be initially formed on the pipe (cast iron), welded in position (steel, plastic) or fixed to the pipe with a rubber seal and separate retaining bolts (concrete). The flanges are made to an Australian Standard so that the diameter and hole pitch/size will align with similar flanges on valves and other pipework accessories.

Figure 14.7 illustrates the various types of pipe jointing methods and the suitability for differing pipe materials.
### Fig. 14.7 Pipe Jointing Methods

<table>
<thead>
<tr>
<th>TYPE OF JOINT</th>
<th>ALUMINIUM</th>
<th>FIBRE CEMENT</th>
<th>CONCRETE</th>
<th>CAST IRON</th>
<th>STEEL</th>
<th>COPPER</th>
<th>PLASTIC</th>
</tr>
</thead>
<tbody>
<tr>
<td>THREADED</td>
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<td></td>
<td>✓</td>
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</tr>
<tr>
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<td>✓</td>
</tr>
<tr>
<td>SOCKET &amp; RUBBER SEAL</td>
<td></td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SOCKET &amp; SOLVENT</td>
<td></td>
<td>✓</td>
<td>✓</td>
<td></td>
<td></td>
<td></td>
<td>✓</td>
</tr>
<tr>
<td>COMPRESSION</td>
<td></td>
<td>✓</td>
<td></td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td></td>
</tr>
<tr>
<td>FLARED</td>
<td></td>
<td>✓</td>
<td></td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td></td>
</tr>
<tr>
<td>FLANGED</td>
<td></td>
<td>✓</td>
<td></td>
<td></td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
</tbody>
</table>

*Note:* The symbols represent the compatibility of each joint type with different materials.
PIPES AND VALVES

14.14 PIPELINE SUPPORTS

Piping won't stand up to stresses and work efficiently without adequate support or suspension. The term suspension is used because often the piping should not be rigidly fixed down. For example, where pipe expansion takes place, rollers are used.

Figure 14.8 shows a variety of pipe hangers and supports. Generally, the supports should occur at no less than three metre spacing and on either side of a valve or heavy equipment.

Fig. 14.8 Pipe Hangers and Supports

14.3 VALVES

A wide variety of valves is available for use on installations conveying water, oil, petroleum, chemical and gases with differing capacities for withstanding high pressure, high temperature range and aggressive chemical attack.

In order to classify valves it is desirable to specify the use in terms of their actual function.

Shut-off valves enable a section of a pipeline to be isolated so that repairs may be effected or as a stop valve when pipe flow is not required.

Throttle or flow control valves may be used to isolate a section of pipework but their primary function is to allow metering of the flow at some value between fully open and fully closed.

A metering valve is designed to have reasonably linear flow characteristics - the percentage flow through the valve equates to the percentage the valve is open. Also, the valve is designed so that it is reasonably easy to operate under load.
A diverting valve has the ability to divert flow from one pipeline to another. In many cases it is also required to shut the flow off completely or even provide a degree of throttling.

Special purpose valves are used for functions such as:

- Preventing reverse flow - a check valve (or non-return valve) is opened by the fluid flow but reverse flow cannot occur.

- Pressure regulators reduce the supply pressure to the required service pressure.

- Relief valves are used to control maximum system pressure. When used on a steam boiler, the relief valve is called a safety valve and its setting and operation has to comply with legal requirements.

Control valves may be classified into gate, globe, plug, ball, butterfly, diaphragm and needle types.

A gate valve (Figure 14.9) is a vertical shutter or gate that rises or falls in the face of fluid flow. The gate may be tapered (wedge type) to form a seal or may be parallel with sprung sealing faces. A screw is used to raise or lower the gate, the screw being turned by a handwheel.

If the handwheel rises as the valve opens, it is called a rising-stem valve - non-rising stem versions are also available. Note that the controlled fluid is in contact with the rise and fall thread so a sealing system is required above the thread to prevent system fluid loss to atmosphere.

Gate valves are general purpose valves, ideally suited for shut off use but also sometimes incorrectly used for throttling duties. The valve is available in a wide range of metals and plastics with flanged or threaded ends.
A globe valve (Figure 14.10) is very similar to the type used in the domestic water supply, and is called a globe valve because of the globular shape of the valve body.

The valve consists of a disc which presses either:

(a) onto an orifice, the valve disc being faced with a material which deforms to produce a seal, or
(b) into an orifice, the valve disc being angled at about 45°, metallic and usually called a plug type disc.

By forming the plug at differing angles the throttling characteristics of the valve may be modified.

Globe valves are normally used where operation is frequent and/or primarily in throttling service to control the flow to any desired degree. The significant feature of this valve is efficient throttling with minimum wire drawing - a localised erosion across the valve seat which looks like the mark left by a wire frequently drawn across the seat.

Because the flow path through the valve is composed of two 90° turns, a globe valve causes a very high pressure drop, but the short disc travel means the valve may be opened and closed quickly.

A needle valve is basically a globe valve but the disc is replaced by a needle-like stem entering into an orifice. Needle valves are used for throttling applications but generally for light duty work. For this reason, the valves are only available up to 25 mm.

Plug valves (Figure 14.11) are the most ancient type of valves, early examples being made of wood. The advantages of a plug valve are:

- Minimum amount of installation space.
- Simple operation.
- Quick action - a 90° turn - so frequent operation is possible.
- Low pressure drop as long as the valve is fully open.
Plug valves are not normally suitable for throttling applications because, like a gate valve, a great percentage of flow change occurs near shut-off point and the resultant high velocity causes valve erosion. Another important characteristic of the plug valve is its easy adaptation to multi-port construction, with up to six outlets. By making the hole in the plug in an L-shape, various inter-connections of ports may be achieved.

One major disadvantage of a plug valve is its tendency to seize (become difficult or impossible to operate) if use is infrequent.
Pipework and Valves

A *ball valve* (Figure 14.12) is similar to a plug valve but the rotating valve section is spherical. This ball rotates between resilient seals which allow a 90° operation with low operating torque.

As with the plug valve, the ball valve should not be used for throttling applications because of erosion at the near-closed point.

![Ball Valve](image1)

Fig. 14.12 Ball Valve

A *butterfly valve* (Figure 14.13) is used where metering or throttling is required on large pipelines. The valve operating torque is low because the butterfly shape tends to be balanced in operation.

Butterfly valves are light (for a given size) and the 90° operation provides fast control of liquids and liquids with suspended solids. For a pressure-tight seal, the butterfly disc engages with a rubber seal set within the valve body.

![Butterfly Valve](image2)

Fig. 14.13 Butterfly Valve

A *diaphragm valve* (Figure 14.14) consists of a flexible diaphragm which is pushed onto a raised portion of the flow path, restricting flow by a pinching action. Diaphragm valves may be used for on/off service or for throttling and are quick opening. The valves are particularly suitable for handling corrosive materials or materials that must not be contaminated because there is no contact between the material and the valve operating mechanism.

Disadvantages of the valve are:
- At high operating pressures, the force required to depress the diaphragm is high so valve operation is hard.
- With the non-linear flow path, the valve has a relatively high pressure drop.
As a guide to the operating characteristics of the valves discussed, Table 14.1 displays general information on size range, operating pressure and temperature range. Note that the values given are a generalisation and depend of the materials used for the seals and valve body.

<table>
<thead>
<tr>
<th>Size</th>
<th>Operating Pressure</th>
<th>Operating Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Min (mm)</td>
<td>Max (mm)</td>
</tr>
<tr>
<td>Gate</td>
<td>1300</td>
<td>-100</td>
</tr>
<tr>
<td>Globe</td>
<td>800</td>
<td>-100</td>
</tr>
<tr>
<td>Needle</td>
<td>25</td>
<td>-100</td>
</tr>
<tr>
<td>Plug</td>
<td>600</td>
<td>0</td>
</tr>
<tr>
<td>Ball</td>
<td>900</td>
<td>0</td>
</tr>
<tr>
<td>Butterfly</td>
<td>900</td>
<td>-100</td>
</tr>
<tr>
<td>Diaphragm</td>
<td>600</td>
<td>-100</td>
</tr>
</tbody>
</table>

Table 14.1 Valve Characteristics

A selection of check valves (or non-return valves) are shown in Figure 14.15 - they are valves that only allow a flow in one direction.

The *swing check valve* is a flat plate pivoted within the valve body in such a way that it is forced open by the fluid flow as shown. The slight face angle ensures that the valve closes as flow stops.

The other valves shown have improved sealing but all cause a higher pressure drop than the swing check valve.
14.4 PIPEWORK DESIGN

As previously discussed in this Chapter, a wide range of pipes is available to convey a fluid from one point to another. When the situation arises where a piping system is to be designed, several factors must be considered:

**Pipe compatibility**  Obviously a steel pipe would not be used for a very corrosive chemical nor a plastic pipe for a high temperature fluid. Pipe manufacturers supply tables of the resistance of their pipe to differing chemicals and the allowable working conditions for the pipe.

**Pressure rating**  All pipes have a pressure rating and in the case of plastic pipes this is modified by the working temperature, as shown in Table 14.2 for unplasticised PVC pipe. The working pressure is stated for a given material/diameter/wall thickness. This value is a proportion of the pressure at which the pipe would burst, because a factor of safety is incorporated.
Cost

The purchase price of the pipe is not the only consideration. A pipe that is cheap to buy may require highly skilled labour to join together so the final installation cost is high. A cheap installation may not last very long under heavy usage so the long-term cost may be high if the system has to be frequently repaired or replaced. For instance, a plastic pipe system may be cheapest to install but if in an exposed position is constantly prone to damage which a steel pipe could withstand.

<table>
<thead>
<tr>
<th>Pipe Temp. (°C)</th>
<th>CLASS 4.5 (MPa)</th>
<th>CLASS 6 (MPa)</th>
<th>CLASS 9 (MPa)</th>
<th>CLASS 12 (MPa)</th>
<th>CLASS 15 (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0.45</td>
<td>0.60</td>
<td>0.90</td>
<td>1.20</td>
<td>1.50</td>
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<tr>
<td>30</td>
<td>0.36</td>
<td>0.48</td>
<td>0.72</td>
<td>0.96</td>
<td>1.20</td>
</tr>
<tr>
<td>40</td>
<td>0.27</td>
<td>0.36</td>
<td>0.54</td>
<td>0.72</td>
<td>0.90</td>
</tr>
<tr>
<td>50</td>
<td>0.18</td>
<td>0.24</td>
<td>0.36</td>
<td>0.48</td>
<td>0.60</td>
</tr>
<tr>
<td>60</td>
<td>0.09</td>
<td>0.12</td>
<td>0.18</td>
<td>0.24</td>
<td>0.30</td>
</tr>
</tbody>
</table>

Table 14.2 Maximum Allowable Working Pressures at Various Temperatures

Frictional loss

The pressure drop caused by the fluid flow having to overcome internal resistance within the pipe. Differing pipe materials cause differing amounts of frictional loss and this loss must be predictable so that the working pressure of a supply pump can be specified or so that the pressure available at the end of a pipeline is known.

The pressure drop in a pipe is dependent on the internal roughness of the pipe, the velocity of fluid flow and the fluid being conveyed. Many formulae exist for calculating pressure drop but for convenience the information may be displayed on a nomograph, as shown in Figure 14.16 for one type of black polyethylene pipe conveying water.

Fig. 14.16 Frictional Loss in Pipe

Reproduced by permission of Union Carbide Australia Limited
As an example of the chart use, 30 litres per minute of water flowing through a 25 mm outside diameter pipe would have a pressure drop of nine metres head for each 100 metre length of pipe.

As one metre head of water equals 9.8 kPa, the loss equates to 88.2 kPa per 100 metre length.

**EFFECT OF FITTINGS**

In a similar way to the pressure loss which occurs within pipelines, a pressure loss occurs across pipe fittings. The magnitude of the pressure loss depends on the fluid velocity (as with a pipe) and with the shape of the fitting.

Considering a tee fitting, fluid flow straight through the tee causes less pressure loss than if the fluid has to turn through 90° within the tee.

It is possible to calculate the head loss or pressure loss caused by a fitting but it is more convenient to express the effect of the fitting in terms of equivalent length. This means that the loss across a fitting is given as the equivalent length of straight pipe that would cause the same loss as the fitting. Then the equivalent length of the fitting can be added to the total pipe length to determine the overall pressure loss.

The equivalent length for fittings used on a particular range of pipe fittings may be obtained from manufacturers technical catalogues, or a more general indication may be determined from the nomograph shown in Figure 14.17.

It provides the equivalent length of steel piping for a range of fittings and includes the effect of sudden changes in pipe diameter.

The example shown indicates that a 50 mm half-open gate valve equates to the pressure loss in 10 metres of 50 mm pipe.
Fig. 14.17 Equivalent Length
COMPRESSORS

15.1 AIR COMPRESSORS

As shown in Figure 15.1 there are two basic groups of air compressors:

(a) Displacement - reciprocating and rotary.
(b) Dynamic - kinetic energy in the air is changed to pressure energy.

Fig. 15.1 Compressor Types

(a) Displacement Compressors (Figure 15.2)
These compressors operate on the displacement principle. Air contained in a chamber is compressed by reducing the volume in the chamber. Compressors in this group are of the reciprocating or rotary piston type construction.

Fig. 15.2 Displacement Compressor
(b) Dynamic Compressors (Figure 15.3)
These compressors operate on the air flow principle. Air is drawn into the chamber and compressed by means of mass acceleration. Compressors in this group are of the radial flow and axial flow types.

![Fig. 15.3 Dynamic Compressor](image)

SINGLE AND MULTI-STAGE COMPRESSORS

The reciprocating compressor operates on the same principle as the two-stroke cycle. As the piston moves down, air is drawn in through the suction valve; on the up stroke the piston compresses the air and discharges it through the delivery valve.

Single-acting compressors have only one compression chamber in each cylinder. Figure 15.4 illustrates the three stages of a single acting unit.

![Fig. 15.4 Single-Acting Compressor](image)

Double-acting compressors have two compression chambers in each cylinder, one on each side of the piston. The piston thus performs compression work both on the forward and on the return stroke.

The machines are equipped with a cross-head which takes up the lateral forces in the crank mechanism, as in Figure 15.5.
Multi-stage (common two-stage) compressors compress the air in the low pressure cylinder to a certain intermediate pressure. The air then passes to an intercooler before it is compressed in the high pressure cylinder to full working pressure.

Operating Pressures
The normal operating pressures for reciprocating piston compressors are:

- Single-stage - up to 900 kPa
- Double-stage - up to 1500 kPa
- Multi-stage - up to 22 MPa (such as for replenishing scuba bottles)

ROTARY COMPRESSORS
In rotary compressors, the compression takes place by back flow from the discharge side each time a rotor tip uncovers the discharge port. This compressor is restricted to low-stage pressure ratios. Usually they operate as single-stage machines, but two- and three-stage versions exist.

1. Roots Blower (Figure 15.6)
Blowers are used where large volumes of air are required at low pressures. The design of the blower is based upon two rotors which intermesh, but maintain their clearance by a pair of external timing gears. These compressors are normally air cooled.
2. Vane Compressor (Figure 15.7)
The rotary vane compressor is a single shaft compressor with a built-in pressure ratio. A rotor with radially moveable blades is mounted eccentrically in a stator housing. When it rotates the vanes are pressed against the stator walls by the centrifugal force. Air taken into the compressor enters the space between the vanes in their most eccentric position where the pocket between two vanes is largest. As the rotor turns, the pocket volume decreases and the air is compressed until the discharge port is uncovered by the leading vane of each pocket.
By injecting copious amounts of oil into the compression space it is possible to lubricate, seal and cool at the same time. The injected oil is reclaimed and recirculated after the compression. Both air-cooled and water-cooled versions exist.

3. Screw Compressor (Figure 15.8)
The screw compressor has a built-in pressure ratio. The counter-rotating screw rotor elements are synchronised by means of timing gears external to the compression space. As the rotors do not touch each other or the casing, lubrication is not required within the compression chamber but is often used to improve sealing and aid cooling.

The absence of inlet and outlet valves and of unbalanced mechanical forces enables the screw compressor to operate at high shaft speeds. Consequently, it combines high capacities with small outside dimensions.

Fig. 15.8 Screw Compressor
DYNAMIC COMPRESSORS

In the dynamic compressor, the pressure rise is obtained by drawing the gas into the centre of a rotating impeller, which then throws the gas to the periphery part of the chamber owing to the centrifugal force.

(a) Radial (Centrifugal) Compressor (Figure 15.9)

The stage pressure ratio is determined by the amount of velocity change and the density of the gas. Operating speeds are high compared to other compressors; 50 000 to 100 000 r/min are common in the aircraft and space industries where space is a major factor. Most commercial centrifugal units operate at around 20 000 r/min.

(b) Axial Compressors (Figure 15.10)

This compressor type is characterised by having its flow in the axial direction. The gas passes axially along the compressor through alternate rows of rotating and stationary blades which impart velocity and then pressure to the gas.

The minimum capacity of this type is around 15 m³/s.

Axial compressors, due to their smaller diameter, operate at higher speeds (usually 25% higher) than centrifugals for the same duty. The axial compressor is best suited for plants requiring large but constant quantities of air. A typical application is a blast furnace blower or in a glass works.
Fig. 15.10 Axial Compressor

Fig. 15.11 Compressor pressure/flow characteristics
Compressors

As a comparison between the capacities of differing types of compressors, Figure 15.11 shows the typical pressure/flow relationships. The chart illustrates that piston compressors are used for high pressure applications.

Rotary (vane) compressors are used for lower pressures but are popular for mobile applications because of their compact nature and ease of control. Extremely high flow rates are available from radial or axial type compressors.

These types of compressors are only economical when operated at or close to full rated output because flow control is effected by restricting the input which in turn lowers the machine efficiency (ratio of power in to power out).

15.2 HEAT OF COMPRESSION

When air is compressed it generates heat, which must be removed if the compressor is to operate efficiently. The type of cooling system selected for this operation depends upon the amount of heat generated by the compressor. The heat produced during compression has several undesirable effects:

1. The ability of air to carry water in vapour form is dramatically increased as the air temperature rises. For instance, for one cubic metre of saturated air at 500 kPa:

<table>
<thead>
<tr>
<th>AIR TEMPERATURE</th>
<th>GRAMS OF WATER</th>
</tr>
</thead>
<tbody>
<tr>
<td>85°C</td>
<td>90</td>
</tr>
<tr>
<td>50°C</td>
<td>20</td>
</tr>
<tr>
<td>20°C</td>
<td>4</td>
</tr>
<tr>
<td>0°C</td>
<td>1.3</td>
</tr>
</tbody>
</table>

   Thus, if the compressed air can be cooled it will precipitate the majority of water previously in suspension.

2. Heating of the compressor causes a reduction in the viscosity of the lubricating oil, with related extra wear and frictional increase. If the heating is excessive, the oil may carbonise and lose its lubricating qualities. The carbon produced lines the valves and blocks the compressor delivery ports.

3. As air temperature increases, its mass density decreases. Thus, for each stroke of the compressor the volume of air may be constant but the actual mass transferred varies with temperature so maximum compressor output occurs when the machine is operating at its lowest temperature.

4. The theoretical operating cycle of a piston compressor is shown in Figure 15.12. Due to the heat of compression the curve follows the law $pV^n = a$ constant, where $n$ is normally about 1.3. If no heat were produced during compression (very slow compression or with perfect cooling) then compression would be isothermal as discussed in Chapter 2 and follow the law $pV = a$ constant. The area of the $pV$ diagram represents work done:

   \[ \text{pressure (N/m}^2\text{) x volume (m}^3\text{)} = \text{work done (N m)} \]

   and as the area of the diagram enclosed by the isothermal curve is less than that enclosed by the $pV^n$ curve, then isothermal (or constant temperature) is obviously the ideal situation.
In order to dissipate the compression heat, it is usual to cool the compression cylinder, cylinder covers and related surfaces but most heat is retained within the compressed air. Small compressors are designed with cooling fins which allow the heat to be removed by radiation. Large compressors, however, are equipped with a fan to remove the heat.

When the drive power for a compressor exceeds 30 kW, air cooling is no longer efficient. Heat generated by the compressor is removed by water circulating through a water jacket. In some cases a cooling tower is used to reduce the temperature of the cooling water before it is recirculated in the air compressor.

Multi-stage compressors are equipped with intercoolers and aftercoolers.

Intercoolers reduce the temperature of the air between stages, while aftercoolers reduce the temperature of the air after the last stage, and before the air enters the receiver.

![Theoretical Cycle](image)

*Fig. 15.12  Theoretical Cycle*

*Air-cooled intercoolers* (Figure 15.13) may consist of a number of finned tubes through which the air under pressure passes, or may be in the form of a radiator. Cooling air is blown across the surface with a fan.

![Air-cooled Intercooler](image)

*Fig. 15.13  Air-cooled Intercooler*
A *water-cooled intercooler* (Figure 15.14) consists of a nest of tubes in the form of a heat exchanger. The nest is enclosed in a shell or housing. The compressed air in some intercoolers passes over the outside of the tubes through a series of baffles to ensure maximum contact with the cooling surfaces of the tubes. In others, the air passes inside the tubes and the coolant is outside.

Perfect intercooling is attained when the temperature of the air leaving the intercooler is equal to the temperature of the air at compressor intake.

Where cooling is applied to the machine the main purpose is to protect it from damage, although some improvement in efficiency is obtained.

Intercoolers are usually provided with a relief valve on the air side and drain cocks on the air side and the water side. The air side drain cock is essential to the removal of water which condenses in considerable quantity in the cooler.

![Water-cooled Intercooler](image)

Fig. 15.14 Water-cooled Intercooler

An *aftercooler* (Figure 15.15) is normally used to cool the air as it leaves the compressor. These are similar in construction to the intercoolers. Their function is to chill the air and remove as much condensate as possible before the air goes into the supply system.

Condensation of water in the pipes of driven appliances can have the undesirable effects of freezing at the exhaust of pneumatic tools, rusting in pressure vessels and washing away of lubricants. Many processes such as printing, glass making, spray painting, cleaning and so on require dry air.
With water-cooled stationary compressors it is considered desirable by some authorities that the aftercooler should have a separate supply of cooling water that is circulated through an outside cooling tower.

Fig. 15.15 Aftercooler
HYDRAULICS AND PNEUMATICS

16.1 PROPERTIES OF A FLUID

To understand the behaviour of a fluid we must study its characteristics, as compared with those of a solid. Note that fluid is a classification for a liquid and a gas - both are fluids and hence the name used for power transmission by a pressurised liquid or a compressed gas is Fluid Power.

A solid has a rigid molecular structure with molecules held in a fixed relationship with each other, tending to resist any change in physical shape. They are, generally speaking, incompressible.

In a gas the molecules move quickly and against each other, in the region of 500 metres/second. Unless confined, a gas will neither remain within definite boundaries nor have any specific shape. All molecules contain heat energy, characterised by some form of motion. If this heat is removed from a gas there will be a decrease in the movement of the molecules and in this state a gas may be changed to a fluid.

The movement of gas molecules provides the gas with its expansive quality and also allows it to be compressed. Also, the volume of a gas tends to change a lot with a small change in temperature. A liquid has characteristics similar to those of a solid, except that the shape of a liquid may readily be changed.

This is the reason why a liquid is readily usable as a flexible form of transmitting mechanical energy, hence the widespread use of oil as a fluid power medium.

The characteristics of a fluid are summarised in three laws formulated by Pascal in 1648:

1. A fluid at rest exerts pressure in all directions at right angles to any surface containing it.
2. The pressure at any point in a fluid is the same in all directions.
3. If the weight of the fluid is neglected, the pressure within a fluid is equal throughout.

These three laws are illustrated in Figure 16.1.

![Fig. 16.1 Pascal’s Laws](image)

Pascal's laws provide the principle of operation of a hydraulic lifting device, such as a car jack shown in Figure 16.2.
Rearranging the statement that force equals pressure multiplied by area to:

\[
\text{pressure} = \frac{\text{force}}{\text{area}}
\]

it may be calculated that the pressure within the fluid caused by pushing in the smaller piston is

\[
\text{pressure} = \frac{200 \text{ N}}{(100 \times 10^{-6}) \text{ m}^2} = 2 \times 10^6 \text{ N/m}^2 = 2 \text{ MPa}
\]

Because pressure is distributed equally throughout the fluid (according to Pascal) the same 2 MPa pressure is applied to the underside of the larger piston.

Now the force output of the larger piston may be calculated from the original statement:

\[
\text{force} = \text{pressure} \times \text{area}
\]

\[
\text{force output} = (2 \times 10^6) \text{ N/m}^2 \times (1000 \times 10^{-6}) \text{ m}^2
\]

\[
= 2000 \text{ N}
\]

Incidentally, one easy way to remember the force/pressure/area relationship is from a triangle:

where force = pressure \times area

\[
\text{pressure} = \frac{\text{force}}{\text{area}}
\]

\[
\text{area} = \frac{\text{force}}{\text{pressure}}
\]

and the F and P segments can be recalled by thinking of the topic - Fluid Power.

Two examples are included to illustrate further uses of this expression.
EXAMPLE 1

In the system shown in Figure 16.3, calculate the pressure below the piston at \( p \). The piston is 90 mm diameter and the piston rod is 30 mm diameter.

![Fig. 16.3]

The pressure at \( p \) is composed of:

(i) The pressure owing to the 1 tonne pulling down on the piston.
(ii) The pressure caused by the 6 MPa acting on top of the piston.

Considering (i):

\[
\text{pressure} = \frac{\text{force}}{\text{area}} = \frac{(1 \times 10^3 \times 9.8)}{\left(\frac{\pi}{4} \times 0.09^2\right) - \left(\frac{\pi}{4} \times 0.03^2\right)} \\
= 1.73 \text{ MPa}
\]

Considering (ii):

The pressure below the piston caused by the 6 MPa acting on top of the piston is higher than 6 MPa, due to the difference between the top and bottom areas, providing an intensification effect. We could calculate the force on the piston caused by the 6 MPa, and then calculate the pressure on the underside of the piston caused by that force. A quicker method is to realise that the pressure intensification caused is proportional to the area ratio of top to bottom.

Thus, pressure below the piston caused by 6 MPa on top of the piston

\[
= 6 \times \frac{\pi}{4} \times 0.09^2 - \frac{\pi}{4} \times (0.09^2 - 0.03^2) \\
= 6.75 \text{ MPa}
\]

So, total pressure = 1.73 + 6.75 = 8.48 MPa

This example of obtaining a pressure higher than the original supply pressure is a common cause of problems in oil hydraulics, where excess pressure can damage seals or burst hoses.
EXAMPLE 2

In the system shown in Figure 16.4, calculate the force available from the 70 mm diameter piston.

![Diagram of a hydraulic system with a 70 mm diameter piston, 6 MPa applied pressure, and a 30 m column of oil.]

Pressure applied to the piston is composed of the 6 MPa applied pressure plus the pressure owing to the head or column of oil. The effect of the head was considered in the third of Pascal’s Laws.

Pressure due to a head of fluid = \( \rho gh \)

So head pressure = 870 kg/m\(^3\) x 9.8 m/s\(^2\) x 30 m = 0.256 MN/m\(^2\)

Pressure applied to the piston = 6 + 0.256 = 6.256 MN/m\(^2\)

Piston force = pressure x area = (6.256 x 10\(^6\)) N/m\(^2\) x (\(\pi/4\) x 0.07\(^2\)) m\(^2\) = 24.08 kN

16.2 HYDRAULICS OR PNEUMATICS?

Fluid systems use fluid under pressure to transmit energy. Fluid power includes the generation, control and application of smooth effective power by means of pumped or compressed fluids. The input is mechanical power and the output is linear or rotary work.

With fluid power:

1. Very high quantities of power can be transferred (hundreds of kilowatts) by a system that is physically smaller than comparable mechanical or electrical systems.

2. Easy, quick speed adjustment is possible by adjusting flow rates.

3. Complete control over output torque (of a motor) or thrust (of a cylinder) is possible by controlling the applied pressure.
4. A built-in resilience cushions shock loading, especially when air is the fluid used.

5. Rapid reversing is possible.

6. Smooth and controllable acceleration and deceleration may be easily achieved.

7. Because the components used are discrete, self-contained items connected by tubing, a system may be modified or enlarged easily without expensive mechanical alterations.

8. With power transmitted by a fluid, the use of electricity in an explosion-prone area is avoided.

9. If no mechanical linkage is used, a fluid power system cannot suffer from loose linkages or backlash.

10. Fluid power components can be located where required and are not restricted to the confines of a mechanically driven system, with positive drives via chain, shaft or pulleys. The fluid power output can be remotely controlled electrically, as for the wing flap-settings on an aircraft.

11. Fluid power output stages will stall at maximum setting, without damage.

12. Relatively small input forces may be used to control large output forces.

Hydraulics, derived from the Greek *hydro* meaning water and *aulis* meaning tube or pipe, was originally based on the use of water under pressure, but for fifty years has operated mainly with oil as the fluid medium.

Pneumatics is derived from the Greek work *pneumos* meaning air or breath.

As a general rule, in choosing a suitable fluid power system:

- If a large amount of force is required as an output, use hydraulics.
- If high speed and rapid response is required, use pneumatics.
- If fine feeds are required and output force is not high, a combination of pneumatics and hydraulics is used.

Pneumatic systems are used where high ambient temperatures exist, such as near a furnace. In explosive areas, pneumatics is again normally favoured because of the absence of fire risk. Pneumatics perform well in very adverse conditions, such as under water, or in a dusty atmosphere (coal-mining, flour milling) or in low temperatures where hydraulic systems would become sluggish.

Hydraulic systems may be used in adverse environments too but may need additional equipment. For example, in an explosive atmosphere the electric drive of an hydraulic system would have to be spark protected. Also, fire-resistant hydraulic fluid would probably be specified.

At low temperatures, some hydraulic systems have an inbuilt heater to maintain the oil at a usable viscosity. In very dirty atmospheres, provisions are made to keep hydraulic equipment as dirt-free as possible and it is common to pressurise the oil reservoir to prevent the ingress of contaminates.
Whereas with pneumatic equipment the used air is vented to atmosphere, in hydraulics the oil is re-circulated and great care must be exercised to keep the oil clean.

In terms of power-for-weight ratio, hydraulics is easily the better because the oil pressure used (15 - 20 MPa) is up to 20 times higher than typical pneumatic pressures (600 - 800 kPa) and thus the output actuator size of an hydraulic unit is smaller for a given force output.

The capital cost of hydraulic equipment is much higher than comparable pneumatic components but again, because of the greater hydraulic output, the relative cost per kilowatt is comparable.

**16.3 HYDRAULIC COMPONENTS**

You have seen a circuit diagram for a vehicle's electrical system and should have realised that for simplicity, symbols are used to represent components in a system. A similar system exists for hydraulics, where it would be pointless to show a picture of a valve when a symbol could be used. The symbol not only shows what the component is but also its basis of operation. The symbol does not, of course, explain the mechanical construction of a component.

Around the world several symbol systems exist but they are all compatible and the Australian version is given in AS 1101.1-1993. The symbols, which show connections, flow paths and the function of the components are composed basically of circles, squares and lines. A selection of the main symbols from AS 1101.1 is shown in Figure 16.5. From the basic shapes the various individual components are formulated.

---

**Fig. 16.5 Pneumatic and Hydraulic Symbols**

For example, Figure 16.6 illustrates two types of pump.

The first is an energy conversion unit (a circle) with hydraulic flow (black arrow-head pointing out).
The second is a variable-displacement hydraulic motor - the black arrow-head which points inwards denoting that the unit is supplied with oil (a motor) rather than delivering oil (a pump).

![Hydraulic Pump Diagram](image1)

Fig. 16.6 Hydraulic Pump

A pictorial view of a simple hydraulic system is shown in Figure 16.7.

![Hydraulic System Diagram](image2)

Fig. 16.7 Hydraulic System

Figure 16.7 shows a hydraulic cylinder (A) that is operated forwards and backwards by oil under pressure. The direction of oil flow is controlled by the hand-operated valve (B), oil being delivered from the pump (E) and returned to the oil reservoir (G) via a filter element (F).

This type of diagram is useful for assembling the components or perhaps for identifying a particular component, but it does not give any indication of how the system works. Also, this type of diagram is very time-consuming to draw!
Figure 16.8 shows the same system in symbol form. Now the precise function of each component can be identified and, if necessary, the circuit traced through from source (pump) back to the oil reservoir via the valve and cylinder. In fact Figure 16.8 is called a *circuit diagram*.

Considering each component (items A-G) of Figure 16.8.

Item A (Figure 16.9) shows a *double-acting cylinder*, which converts hydraulic flow and pressure to mechanical velocity and force. Double-acting means that travel both ways (in and out) is achieved by using oil flow.

![Fig. 16.8 Hydraulic System in Symbol Form](image1)

In hydraulic systems, widespread use is also made of *single-acting cylinders*, where the piston is extended by oil flow but is retracted by an external load pushing the piston back.

![Fig. 16.9 Double-Acting Cylinder](image2)

Item B shown in Figure 16.10 is a three position, directional control valve.

It is operated by a lever which moves the valve to either of its extreme positions (left or right) and when the lever is released the valve springs back to the central (or neutral) position.
The directional control valve is a good example of the advantages of working in symbols, the three discrete positions of the valve may be shown. Thus, in centre position, the oil pressure is at a minimum (minimum resistance to flow) and the energy consumption is at a minimum. Furthermore, the avoidance of wasted energy means the oil temperature stays lower.

With the operating lever released (Figure 16.11) oil flows through the valve and back to the oil reservoir.

This is called an open-centre system because in the neutral position the valve is open for free return flow.

In the mid position the cylinder lines are blocked, thus effectively locking the cylinder in position.

When the operating lever is moved so that the valve changes to the left-hand square (Figure 16.12), oil flow through the valve causes the cylinder to extend.

Oil from the other end of the cylinder flows back through the valve to the oil reservoir.

Note that the valve must be held in this position; if the lever is released the valve will spring back to the neutral position.

When the valve-operating lever is moved the other way so that the right-hand square is connected (Figure 16.13), oil flow reverses through the valve and the cylinder retracts.

Item C (Figure 16.14) is a flow control valve and its function is to control the flow of oil from the cylinder which, in turn, controls the speed at which the cylinder extends.
The valve is composed of three main parts, as shown in symbol form in Figure 16.15.

The flow control is a variable orifice that places a restriction in the oil flow. It works by causing a large pressure drop in the flow and the back pressure created causes excess oil to dump via the relief valve (item D) and only enough oil is available at the cylinder to provide the required speed.

The pressure compensation portion of the valve is included to modulate the valve as the system pressure varies such that the valve flow is constant. System pressure depends on the loading placed on the cylinder and as this load/pressure varies, the oil flow through the orifice will also tend to vary. Pressure compensation is achieved by another orifice in the valve that adjusts to suit the system pressure.

The non-return valve (or check valve) is included so that the speed control is only obtained in one direction of cylinder travel. As shown in Figure 16.8, the oil is being metered out of the cylinder to control its outward stroke or extension, and thus this method is called meter-out speed control.

Item D (Figure 16.16) is a relief valve and its function is to control maximum oil pressure in the system.

The relief valve is the most essential component in the system because it acts as a safety device, allowing oil to flow back to the oil reservoir if the pressure is too high.

The symbol illustrates that the relief valve is normally closed but when the supply pressure is high enough to open the valve against the adjustable spring, oil flows through the valve. (Figure 16.17)
Every hydraulic system must have a relief valve of some type (including, incidentally, the lubrication system of a car) and on many hydraulic systems more than one relief valve is fitted.

If a relief valve is not used, or if the pressure setting of the valve is too high, either the pump drive will stall, the pump drive coupling will break, the pump will break or an oil line will burst - in any event an undesirable result.

Item E (Figure 16.18) is an oil pump, driven in this case by an electric motor. Note that the pump symbol only denotes a device that delivers oil (outward pointing arrow head) and does not give any indication as to the type of pump or its principle of operation.

The filter F (Figure 16.19) is almost as important as a relief valve. When the hydraulic system operates, small particles of metal enter the oil (owing to wear) and other contaminates occur (owing to the oil deteriorating with heat). The filter removes these items from the circulating oil and keeps system wear to a minimum.

The filter traps particles larger than about 40 micrometres and contains an inbuilt relief valve that bypasses the oil if the filter element becomes clogged. It is normally placed in the line which returns oil to the reservoir, which means that the reservoir oil is clean and thus the pump is supplied with clean oil.

Item G (Figure 16.20) is the oil storage tank or reservoir. Its function is not only to store oil but to provide a place for the oil to cool down, a place for unwanted materials (sludges, water) to settle and for air to separate from the oil.

Note that in symbol form the reservoir is shown several times, but this is done to avoid drawing lots of lines back to one point. There is only one reservoir (Figure 16.21).
SYSTEM OPERATION

As previously shown in the complete hydraulic circuit in Figure 16.8, oil flows from the pump to valve B and returns to the reservoir G. Meanwhile the cylinder is locked in position because the two lines from the cylinder to valve B end at a blocked port.

When valve B is operated one way, oil flows to the back of the cylinder and causes the cylinder to extend - that is, the piston rod moves out.

As the piston moves out (Figure 16.22) oil on the right-hand side is pushed out and has to travel through flow control valve C to return to the reservoir.

The setting of valve C controls the speed of the cylinder extending.

At the end of the stroke, with the cylinder fully extended, or if the piston is stopped by a heavy load, the system pressure immediately increases because the pump is a positive displacement device trying to push oil into a stationary system.

The higher pressure opens the relief valve and oil is discharged back to the reservoir.

If control valve B is operated the other way the system reverses (Figure 16.23).

Oil flows through the non-return port of flow control valve C and makes the cylinder retract - that is, the piston rod moves in.

During this operation the cylinder speed is not controlled by valve C.

Note that to make the cylinder move either way the control valve B must be held in position - if it is released it will spring back to the middle (or neutral) position.
16.4 HYDRAULIC APPLICATIONS

CLOSED LOOP SYSTEM

A closed loop system is one in which the oil flows continuously around the system rather than returning to a reservoir after use, as in the open loop system in Part 16.3 of this text. A basic closed loop system is shown in Figure 16.24.

![Fig. 16.24 Closed Loop Principle](image)

Oil flows from the pump A to drive the hydraulic motor B. The arrow on the pump symbol shows it has variable displacement which means that its output can be varied, and the black arrowheads each way mean that its output can be reversed.

Thus, by varying the pump A, the motor B can be varied, not only for speed but also in both directions of rotation.

The important point is that the oil circulates around the system and never reaches the open air - it is a closed loop system.

While this principle is very good, in practice, additional equipment is required to overcome certain problems. A more complete closed loop system is shown in Figure 16.25.

First you will notice that the symbols for the pump A and hydraulic motor B have been changed slightly by the addition of a dotted line leading to a reservoir symbol. These dotted lines represent a DRAIN from the pump (and motor).

You will realise that on any piece of equipment involving oil under pressure it is virtually impossible to prevent leakage occurring (especially as the equipment starts to wear).

![Fig. 16.25 Closed Loop System](image)

In fact a slight leakage through the pump is desirable because if the adjustable pump were set to half-way - that is, no output - then unless leakage flow occurred, the pump would not receive any lubrication or cooling.
Thus, if leakage is occurring it has to be made up (or replenished) from a reservoir and this is the function of the pump at C. This pump is called a BOOST or CHARGE pump and its job is to top-up the system with oil.

To allow oil to enter the system, when required, two non-return valves E are included. To relieve excess pressure from the boost pump line a relief valve D is included. Although not shown in the circuit diagram, a filter would also be part of the system.

One further problem may occur in a closed loop system. Dangerously high pressures could occur in the system if:

- The hydraulic motor is overloaded.
- The pump is suddenly switched to reverse flow - the hydraulic motor has to stop before it reverses and actually turns into a pump while it is slowing down.

For these reasons, additional equipment is included to relieve excess pressures in the system, such as the cross line reliefs shown in Figure 16.26.

![Fig. 16.26 Cross Line Relief Valves](image)

**HYDRAULIC PRESS**

The circuit shown in Figure 16.27 indicates the principle of an hydraulically operated press.

With hydraulic equipment the maximum speed is normally limited by the maximum oil flow available from the pump.

To operate a large press, the pump flow would need to be enormous to achieve an economic speed so the circuit shown uses the jack rams to lower the press when valve B is operated.

As the main ram extends, it draws oil in from the reservoir through valve J. When the press is down the system pressure rises and valve D allows oil from the pump to act on the main ram and the press stroke is completed with the full force available.

Valve G is a relief valve that may be adjusted by the press operator to control maximum pressing force. Valve C is a counterbalance valve that holds the press up if the pump is switched off and also prevents the press from falling at too high a speed because of its own weight.
Note that in this circuit the pump is variable displacement and also pressure compensated, which means it automatically reduces flow as system pressure approaches a preset value.

CASCADE SYSTEM

The circuit shown in Figure 16.28 is of a machine which contains a hydraulically-operated clamp, ejection cylinder and a marking (stamping) cylinder. The clamp cylinder is a conventional unit with control by the manual valve.

The ejection cylinder extends at high speed with the valve setting shown because both ends of the cylinder are coupled together. The larger area on one end ensures that the cylinder extends. The oil flow from the piston rod end joins with the pump flow to provide rapid cylinder extension.

The other valve positions provide conventional cylinder control. As the system pressure rises valve A opens which provides oil to the marking cylinder.

Valve A sequences the system by ensuring that the clamp is applied before the marking cylinder may be operated. Valve B controls maximum marking cylinder pressure. A dual pump is used in this system with an unloader valve.

Under low pressure conditions both pumps send oil into the system, but above a certain preset pressure the unloader valve opens and by-pass oil flows from the larger pump back to the oil reservoir.
16.5 PNEUMATIC COMPONENTS

In Part 16.3 the need for symbols for hydraulic components was discussed; the same system of symbols is used for pneumatic items (Australian Standard 1101 Part 1).

Figure 16.29 shows a pneumatic system in which, once valve E is initiated, the cylinder oscillates backwards and forwards until valve E is closed.

This sort of diagram is very useful for identifying particular components, especially if the diagram is exactly as the components are laid out on the machine. It would also be useful for connecting the various components together with pipework.

However, its main failing is that it cannot be used to find out how the system operates and cannot be used to trace problems, because it does not show the way that each valve operates. Also, it takes a long time to draw and only matches one particular make of equipment.

Fig. 16.28 Machine Hydraulic Circuit
Chapter Sixteen

16.17

Figure 16.30 shows the same system but in SYMBOL form, drawn to Australian Standard 1101 Part 1.

Considering the system components:

Part A is a double-acting cushioned cylinder (Figure 16.31). Cushioned means that internal air is trapped to slow down the piston at the end of its stroke. The arrow across the cushion symbol denotes it is adjustable.
Part B (Figure 16.32) is a two-position valve which has five ports or external connections. This type of valve is often called a 2/5-way valve. The valve directs air to or from the cylinder and its two squares show the two differing flow conditions. It is changed over by an air pilot signal which is just an air pressure acting on a small piston on the end of the valve.

![Fig. 16.32 Two-position Valve](image)

Valves C and D (Figure 16.33) are identical but in symbol form they look different because valve C is initially operated and valve D is not operated.

![Fig. 16.33 Roller-operated Valves](image)

The function of these valves is to detect the position of the cylinder and pass a pilot signal on to the next part of the system at the appropriate time. The valves are 2/3-way, operated by a cam roller on top and returned by a spring.

Valve E (Figure 16.34) is the same type of valve as C and D but is operated by a hand lever. Its function is to stop and start the system operating.

![Fig. 16.34 Lever-operated Valve](image)

Filter F (Figure 16.35) traps water particles, dirt and other foreign matter which could cause corrosion or damage to the moving parts in the system.

The filter is always situated just before the point where the air is going to be used and traps particles bigger than 40 micrometres.
Pressure regulator G (Figure 16.36) reduces the supply pressure to a suitable value to operate the system.

If the system pressure is unregulated:

- Output force from the cylinder would be too high, causing component damage.
- Varying pressure causes varying output forces which means that the machine output will not be consistent.
- The lower the air pressure used, the lower the machine running cost.

Lubricator H (Figure 16.37) supplies a carefully metered quantity of oil which it mixes with the air in a fine mist to provide lubrication of dynamic components. It is a proportional device, increasing the oil input as the air flow increases.

SYSTEM OPERATION

As previously shown in Figure 16.30, the system is at rest with air passing through valve B to hold the piston in the retracted position. Air also passes through valve C to hand control valve E where it waits for valve E to be changed over.
When valve E is changed over (the system is switched on) air goes to the right-hand end of valve B, causing it to change over and make the piston extend.

When the piston is fully extended, it hits valve D and the waiting air travels to the left-hand end of valve B, resetting it to its original position and thus returning the cylinder to its original position.

If valve E is still open then the system will repeat its operation. It is not hard to visualise that any number of cylinders may be operated in any desired sequence by the use of suitable control valves.

### 16.6 PNEUMATIC APPLICATIONS

Some examples of pneumatic systems are included but it should be realised that the applications of pneumatics are practically limitless - it is a system used in any instance where a component needs to be pushed, lifted, pulled, rotated, squeezed, ejected etc.

### PACKAGING MACHINE

The circuit shown in Figure 16.38 uses a double-acting cylinder controlled by valves 10 and 11.

![Fig. 16.38 Packaging Machine](image)

Additional specifications for the machine include:

(a) During setting-up procedures, the cylinder must be capable of being inched in both directions.
(b) During normal running, the cylinder must not operate unless the machine guard is in position.
Valve 1 has two positions: 'Set' and 'Run'. In the 'Set' position, as shown, the system may be inched by operating either valve 2 or valve 3 which in turn changes over valve 4 to control the cylinder.

Note that valve 4 has three positions, returning to the mid position automatically if no change-over signal is present on the ends. When valve 1 is set to the 'Run' condition the system operates automatically, as long as the guard is kept down.

**CONVEYOR SYSTEM**

As shown in Figure 16.39, the cylinder oscillates and operates a ratchet system to advance a conveyor. This method is used where speed control is required or where a step-type movement of the conveyor is desirable.

When valve 2 is operated the cylinder starts to move backwards and forwards automatically, valve 1 giving a signal to retract the cylinder and valve 5 giving a signal to valve 4 to extend the cylinder.

Valve 8 is used to detect whether the conveyor becomes blocked and will stop the conveyor. Also valve 6 may be used to stop the conveyor in an emergency.

If either valve 8 or valve 6 are operated the system must be reset by valve 9 - an additional safety feature to ensure the conveyor does not start automatically after an emergency stop.

![Fig. 16.39 Conveyor System](image-url)
AUTOMATIC CLAMP AND DRILL

The circuit shown in Figure 16.40 is the basis of a popular form of pneumatic automation - the clamping of a component and the drilling of one or more holes. In this particular application an air-motor driven drill is used with built-in air feed and a built-in stroke sensing valve.

In operation, a component is placed in the vice or drilling fixture and valves 1 and 2 are depressed. The use of two hand-operated valves provides a degree of safety by encouraging the operator to use two hands to start the sequence.

When valves 1 and 2 are depressed the clamp cylinder is applied, the drill starts and feeds into the component. When the correct drill depth is reached a signal from the AFD (Air Feed Drill) reverses the drill (valve 5) and opens the clamp (via valve 3).

Fig. 16.40  Automatic Clamp and Drill
SEALS, PACKINGS AND FILTERS

17.1 PRINCIPLE OF SEALING

When a fluid at a higher pressure than the surrounding regions needs to be contained, some form of seal is required. Basically, a seal operates by:

(a) deforming to suit the gap to be sealed.
(b) providing a highly resistive path to fluid that tries to escape. A minimal clearance between the seal and the adjacent metal surface means that fluid leakage is greatly retarded owing to frictional pressure drop.

The concept of a minimum seal clearance providing minimum fluid leakage is acceptable for a static sealing situation, such as a cylinder-head on a car engine, but if the adjoining parts are rotating or sliding, minimum clearance may seal but can cause high frictional losses.

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High friction not only wastes power but also the heat generated can damage the seal. Thus, for moving (or dynamic) sealing applications a compromise must be obtained between a good seal and low frictional losses. The search for low frictional loss/long life sealing compounds has led to a bewildering array of seal materials being available, especially in the range of plastic materials, and has caused an extensive range of seal shapes to be evolved.
Some seal materials/shapes suit high temperature applications, some suit high pressure and others may have the capacity to work even after being left un-pressurised for weeks. Some seals perform very well but require skilled labour to install. Other types tolerate a high degree of seal misalignment but are expensive.

17.2 SEAL TYPES

Seals and packings may be classified into groups, as shown in Figure 17.1. Note that the classifications shown may be further sub-divided into additional types, especially in terms of materials used.

17.3 STATIC SEALS

A gasket is any device that maintains a barrier against the transfer of fluids across mating surfaces of a mechanical assembly when the surfaces do not move relative to each other.

The prime factor in any gasket application is the minimum sealing stress. This is the minimum stress (or gasket compression force divided by gasket area) necessary to make the gasket material conform to the imperfections of the sealing face, as shown in Figure 17.2. Force is also required to ensure that the porous structure of the gasket material is closed.

![Gasket Operation](image)

Common types of gasketed joints are shown in Figure 17.3.

**TYPICAL GASKET MATERIALS**

**Paper**
Simple low-cost joint for dust seals etc. when used in conjunction with a jointing compound. Impregnated papers are used for petrol and oil applications.

**Cork**
Suitable for low loads, such as thin metal flanges with poor surface finish. A common application is for an automotive rocker-box cover or oil sumps.

**Cork/Rubber**
Good general purpose material for low to medium pressures. Will withstand a variety of fluids.

**Rubber**
Excellent sealing material. When reinforced may be used for high pressure. A rubber gasket has the advantage that it may be distorted if required to assist assembly.
Plastics
Some plastics such as PTFE are almost inert to any chemicals and are therefore frequently used as a protective skin for more resilient gasket materials. Other plastics such as nylon are very tough and durable and are suitable for abrasive conditions.

Non-Metallic Types
A wide variety of metals and fillers are available, such as fibre and millboard - contained in a thin metal skin of stainless steel, copper or aluminium. Typical applications include cylinder-head gaskets for petrol and diesel engines and exhaust manifolds.

A flexible static seal is a type that has small amount of preload but deforms to provide a seal as pressure is applied. Some examples are shown in Figure 17.4.

In each case minimal sealing exists until a pressure differential occurs across the flexible seal and causes seal deformation.

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Seals, Packings and Filters

Sealants have gained popularity in recent years for static applications owing to the availability of chemically stable materials such as silicone. Often a semi-liquid sealant may be used where previously a gasket was utilised.

Actually, use of a sealant is similar to a gasket but the sealant is applied as a liquid or paste, as shown in Figure 17.5.

![Fig. 17.5 Sealant Application](image)

Sealants may be hardening or non-hardening types. Non-hardening types are popular because of their ability to deform under load and then return to their original shape.

Sealants generally have the ability to adhere to a surface to which they are originally applied, and may be applied directly to a surface or to an intermediary such as paper or metal, shown in Figure 17.6.

Diagram 17.6 (a) is a pure sealant gasket.

Diagram 17.6 (b) is a sealant bead applied to a paper carrier.

Diagram 17.6 (c) is an example of the sealant applied to either side of a metal former.
17.4 RECIPROCATING SEALS

An *exclusion device* is used to prevent the entry of foreign material into the moving parts of machinery. When a shaft is reciprocating, such as a cylinder rod moving in and out of a piston, external dirt must be prevented from entering the cylinder.

One method of exclusion, shown in Figure 17.7, is to fit a flexible bellows over the shaft such that the bellows will flex to the full shaft extension.

Fig. 17.6 Sealant Formats
Unfortunately, a flexible bellows is prone to damage that will allow the entry of dirt. More positive exclusion is provided by a *wiper seal*, shown in Figure 17.8.

Many variations of this seal exist, often called a *scraper seal*. The major consideration is that the lips of the seal keep in contact with the shaft. The lip is normally short and sturdy to overcome the resistance to foreign matter. At the same time, the lip must be flexible enough to allow for slight sideways travel of the shaft in service.

*Compression Packings* create a seal by being squeezed between the reciprocating shaft and the seal retainer.

As shown in Figure 17.9, the seal recess is called the *Stuffing Box* and the seal retainer and loading device is the *Gland*. 

---

**Fig. 17.7 Flexible Bellows Seal**

**Fig. 17.8 Wiper Seal**

**Fig. 17.9 Stuffing Box**
The packing material has to be a combination of a stable compound and also contain lubricant.

Examples of packing composition are shown in Figure 17.10. The packing is composed of materials such as cotton or jute, and impregnated with a lubricant such as graphite or molybdenum disulphide.

Compression packings have always been a very popular form of sealing because installation may be carried out without dismantling the complete unit. Also, as wear occurs, the packing may be further compressed to regain the original sealing conditions.

Compression packings are liable to be incorrectly used however, especially in terms of excessive tightening of the gland in an effort to reduce seal leakage. The excess loading causes high frictional losses between the reciprocating shaft and the seal material, causing heat, wear and possible loss of lubrication and shaft damage.

These packings are generally associated with slow-moving installations but they are used with rubbing speeds of up to 20 metres per second (400 r/min on a 100 mm diameter shaft). Some packings may be used at temperatures in excess of 550°C.

Moulded Seals in many cases perform a similar function to compression packings but are moulded to suit the shaft or bore diameter and have a cross-section that provides a flexible sealing lip.

The most common moulded seal is an O-ring - a circular seal with a round cross section.

Some other examples of moulded seals are shown in Figure 17.11.

Diaphragm Seals consist of a flat disc of flexible material, usually a fabric-reinforced rubber. The diaphragm may be cut from a flat sheet or moulded in the required shape. Diaphragm seals are used in such applications as a diaphragm valve, discussed in Chapter 14.
**Seals, Packings and Filters**

**Fig. 17.11 Moulded Seals**

*Reproduced by permission of Hallite Australia Pty Ltd*
17.5 ROTARY SEALS

A rotary seal is used to control fluid leakage past a rotating shaft. Many rotary seals are similar to seals used for reciprocating systems, such as compression packing or a moulded seal.

A *Labyrinth Seal* is used on a shaft that rotates at high speed and is subjected to high temperatures, such as in a steam or gas turbine.

A series of annular grooves, as shown in Figure 17.12, present a high resistance path to the contained fluid.

![Labyrinth Seal](image)

**Fig. 17.12  Labyrinth Seal**

*Mechanical Seals* are gradually replacing compression seals on installations such as centrifugal water pumps. The principle of a mechanical seal is shown in Figure 17.13.

Initially, the spring keeps the seal face and seal seat in contact, but as fluid pressure is applied the seal is maintained. The sealing faces are of dissimilar materials - often materials such as ceramic or carbon are used.

![Mechanical Seal](image)

**Fig. 17.13  Mechanical Seal**

*Reproduced by permission of Penton Publishing Co.*

*Oil Seal* is the usual name given to a radial lip seal, as shown in Figure 17.14.

One complete seal unit is an interference fit within the component. Usually, a light garter spring is included to maintain loading on the lip of the seal.
Note that this type of seal is not designed to retain fluid under high pressure. Also, the seal relies on the contained fluid to provide lubrication for the seal lip. Running this type of seal dry can result in rapid failure.

![Radial Lip Seal](image)

**Fig. 17.14  Radial Lip Seal**

### 17.6 PROPERTIES OF SEAL MATERIALS

Rubber seals are designed to be dimensionally larger (at least in cross-section) than their associated metal housings, and they are thus subjected to some mechanical deformation on assembly, giving rise to various stresses in the material.

In addition to the assembly stresses it is likely that the seal will be subjected to aggressive fluids, pressure, high and low temperatures, and dynamic motions and vibrations, all of which will combine to increase the original stress level.

Therefore in the interests of long-term economy it is vital that, when selecting a seal material, the optimum combination of physical properties is carefully considered and balanced against the service requirements.

The properties that combine to produce a good seal material from an engineering point of view are generally recognised to be:

- fluid resistance
- abrasion and extrusion resistance
- temperature range
- dynamic recovery.

The relative importance of these properties will of course vary, depending on the application of the seal.

### FLUID RESISTANCE

The majority of seals manufactured today are for use in hydraulic systems where it is necessary to resist the effects of mineral or petroleum-based oils and greases.

The degree of resistance is normally gauged by measuring the change in volume of a rubber specimen after soaking at an elevated temperature in the particular oil for a specified period. Changes in mechanical properties such as hardness and tensile strength are also used as a general guide.

Most rubbers, when in contact with hydrocarbon oils, exhibit a positive change in volume and swell to a greater or lesser degree. If the fluid is volatile, the process can be seen to be reversible. For instance, seals that have been immersed in fuel and become swollen are seen to return to their original dimensions when left
to dry out. However, it is necessary to be aware that the strength of rubber can be considerably reduced by an appreciable increase in swell.

It is normally recommended by seal manufacturers that for dynamic conditions, swell should be limited to approximately 10 per cent by volume.

**ABRASION AND EXTRUSION RESISTANCE**

Abrasion and extrusion resistances are very important in the choice of a seal material especially for dynamic situations and although a seal may possess the necessary fluid and temperature characteristics, if it does not have strength or toughness its service life will be unacceptable.

A good indication of the degree of toughness a material possesses can be obtained from its tensile strength. The tensile strengths of most general seal materials lie within the range 5-50 MPa.

Silicone materials have low tensile strength and exhibit comparatively low levels of strength and abrasion resistance, and thus rapid wear takes place in applications with high relative motion, or low lubrication, or both. On the other hand, polyurethanes, which have high values of tensile strength, are capable of operating under arduous conditions with rough surfaces and a degree of contamination.

Tensile strength together with hardness also influences the extrusion resistance of a material. It should be noted that tensile strength decreases as the temperature increases, and care is therefore required in the selection of a compound for a particular application. For instance, Viton can be considered to be roughly twice as strong as silicone rubber at room temperature but at temperatures in excess of 150°C it is weaker.

**TEMPERATURE RANGE**

Extremes of temperature cause dramatic changes in the properties of rubber. Prolonged exposure to high temperatures results in the permanent loss of rubber-like properties owing to chemical changes and degradations. However, these changes are time-dependent and, in critical applications, can to a certain extent be catered for by a planned replacement scheme.

For example, it may be possible to select a nitrile rubber for use at 180°C providing a service life of less than 50 hours is acceptable. For a more normal service life, say up to 5000 hours, a maximum temperature limitation would be in the region of 130°C with a working temperature of around 90°C.

The effect of extreme low temperature on rubber is somewhat different since the changes in physical properties are neither permanent nor time-dependent - that is, the rubber can be quite safely returned to its original state if it is gradually brought up to ambient temperature.

As the temperature decreases, the stiffness of the rubber increases until eventually the brittle state is reached. The ability to recover from a deforming force is therefore considerably reduced at low temperature, a fact that must be taken into account when selecting seal materials for applications with pulsating pressures.

A particular material may well maintain a seal at very low temperature under static pressure conditions, but leak under pulsating conditions at the same temperature owing to its inability to follow rapid cylinder dilations. The difference in temperature for successful sealing under these two conditions may well be as much as 20-40°C.
DYNAMIC RECOVERY

The ability to recover, even if only partially, from a deforming force is an essential quality for a seal material. This recovery provides the necessary initial sealing force to the surfaces and the ability to compensate for sudden movements caused by pressure pulsations and dimensional variations.

Three factors are quoted with respect to dynamic recovery:

*Compression set* is the amount of deformation retained by the rubber after a compressive load has been removed, and is usually measured after a number of days at elevated temperature in order to accelerate the test or simulate the application. Low values of compression set indicate high degrees of recovery.

*Stress relaxation* is the fall off in stress when rubber is subjected to a constant strain over a specific period of time. Although this is not of great importance in static situations at constant pressure, it becomes critical in dynamic situations and where seals are subjected to pulsating pressures, since it is one of the factors controlling the ability of the seal to recover and follow sudden movements.

*Rebound resilience* is an indication of the speed of recovery, a high value of which indicates that the material selected will possess the necessary ability to accommodate all the movements previously mentioned.

17.7 FILTERS

Strainers and filters are used to remove contaminating particles from a system. The terms *strainer* and *filter* are often used interchangeably because they have a common function.

A *strainer* may be defined as "a device for the removal of solids from a fluid wherein the resistance to motion of such solids is in a straight line".

A *filter* may be defined as "a device for the removal of solids from a fluid wherein the resistance to motion of such solids is in a tortuous path".

Strainers are generally designed of a fine mesh wire screen or a screening element made of specially processed wire of varying thickness wrapped around a metal frame (Fig. 17.15). Strainers do not provide the fine filtrating action, as do filters; thus they offer less resistance to flow. For this reason, strainers may be used in a pump inlet line to protect the pump from the larger sizes of detrimental particles.

Filters provide much finer filtration than strainers and consist of a material which, as the previous definition of a filter states, provides a tortuous path for solids.

Micron (or micrometre) is the term used to specify filter and strainer degree of operation, with filters being rated in *nominal value* or *absolute value*. 
The term nominal filtration indicates the filter’s ability to remove from the fluid up to 98 per cent of solid particles that are equal to or larger than the micronic rating of the filter. Thus, a filter nominally rated at 40 micrometre would remove 98% of solids more than 40 micrometre in size.

The absolute rating of a filter indicates the element pores through which the fluid must flow, but as the filter is used the holes become partly obstructed.

Thus, a filter rated at 60 micrometre absolute would have a nominal rating of about 35 micrometre. The absolute rating may be considered as the theoretical filtration level and the nominal rating considered as the practical filtration level.

Some basic filter designs are shown in Figure 17.16.

**FILTER MATERIALS**

There are several basic types of filter material available - paper, sintered metal, woven wire cloth, fabric and certain types of ceramic and plastic. The choice of filter material is determined by the fluid conveyed, chemical compatibility, working temperature, ability to withstand pressure and ability to be cleaned.

Paper elements, in the form of a circular concertina fold, are widely used for oil filters for vehicles or water filters for small swimming pools. By reinforcing the paper, nominal filtration down to 0.5 microns is possible and the filter elements withstand up to 150°C.

In general, the sintered metal powder media, including bronze and stainless steel, provide depth filtration with nominal ratings of 2 to 65 microns. The absolute ratings range from 13 to 100 microns.

Ceramic or plastic media are also used, but higher pressure drops may be expected as well as less resistance to mechanical and thermal shock.

Woven wire cloth elements are usually made of stainless steel and provide nominal filtration ratings of 2 to 100 microns with absolute ratings from 12 to 200 microns (micrometres). This type of filter media provides resistance to corrosion and fatigue.
Filtration is accomplished by essentially two classes of filters, surface filters and depth filters.

The surface filter accomplishes all its filtering action at the surface of the media. The pores are all very nearly of uniform size, and the action approaches the absolute. This class of filter is easy to clean and resists migration, but it has a comparatively low dirt-holding capacity.

The depth filter operates throughout the volume of the filter material by presenting many tortuous passages through which the fluid must flow. The pores or passages are not of uniform size, and the entrapment of particles depends on the depth and nature of the various passages. The action of this type of filter is primarily statistical because of the different sizes of the passages and the effect of fluid velocity, which could dislodge trapped particles if rapidly increased.

Depth filters operate successfully at lower flow rates and at relatively low pressure drops.

An example of a depth filter would be a circular wound yarn, which can provide a nominal rating of 1 to 100 microns.
REFRIGERATION

18.1 PRINCIPLES OF REFRIGERATION

Before considering the processes and methods of refrigeration some revision is necessary of the information given in Chapter 3 regarding steam.

SUPERHEAT

If heat is applied to a saturated vapour the result will be superheated vapour, the heat applied being called superheat. Since a conditional change has already taken place, sensible heat enters the picture - it is this that causes a temperature increase in the vapour.

The specific heat of a medium changes at the transition from the liquid phase to the vapour phase. For example, only 1.9 kJ is required to heat 1 kg steam 1°C, whereas the same temperature increase in water requires 4.187 kJ.

THE CONDENSATION PROCESS (Figure 18.1)

The reverse of a conditional change from liquid to vapour is from vapour to liquid, a process which is called condensation (precipitation). Instead of applying a certain quantity of heat it is necessary to remove the same quantity to transform vapour into liquid. Again, pressure determines the temperature at which condensation occurs.

Fig. 18.1 Condensation

TEMPERATURE/ENTHALPY DIAGRAM

The characteristics of a substance can be illustrated in a temperature/enthalpy diagram, where enthalpy is the abscissa and temperature the ordinate.

Enthalpy is often called heat content and is the sum of the energy applied to a medium. For clarification, water at atmospheric pressure has been chosen as an example in Figure 18.2.
The diagram begins with water at a temperature of 0°C where the enthalpy is also 0 kJ/kg. The application of sensible heat produces a conditional change from A to B (the evaporating temperature of water).

The difference between A and B corresponds to a temperature rise of 100°C. As previously mentioned, every 1°C rise in temperature requires 4.187 kJ; therefore the total heat that must be applied is 418.7 kJ.

Thus, the heat content equals the enthalpy and becomes 418.7 kJ/kg of water. (If you consult the ‘steam tables’ you will find a more precise value of 419.1 kJ/kg.)

Line BC corresponds to the latent heat of evaporation that is required to transform 1 kg of water (at point B) to dry saturated steam (at point C).

The evaporating heat of water at atmospheric pressure is 2256 kJ/kg and the enthalpy is the sum of the total heat supplied:

\[
\text{enthalpy} = 420 + 2256 = 2676 \text{ kJ/kg}
\]

It is important to note that no temperature increase occurs between B and C. Line CD shows the effect of the sensible heat on the steam, i.e. the superheat where at 120°C the total enthalpy is 2710 kJ/kg.

**PRESSURE/ENTHALPY DIAGRAM**

As was previously explained, the temperature/enthalpy relation is dependent on pressure. However, to show the temperature/enthalpy characteristics for media would involve making diagrams for all possible pressures.

Since this is clearly impractical a more flexible pressure/enthalpy diagram is used instead. Such a diagram is shown in Figure 18.3. A pressure is chosen for the ordinate, as a rule graduated in accordance with a logarithmic scale.
In refrigeration it is necessary to work with different pressures and temperatures and this diagram offers a practical way of graphically determining energy exchange in a plant.

![Pressure/Enthalpy Diagram](image)

**18.2 REFRIGERANT CIRCUIT**

Each item of the circuit will be considered separately to clarify the final overall system.

**EVAPORATOR (Figure 18.4)**

A refrigerant in liquid form will absorb heat when it evaporates and it is this conditional change that produces cooling in a refrigerating process. If a refrigerant at the same temperature as ambient is allowed to expand through a hose with an outlet to atmospheric pressure, heat will be taken up from the surrounding air and evaporation will occur at a temperature corresponding to atmospheric pressure.

If in a certain situation pressure on the outlet side (atmospheric pressure) is changed, a different temperature will be obtained since this is analogous to the original temperature - it is pressure-dependent.

The component where this occurs is the evaporator, whose job it is to remove heat from the surroundings, i.e. to produce refrigeration.

![Evaporator](image)
Refrigeration

COMPRESSOR (Figure 18.5)

The refrigeration process is, as implied, a closed circuit. The refrigerant is not allowed to expand to free air.

When the refrigerant coming from the evaporator is fed to a tank the pressure in the tank will rise until it equals the pressure in the evaporator. Therefore, refrigerant flow will cease and the temperature in both tank and evaporator will gradually rise to ambient.

To maintain a lower pressure, and, with it a lower temperature, it is necessary to remove vapour. This is done by the compressor which sucks vapour away from the evaporator. In simple terms, the compressor can be compared to a pump that conveys vapour in the refrigerant circuit.

![Fig. 18.5 Compressor](image)

In a closed circuit a condition of equilibrium will always prevail. To illustrate this, if the compressor sucks vapour away faster than it can be formed in the evaporator, the pressure will fall and with it the temperature in the evaporator.

Conversely, if the load on the evaporator rises and the refrigerant evaporates more quickly, the pressure and with it the temperature in the evaporator will rise.

CONDENSER (Figure 18.6)

The refrigerant gives off heat in the condenser, and that heat is transferred to a medium having a lower temperature. The amount of heat given off is the heat absorbed by the refrigerant in the evaporator plus the heat created by compression input.

![Fig. 18.6 Condenser](image)
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The heat transfer medium can be air or water, the only requirement being that the temperature is lower than that which corresponds to the condensing pressure.

The process in the condenser can otherwise be compared with the process in the evaporator except that it has the opposite sign, i.e. the conditional change is from vapour to liquid.

**EXPANSION PROCESS** (Figure 18.7)

Liquid from the condenser runs to a collecting tank, the receiver. This can be likened to the tank on the evaporator mentioned above. Pressure in the receiver is much higher than the pressure in the evaporator because of the compression (pressure increase) that has occurred in the compressor.

To reduce pressure to the same level as the evaporating pressure a device must be inserted to carry out this process which is called a throttling or expansion valve.

Such a device is therefore known either as a throttling device or an expansion device. Ahead of the expansion valve the liquid will be a little under boiling point. By suddenly reducing pressure a conditional change will occur; the liquid begins to boil and evaporate.

This evaporation takes place in the evaporator and the circuit is thus complete.

---

![Fig. 18.7 Expansion Process](image1)

**HIGH AND LOW PRESSURE SIDES OF THE REFRIGERATION PLANT**

There are many different temperatures involved in the operation of a refrigeration plant since there are such things as sub-cooled liquid, saturated liquid, saturated vapour and superheated vapour. There are however, in principle, only two pressures: evaporating pressure and condensing pressure. The plant then is divided into high pressure and low pressure sides, as shown in Figure 18.8.
THE COMPLETE CIRCUIT

The refrigerating system is compared with the pressure/enthalpy diagram in Figure 18.9. When the liquid passes through the expansion valve it changes from A to B, that is, there is a drop in pressure but no change in enthalpy.

At the evaporator input B there is a mixture of liquid and vapour, while at the evaporator outlet, C is a saturated vapour. C to D indicates the refrigerant passing through the compressor.

At the condenser inlet D, the saturated vapour starts to give off heat to the outside so that enthalpy changes to A.

The change in the condenser is in two stages. First it changes from a strongly superheated vapour at D to a saturated vapour at E, then condensation occurs towards point A.

18.3 REFRIGERATION COMPONENTS

COMPRESSOR

The job of the compressor is to draw vapour from the evaporator and force it into the condenser. The most common type is the piston compressor, Figure 18.10, but other types have won acceptance on larger systems, such as centrifugal and screw compressors.

The piston compressor covers a very large capacity range, right from small single-cylinder models for household refrigerators up to 8 to 10-cylinder models with a large swept volume for industrial applications.
In the smallest applications the hermetic compressor is used, where compressor and motor are built together as a complete hermetic (sealed) unit, as in Figure 18.11.

For larger plant the most common is the semi-hermetic compressor, shown in Figure 18.12. The advantage here is that shaft glands can be avoided; these are very difficult to replace when they begin to leak. However, the design cannot be used on ammonia plant since this refrigerant attacks motor windings.

Still larger refrigerant compressors, and all ammonia compressors, are designed as open compressors, that is, with the motor outside the crankcase.

**CONDENSER**

The purpose of the condenser is to remove the amount of heat that is equal to the sum of the heat absorbed in the evaporator and the heat produced by compression. There are many different kinds of condensers.

**Shell and Tube Condenser**

This type of condenser (Figure 18.13) is used in applications where sufficient cooling water is available. It consists of a horizontal cylinder with welded-on flat end caps which support the cooling tubes. End covers are bolted to the end plates.

The refrigerant condensate flows through the cylinder, the cooling water through the tubes. The end covers are divided into sections by ribs. The sections act as reversing chambers for the water so that it circulates several times through the condenser.
If it is desirable or necessary to cut down on the amount of water, an evaporating condenser can be used instead (Figure 18.14). This type of condenser consists of a housing in which there is a condensing coil, water distribution tubes, deflection plates and fans.

The warm refrigerant vapour is led to the top of the condensing coil after which it condenses and runs from the bottom of the coil as liquid. Water distribution tubes with nozzles are placed over the condensing coil so that water is spread over and runs down the coil.

The fans direct a strong flow of air across the condensing coil. When the falling drops of water meet the upward air flow some of the water will evaporate. This absorbs the necessary evaporating heat from the refrigerant vapour and causes it to condense.

The principle involving water evaporation is also used in connection with cooling towers (Figure 18.15). These are installed when the most practical is to place a shell and tube condenser near a compressor. The water is then circulated in a circuit between condenser and cooling tower.

In principle, the cooling tower is built up the same as the evaporating condenser, but instead of condensing elements there are deflector plates. Air is heated on its way through the tower by direct contact with the trickle of water travelling downwards and is able therefore to absorb an increasing amount of moisture coming from part-evaporation. In this way the cooling water loses heat.
Water loss is made up by supplying more water. It is possible to save 90-95% water consumption by using evaporating condensers or cooling towers, when compared to the water consumption of shell and tube condensers.

If, for one reason or another it is not possible to use water in the cooling process an air-cooled condenser is used (Figure 18.16), such as on a domestic refrigerator.

Since air has poor heat transfer characteristics compared with water, a large surface on the outside condensing tubes is necessary. This is achieved by using large ribs or fins and if possible, using fan circulation.

**Fig. 18.15  Cooling Tower**

**Fig. 18.16  Air-cooled Condenser**

**EXPANSION VALVE  (Figure 18.17)**

The main purpose of the expansion valve is to ensure a sufficient pressure differential between the high and low pressure sides of the plant. The simplest way of doing this is to use a capillary tube inserted between the condenser and evaporator. The capillary tube is, however, only used in small, simple appliances like refrigerators because it is not capable of regulating the amount of liquid that is injected into the evaporator.
A regulating valve must be used for this process, the most usual being the thermostatic expansion valve, which consists of a valve housing, capillary tube and a bulb. The valve housing is fitted in the liquid line and the bulb is fitted on the evaporator outlet.

![Expansion Valve Diagram](image)

**Fig. 18.17 Expansion Valve**

Figure 18.18 shows an evaporator fed by a thermostatic expansion valve. A small amount of liquid is contained in a part of the bulb. The rest of the bulb, the capillary tube and the space above the diaphragm is the valve housing which is charged with saturated vapour at a pressure corresponding to the temperature at the bulb.

The space under the diaphragm is in connection with the evaporator and the pressure is therefore equal to the evaporating pressure.

The degree of opening of the valve is determined by:

- The pressure produced by the bulb temperature acting on the top surface of the diaphragm.
- The pressure under the diaphragm which is equal to the evaporating pressure.
- The pressure of the spring acting on the underside of the diaphragm.

During normal operation, evaporation will cease some distance up in the evaporator. Then, saturated gas appears, which becomes superheated on its way through the last part of the evaporator. The bulb temperature will thus be evaporating temperature plus superheat. For example, at –10°C evaporating temperature the bulb temperature could be 0°C.

If the evaporator receives too little refrigerant the vapour will be further superheated and the temperature at the outlet pipe will rise. The bulb temperature will then also rise and with it the vapour pressure in the bulb element since more of the charge will evaporate.

Because of the rise in pressure the diaphragm becomes forced down, the valve opens and more liquid is supplied to the evaporator. Correspondingly, the valve will close more if the bulb temperature becomes lower.
EVAPORATION SYSTEMS (Figure 18.19)

Depending on the application, various requirements are imposed on the evaporator. Evaporators are therefore made in a series of different versions.

Evaporators for natural air circulation are used less and less because of the relatively poor heat transfer from the air to the cooling tubes. Evaporator yield is increased significantly if forced air circulation evaporators are used.

As the name implies, a liquid-cooling evaporator cools down liquid. The simplest method is to immerse a coil of tube in an open tank. Closed systems are coming into use more and more. Here, tube coolers made similar to shell and tube condensers are employed.

REFRIGERATION SYSTEMS

A typical system is shown in Figure 18.20, as would be found in a supermarket cool store or a butcher’s cool room.

The compressor unit may be located some distance from the evaporator, so that the condenser may be located near to a supply of cooling air.
Figure 18.21 provides an indication of the distribution of temperatures and pressures in the same plant.
Chapter Eighteen

18.4 APPLICATIONS OF REFRIGERATION

For convenience of study, refrigeration applications may be grouped into six general categories:

1. Domestic refrigeration.
2. Commercial refrigeration.
3. Industrial refrigeration.
4. Marine and transportation refrigeration.
5. Comfort air conditioning.
6. Industrial air conditioning.

It will be apparent in the notes which follow that the exact limits of these areas are not precisely defined and that there is considerable overlapping between the areas listed.
DOMESTIC REFRIGERATION

Domestic refrigeration is rather limited in scope, being concerned primarily with household refrigerators and home freezers. However, because the number of units in service is quite large, domestic refrigeration represents a significant portion of the refrigeration industry.

Domestic units are quite small, some as low as half a kilowatt input power.

COMMERCIAL REFRIGERATION

Commercial refrigeration is concerned with the designing, installation and maintenance of refrigerated fixtures of the type used by retail stores, restaurants, hotels and institutions for the storing, displaying, processing and dispensing of perishable commodities of all types.

INDUSTRIAL REFRIGERATION

Industrial refrigeration is often confused with commercial refrigeration because the division between these two areas is not clearly defined. As a general rule, industrial applications are larger in size than commercial applications and have the distinguishing feature of requiring an attendant on duty, usually a licensed operating engineer.

Typical industrial applications are ice plants, large food-packing plants (meat, fish, poultry, frozen foods, etc.), breweries, creameries, and industrial plants such as oil refineries, chemical plants, rubber plants, etc. Industrial refrigeration includes also those applications concerned with the construction industry.

MARINE AND TRANSPORTATION REFRIGERATION

Applications falling into this category could be listed partly under commercial refrigeration and partly under industrial refrigeration. However, both these areas of specialisation have grown to sufficient size to warrant special mention.

Marine refrigeration, of course, refers to refrigeration aboard marine vessels and includes, for example, refrigeration for fishing boats and for vessels transporting perishable cargo as well as refrigeration for the ship's stores on vessels of all kinds.

Transportation refrigeration is concerned with refrigeration equipment as it is applied to trucks, both long distance transports and local delivery, and to refrigerated railway wagons.

AIR CONDITIONING

As the name implies, air conditioning is concerned with the condition of the air in some designated area or space. This usually involves control not only of the space temperature but also of space humidity and air motion, along with the filtering and cleaning of the air.

Air conditioning applications are of two types, either comfort or industrial, according to their purpose. Any air conditioning which has as its primary function the conditioning of air for human comfort is called comfort air conditioning.

Typical installations of comfort air conditioning are in homes, schools, offices, churches, hotels, retail stores, public buildings, factories, automobiles, buses, trains, planes, ships, etc.

On the other hand, any air conditioning which does not have as its primary purpose the conditioning of air for human comfort is called industrial air conditioning. This does not necessarily mean that industrial air
conditioning systems cannot serve also as comfort air conditioning coincidentally with their primary function. Often this is the case, although not always so.

The applications of industrial air conditioning are almost without limit both in number and in variety. Generally speaking, the functions of industrial air conditioning systems are to:

1. control the moisture content of hygroscopic materials
2. govern the rate of chemical and biochemical reactions
3. limit the variations in the size of precision-manufactured articles because of thermal expansion and contraction
4. provide clean, filtered air which is often essential to trouble-free operation and to the production of quality products.
Chapter Nineteen

MATERIALS HANDLING

19.1 INTRODUCTION

The main purpose of materials handling is to achieve an effective and economical handling system at every stage in the industrial and commercial process, right from the introduction of the raw materials to the time when the products are finally put to use. However, before we can answer the question "How should we handle?", we need to carry out a detailed systematic analysis to answer such questions as "Is handling really necessary?" ... "Can it be avoided by revision of plant layout or adoption of more integrative production processes?"..."In what form should the materials be handled and with what frequency?"

Knowing all the answers, we can then decide on the type of equipment or design of installation which will be the most effective and economical for the particular handling requirements.

A considerable amount of equipment designed specifically to facilitate handling is on the market, but an effective materials handling system can be achieved only by correctly matching up the right equipment to the real needs of the situation.

DEFINITION AND OBJECTIVES

Materials Handling can be defined as: "Techniques employed to move, transport, store or distribute materials with or without the aid of mechanical appliances".

Its general objectives can be summarised in the saying "Getting the right goods safely to the right place at the right time at the right cost". A more cynical definition might be that material handling of a product adds something to the cost but nothing to the value.

Any handling and distribution system should:

- Function efficiently and effectively:
  - Efficiently - at the lowest overall cost
  - reliably (and be easily serviced)
  - safely
  - Effectively - do the job that really needs to be done.

- Help maximise the profits (or minimise the costs) of the procurement or storage or manufacturing or packaging/marketing/ distribution system as a whole.

- Be worth buying (that is, give an effective return on investment).

- Make the best use of what is already available.

More specifically the objectives can be detailed as follows:

To Lower Unit Materials Handling Costs
Handle as many pieces as possible as one load (unit load). Reduce rehandling time. Take the shortest route economically possible. Get there on time.

To Reduce Production Time
Waiting time can be eliminated by using good materials, handling methods and equipment, and by planning and controlling handling so as to ensure a continuous, uniform and maximum rate of working.
Remember that a product spends most of its time during manufacture waiting to be processed, moved or inspected. The first step to full automation in production is often the introduction of automatic handling methods between these stages.

**To Reduce Overheads**
Co-ordinated handling ensures that the material is at the right place at the right time. Introduction of the correct equipment should cut the labour force (indirect as well as direct) involved in handling, and also reduce the risk of damage in transit.

**To Conserve Floor Space**
Materials handling and plant layout studies go together, and handling equipment can be integrated with production machines to make the most of the space available. For example, many mechanical handling aids enable movement and storage to be taken off the floor and into the unused "air space" above. Matching materials handling with production reduces stock holdings and minimises work-in-progress.

**To Prevent Injuries To Workers**
Over 25 per cent of all industrial accidents occur in handling materials. Improvements in handling methods, combined with wider use of mechanical aids, would eliminate a great deal of manual lifting and so reduce the number of accidents.

### 19.2 CONVEYORS AND ELEVATORS

Conveyors are usually employed where the flow of work is steady, continuous and along fixed paths. Conveyors and elevators are available as fixed installations or in portable or mobile form.

**CONVEYORS AND ELEVATORS USED FOR MOVING BULK AND LOOSE MATERIALS**

**Troughed-belt Conveyors (Figs. 19.1 and 19.2)**
Troughed-belt conveyors incorporate a pair of pulleys, known as end drums, one of which drives the conveyor belt. The carrying strand of the belt is supported on troughing idlers and the return strand on flat rollers, the whole being based on a supporting structure.

The capacity of the conveyor is governed by the speed, width and depth of the belt troughing, and, of course, the strength of the belt is chosen to meet capacity requirements.

Belts are generally rubber covered, but can also be obtained in solid woven fabrics, nylon, etc. They are either manufactured endless or vulcanised on site, or manufactured in lengths and joined on site by belt fasteners.

Until recently, the great majority of conveyor belts were made from "India rubber and canvas" (I R & C). In recent years man-made fibres, such as rayon, nylon and polyesters, have been widely used for the body fabric of conveyor belts. They are stronger than cotton, and have as a result enabled a lighter belt, with fewer plies than the I R and C type, to be made for the same duty.

The diameter of the end drums of a conveyor is governed by the stiffness (number of plies) of the belt, so the greater flexibility of man-made yarn has enabled the end drums to be made smaller, all of which has reduced the weight, and, possibly the cost of the conveyor.
The "troughing" is imparted to the belt by means of the idler rollers carrying it. The idlers are generally of three- or five-roller type, and their length and diameter depend on the width and weight of the conveyor belt.

The wing rollers of the standard three-roller idler set used with I R and C conveyor belts are angled 20° to 30° to the horizontal. However, as man-made materials have greater flexibility than cotton, their use has enabled the trough to be made deeper than is possible with I R and C belts, and deep-troughed carrying idlers, with wing rollers set at 50° to 70° to the horizontal, are now available.

The deeper troughing enables a greater volume of material to be carried for the same width of belt, and in many cases it can lower the cost of the installation. Troughed belt conveyors can carry material up slopes as great as 16° to the horizontal. The slope can be made even steeper if the belt is fitted with special surfaces or with flights.

Troughed-belt conveyors range from 250 mm to 1500 mm in width. By adding sections, they can be designed to convey material over almost any distance, and, depending on the particular material, at rates reaching several thousand tonnes per hour.

**Side-Walled Conveyors (Figure 19.3)**

One method of increasing the carrying capacity of belt conveyors is to use what are known as walled belts - flat belts extended to form side walls.

**Flat-Belt Conveyors**

Flat belts may sometimes be used to handle bulk materials. The belt runs on flat tables or on rollers or on a combination of both. Side boards and skirts may be fitted to retain the material.

The material can be discharged using ploughs which sweep the material off the belt.
Steel-Band Conveyors

Steel-band belts, manufactured in carbon steel or stainless steel, offer advantages if the material to be conveyed is very hot (the bands can operate at temperatures as high as 250°C), acidic, sticky or wet, or when cleanliness is of paramount importance.

They are widely used in the food and chemical industries. The belt can be up to 300 mm wide, but the minimum diameter of the end pulleys is normally much greater than that of the end pulleys of rubber-belt conveyors.

Woven-wire Belt Conveyors

Mesh belts woven from stainless steel, mild steel, Monel or other metals are used to convey materials through ovens and quenching baths and, as hot air can be readily passed through the mesh, in washing and drying plants.

As in the case of steel-band conveyors, the end pulleys are of relatively large diameter.

One method of increasing the carrying capacity of belt conveyors is to use what are known as walled belts - flat belts extended to form side walls.

Apron-Conveyors (Figure 19.4)

The apron-conveyor consists of a continuously moving transporter attached to a pair of parallel, endless roller chains running on tracks supported by the conveyor framework, and carrying overlapping or interlocking plates or pans.

Apron-conveyors are rugged, relatively heavy and slow-running (moving at about 25 metres per minute). They can handle large quantities of heavy, lumpy, abrasive or hot materials, can operate up and down slopes, and can withstand conditions of impact loading for which belt conveyors are quite unsuitable.

Screw Conveyors

These are a development of the Archimedean screw. The material is moved along by the action of a screw or worm in the form of a helix rotating in a trough or tube. Generally manufactured throughout of metal (the troughs are sometimes coated or rubber-lined), screw-conveyors are suitable for handling wet, hot and otherwise difficult materials. The trough-type can be fitted with top covers to form a seal, retaining the materials and preventing ingress of moisture or contamination.

Various types of screws are available:

- full bladed - for maximum conveying capacity;
- ribbon type - suitable for materials that tend to adhere to the screw or screw shaft;
- or specially designed types which agitate or mix the material during conveying.
Screw conveyors can operate in the horizontal plane, on a slope or even vertically (although the carrying capacity is reduced as the slope increases).

They can handle a wide variety of materials (ranging from flour to fish waste, wet concrete to coal), but the material must be of a type that will readily drop out of the discharge apertures, otherwise the action of the screw will soon cause it to pack solid. They are not recommended for stringy material, such as paper trimmings or straw.

The screws of the traditional type of screw-conveyor run at speeds of 40 to 160 r/min. However, for higher speeds high-speed screw-conveyors (also known as auger-conveyors) are available. They are capable of handling most materials with free-flowing and non-abrasive characteristics, and can operate at very steep angles but, in general, they are best suited to intermittent or light duty.

**Vibrating-Conveyors**

These fall into two main groups:

- vibrating-conveyors with a high-frequency wave form, usually imparted by electromagnets or unbalanced pulleys;

- oscillating conveyors operating at slow speeds, with the oscillation imparted by an eccentric or crank shaft.

In both types of conveyor the material moves along a metal or plastic trough, covered or uncovered, or tube. The movement is a gentle one, for only the trough is in contact with the material, and this taken with the fact that the trough's surface can be kept smooth and clean, makes these conveyors attractive for handling foodstuffs or other material that must be protected from contamination.

Vibrating conveyors are generally built up from linking section of about 3 m in length. The material can be made to travel up slopes of up to about 10° and with spiral conveyors some materials can be elevated vertically.

**Gravity Bucket Conveyor/Elevators  (Figure 19.5)**

In this type of equipment, freely swinging buckets are carried between a pair of parallel endless chains which can be made to follow any path from vertical to horizontal.
The buckets are loaded by a specially designed feeder, and tipped or inverted to discharge. The device discharging the bucket can be set at any point where the conveyor has a horizontal run, and can be remotely controlled. Material can therefore be delivered into any one of a number of bunkers - even, by feeding different types of material into different buckets, as a controlled mixture.

The bucket-conveyor is slow-running (usually moving at a speed of about 15 metres per minute), rugged and capable of giving very good trouble-free service.

The buckets are generally made of steel, and hence can be used to carry hot, abrasive materials or those that tend to compact or cling together.

Conveyors of this type are widely used in coal-fired power stations for feeding coal into different bunkers and for handling hot ashes.

**Bucket-Elevators (Figure 19.6)**

There are many designs of this type of elevator, all of which consist of an endless chain, or chains, or an endless belt, to which buckets are attached, either at pitched distances or adjoining or overlapping one another, to form a "continuous bucket".

Generally, the elevators are totally enclosed by a metal casing which can be made dust or even gas-tight.

Material is fed in near the bottom or boot of the elevator, and is either dredged from the boot by the buckets, or caught by the buckets as they pass the inlet chute.

It is discharged either by gravity or by centrifugal action as the buckets invert after passing over the head of the elevator.

Various bucket-elevators are available, each with particular features which make them specially suitable for handling certain types of material. They include:

Chain- and bucket-elevators: they operate either vertically or in an inclined plane, with the buckets set at regular intervals or pitch. The chain usually moves at about 45 metres per minute and capacities ranged from 10 to 100 tonnes per hour, depending on the size and weight of the material. Bucket elevators can discharge at heights up to about 60 metres. They are suitable for handling free-flowing materials.

Centrifugal bucket elevators: these are generally vertical elevators, with buckets running at a speed of 60 to 150 metres per minute, high enough to fling the material clear of the buckets on discharge. The buckets are attached to a chain or the face of a belt, and are pitched at regular intervals to permit the material to be freely discharged as the buckets pass over the head sprockets or pulley.
Skip hoists

When very large quantities of material have to be elevated, or when the material consists of large lumps, a skip-hoist may be the most suitable and, probably, the only choice.

19.3 PACKAGE CONVEYORS

When a material or a component is packaged, its conveyance is possible by sliding or rolling.

Gravity Chutes

Perhaps the simplest method of using the force of gravity in handling packages or loose items is the chute.

Chutes can be curved or straight; flat bottomed or with bases shaped to conform with the loads handled (e.g. sacks); and manufactured in steel, timber or plastics, or lined.

Gravity-Roller Conveyors (Figure 19.9)

The gravity-roller is a cheap and convenient form of conveyor, readily assembled and adjusted, and suitable for handling a very wide range of loads.

Within limits, any load with a rigid, smooth base can be moved on roller conveyors; on the other hand, quite small protuberances can prevent free movement.

For instance, a timber box with steel strapping set at right angles to the rollers of the conveyor might be entirely satisfactory for gravity movement, but if the nail heads protrude from the strapping, they could catch in the rollers and inhibit free movement.
Similarly, a carton made from good hard paper board and well sealed and taped, could form an ideal load for conveying by gravity-roller, but, if softened by repeated use, or if its flaps are not properly secured, might pose difficulties, or at the very least require a much steeper slope for free movement.

Gravity-roller conveyor track is usually manufactured in 2.4 to 3 metre lengths, which can be linked together to form a continuous track. The carrying capacity is related to the diameter of the roller (British Standard 2567: Steel non-powered Roller Conveyors lists eight sizes of rollers, varying from 25 mm to 75 mm in diameter), and the pitch of the rollers must be such as to ensure that three rollers are always under the load.

**Flat-Belt Conveyors (Figure 19.10)**

Flat-belt conveyors, with the belt carried on rollers and/or sliding on a smooth surface, are suitable for moving relatively light-weight packages. They are available in units which can be joined together to form the desired system, and they are relatively cheap.

If India Rubber and Canvas (I R and C) is used for the belting, its underface is generally *frictioned* (that is, it is left with no rubber covering) in order to reduce the coefficient of friction between the belt and the table on which it slides.

![Fig. 19.10 Flat Belt Conveyor](image)

**Slat Elevator/Conveyor (Figure 19.11)**

The slat conveyor consists of a pair of parallel endless chains carrying timber or metal slats. The slats may be close or open pitch, fitted with flights to prevent loads sliding back, or of special shape to suit the load.

Slat conveyors can carry loads of considerable individual weight, and the slats can be quite wide: slats 1.5 m wide are common, and in some cases they are wide enough for an operator to stand on the moving conveyor to carry out his or her work.

The conveyor path can be sloped up to an angle of about 25° without flights, and to an even greater angle with flights. Since the conveyor chains run in tracks, the path can be changed from the horizontal to sloping and back again without problems.
Overhead Chain or Cable Conveyors (Figure 19.12)

This type of conveyor can follow a labyrinthine path with no difficulty at all, and the direction can be varied as required in both the horizontal and vertical planes. The loads, anything from a few kilograms in weight, are moving (endless) chain.

The hangers can be made to tip and discharge their loads automatically at pre-determined points, and they can be made to move at anything from a few millimetres per minute to 30 m per minute.

With some of the lighter types of conveyor the track can actually ascend vertically, but with heavier types the maximum gradient is usually about 30°, and the minimum radius for curves is about 4 metres.

19.4 PNEUMATIC CONVEYING

Pneumatic conveying of bulk solids has many attractions, particularly when the materials to be moved are powdered or granular. Stringy materials can be moved in this way, but special attention has to be given to inlet and discharge points.

This system makes use of ducts or pipes (known as transport lines) through which the material is carried, mixed with air or another suitable gas. The lines can follow tortuous paths, passing through floors and walls or under roads, and changing from vertical to horizontal in a way with which mechanical conveyors just could not compete.

The pneumatic-conveying system can be made dust-proof, and can be even totally enclosed, and very few operators are required for normal operation.
Fine, free-flowing powders or granules are the easiest to convey pneumatically; materials that tend to stick or compact or are abrasive are less suitable. Pneumatic conveyors are capable of moving material many hundreds of metres, and in very considerable quantities, for the rate of conveying is maintained as long as material is being fed into the system.

However, they have a higher power consumption per unit of material conveyed than mechanical conveyors and elevators.

Abrasion of transport lines, particularly at bends, can be troublesome, and the possibility of risk of explosion as a result of generation of static electricity must be constantly guarded against.

It is as well to point out here that expertise in pneumatic conveying of solids is still concentrated largely in the hands of the established manufacturers and research establishments and that the theory available from textbooks is more useful as a guide than as a basis for detailed application.

**NEGATIVE AND POSITIVE PRESSURE SYSTEMS**

The two most common pneumatic conveying systems are:

- negative-pressure or vacuum system;
- positive-pressure or blowing system.

In both, the solid material has to be:

- fed into the air stream;
- conveyed by the air stream;
- separated from the air stream and discharged from the system.

The conveying air has then to be disposed of, either by discharge to atmosphere (usually after passing through a filter) or by recirculation. Negative-pressure systems are usually chosen when handling hazardous materials, or when picking up from a number of collection points to convey to a single discharge point (for example, scrap-collection or dust-collection systems).

Positive-pressure systems are often chosen when material from a single point (a bulk storage site) has to be delivered to a number of discharge points, such as extruder machines in a plastics factory.

**Negative-Pressure Systems (Figure 19.13)**

In the simplest negative-pressure system, the material is fed into a hopper and then sucked into the conveying line. For discharge it is led into a cyclone, in which the air expands, loses velocity, and can no longer hold the solid material in suspension.

The material drops to the base of the cyclone, and thence, by gravity, into a power-driven rotary-vane valve. The conveying air continues through the cyclone, passes through a filter (which extracts any very fine dust it may still be carrying), and so back to the pump, fan or compressor for recirculation.
Positive-Pressure Systems (Figure 19.14)

In the simplest positive-pressure system, the conveying air is blown along the transport line and picks up material introduced into the line by a power-driven rotary-vane valve or other type of feeder. At the end of the journey the air-borne solids are carried into a cyclone for discharge, dropping out of the air stream because of the reduced velocity and then falling by gravity into a power-driven rotary-vane valve or other type of seal.

The conveying air continues on through the cyclone and thence (via an air filter or dust collector) to the atmosphere.

FLUIDISATION

The material being handled is fluidised by enveloping each of its discrete particles in air, so that they offer little or no resistance to relative movement, and the material flows like a liquid. The technique is being increasingly used to facilitate the discharge of powdered or granular material from bins, hoppers and silos, and for high-pressure conveyance in ducts.

It can be applied only to materials which fluidize readily (e.g. flour, sand, fly-ash, wheat), and which, whilst reasonably free-flowing, will retain the entrained air, so that there is no risk of solids dropping out and blocking the conveying ducts.
**Fluidising Conveyors (Figure 19.15)**

The simple fluidizing conveyor consists essentially of a trough, divided by means of a porous membrane from an air chamber. A small quantity of dry, low-pressure air is fed into the air chamber, whence it passes through the porous membrane (e.g. solid woven cotton belting, plastic or ceramic tiling) and into the material being conveyed.

When the correct quantity of air is filtering through the membrane into the material, the material will begin to boil. At this stage the material will behave like a liquid, and if a slight slope is imparted to the conveyor, will flow to the lower end.

![Fluidising Conveyor](image)

With suitable materials (cement is an outstanding example), and provided the distances involved are relatively short, this type of conveyor is very useful, for there are no moving parts to become clogged with powder, and dust is little or no problem. With specialised equipment, a fluidized powder may even be pumped, as if it were a liquid.

**Liquid As A Conveying Medium**

Slurries, for example concrete, can be pumped over short distances. Powdered solids in liquid suspension, e.g. coal in water, can be moved economically by pipeline over long distances.

**19.5 CRANES AND LIFTING EQUIPMENT**

**HAND-OPERATED EQUIPMENT**

Lifting blocks, cranes and winches are available for manual operation in capacities up to about 25 tonnes. Hand-operated equipment is still very widely used for lifting loads of up to about 10 tonnes, and is quite efficient where the height of lift is not too great and usage is intermittent.

Hand equipment up to 25 tonnes in capacity is often installed for periodic use in plant maintenance, or in remote buildings with no electric power, or in generating stations (in case of power failure).

Mobile hand-operated lifting blocks are used for construction and similar work or for jobs which occur only infrequently, e.g. for lifting out the rollers of a printing press. Hand-operated winches are also frequently used in erecting buildings and plant. Equipment for making horizontal pulls is usually manually operated.

**Chain Blocks (Figures 19.16 and 19.17)**

There are three types of hand-operated pulley block or hoist block which use steel chain to lift the load:
• worm-geared
• spur-geared
• triple-geared.

They are available in capacities ranging from 0.25 to 25 tonnes. The smaller sizes lift the load on a single fall of chain, whereas a 25-tonne block would have six or eight falls of chain. The older worm-geared block is now giving way to the spur-geared block or to the triple-geared block, both of which have a higher efficiency and can be made much lighter.

The load is lifted by pulling on the hand chain, and is automatically held at any height by self-sustaining brakes, without the need to maintain the pull on the hand chain. The load which a pulley block can lift is related to the velocity ratio of the mechanism; i.e. the distance the effort moves divided by distance the load moves.

With a 1-tonne triple-gear block, it might be necessary to pull 10 metres of hand chain in order to lift the load 300 mm.

**Hand-Operated Winches (Figure 19.18).**

Where the position is fixed, and the lift high or the load heavy, crab winches lifting on wire rope and operated by handles are frequently more attractive than pulley blocks, which tend to become heavy and cumbersome as their capacity increases.

The winch is often located at floor level, with the lifting rope led over a system of sheaves (pulleys) producing the required mechanical advantage.
Horizontal Pullers (Figure 19.19)
For limited horizontal pulls, and for angled or even vertical pulls, a device known as a leverpull or pullift is useful. It is light, can be manually operated by means of a ratchet-type lever in restricted positions; is available in capacities ranging from 0.25 to 3 tonnes, and generally moves the load up to about 2 metres.

For longer horizontal movement, devices pulling with wire ropes (for example, monkey winches) are available.

Electric Hoist Blocks (Figure 19.20)
Electric hoist blocks, which lift on one or more “falls” of steel wire rope, are widely used in industry. They are available in various types and capacities (ranging from 0.25 to 7.5 tonnes) and, depending on the type and capacity, lifts to about 15 metres.

The standard unit lifts at a speed of 3 to 20 metres per minute. Dual-speed blocks with a slow speed one-quarter that of the fast speed are on the market.
The wire lifting ropes are wound around a grooved barrel, and the load is sustained in any position by the use of a solenoid-operated brake. Devices to control the lowering speed can also be fitted. All electric hoist blocks incorporate an overwinding circuit breaker which stops the motor when the hook reaches a predetermined height.

Circuit breakers can also be fitted to prevent overlowering, in which case they are generally actuated after the barrel has made a predetermined number of revolutions.

**Overhead Travelling Cranes (Figs 19.21 and 19.22)**

Overhead travelling cranes, sometimes called bridge or gantry cranes, run on a pair of parallel tracks (the gantry). They are available in forms suitable for hand or electric operation or a combination of both, and can be of single- or double-girder design:

There are two main types:

- overslung, in which the cross girders are borne on top of the end carriages, as in Figure 19.21
- underslung, in which the cross girders are suspended from the end carriages, as shown in Figure 19.22.

Single-girder cranes incorporate a block and trolley running on the bottom flange of a single cross girder; double-girder cranes lift by means of a crab running on the top flanges of a pair of parallel girders.
Hand-operated single-girder cranes vary in capacity from 0.5 to 15 tonnes, and spans range from 3 to 15 m; hand-operated double-girder cranes vary in capacity from 2 to 30 tonnes, with a span range similar to that for single-girder types.

Electrically-operated single-girder cranes, or single-girder cranes using an electric block for hoisting but with cross travel and longitudinal movement achieved manually, are available in capacities ranging from 0.5 to 5 tonnes.

Electrically-operated double-girder cranes can be obtained in versions capable of handling maximum loads of 1 to 200 tonnes or more and with spans up to 30 m or more (the cross girders of the larger cranes being of box-section or lattice types).

**Mobile Cranes**

Mobile cranes are used for a very wide range of applications. They can be roughly divided into three classes:

- workshop cranes
- yard cranes
- contractors or erection cranes.
Mobile cranes are available in many capacities, and powered either by diesel or by petrol engine. The engines either drive direct, or are used to generate electricity to drive the hoisting, **luffing** (raising of the jib) or **slewing** (swinging the jib sideways) mechanisms.

The rated capacity of mobile cranes is influenced by the outreach of the jib (that is, the overturning moment) and changes as the jib boom changes its angle. The following figures illustrate this effect.

Assuming the crane has a maximum capacity of 6 tonnes and a 6 metre jib, the rating varies as follows:

<table>
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<tr>
<th>Radius of jib (m)</th>
<th>Safe Load (tonne)</th>
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</table>

The above ratings apply to the crane in the fully mobile state. Some cranes can make their maximum lifts only when supported on stabilising jacks.

**Workshop Cranes**

Workshop cranes are designed to be low enough (generally with the jib in the horizontal position) to pass under doorways or obstructions at least 3 metres from the floor. Capacities range from 1 to 10 tonnes, and reach extends to about 6 metres.

**Yard Cranes**

Yard cranes can be obtained in capacities from 3 to 20 tonnes and with jibs of up to 12 metres, which can sometimes be fitted with extensions.

**Contractors Cranes**

The range is very wide, with jibs of 30 metres or more. For movement these cranes are fitted with heavy-duty pneumatic tyres or crawler tracks (frequently they are lorry-mounted), but stabilisers have to be used when lifts are being made. They are employed mainly by building and steel-work contractors (hence the name) or hired to carry out specific erection tasks.

**19.6 POWER REQUIRED FOR MECHANICAL HANDLING**

The power required to move a given quantity of material over a given distance in a stated time is dependent on a large number of factors, so we will only consider the variables applicable to a belt conveyor system.

The total power required is a combination of three quantities:

- to move the belt
- to move the load horizontally
- to elevate the load.

The power to move the belt itself is a function of the belt material, construction and method of support, and is determined from the manufacture's catalogue.
The power to move the load is basically determined from:

\[ \text{Power} = \text{force} \times \text{velocity} \]

In terms of a belt, the *force* is the tension in the belt, and the *velocity* is simply the belt linear speed. The factor that is hard to determine is the belt tension (or force in the belt), because it is a function of:

- belt material, and thus the allowable tension
- type of support rollers
- density of the material conveyed
- volume (or mass) of the material conveyed
- belt length and width

Values tend to be empirical, supplied by the belt manufacturer and based on a series of practical tests. A manufacturer will supply tabular data to aid belt power calculations, similar to Tables 19.1, 19.2 and 19.3.

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Table 19.1
Chapter Nineteen

19.19

Table 19.2

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Table 19.3

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<td>136.1</td>
<td>171.5</td>
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<td>272.2</td>
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</table>

For example:

A 125 metre long conveyor carries 400 tonnes per hour on a 650 mm wide belt travelling at 2.5 metres per second up a 17° incline (33 metres lift). Estimate the power required and the belt tension.

From Table 19.1, using input values of 650 mm wide and 125 m long:

\[
\text{Power to move the belt} = 1.9 \times 2.5 \text{ m/s} = 4.8 \text{ kW}
\]
From Table 19.2, using input values of 400 t/h and 125 m length:

\[
\text{Power to move the loaded belt} = 4.8 \text{ kW}
\]

From Table 19.3, for 33 m (25 + 8) elevation:

\[
\begin{align*}
\text{Power to elevate 25 m} &= 27.2 \text{ kW} \\
\text{Power to elevate 8 m} &= 8.6 \text{ kW} \\
\text{Total power} &= 45.4 \text{ kW}
\end{align*}
\]

\[
\begin{align*}
\text{Power} &= \text{belt tension} \times \text{speed} \\
45.4 \times 10^3 &= \text{belt tension} \times 2.5 \\
\text{Effective Belt tension} &= 18.2 \text{ kN}
\end{align*}
\]

For a 650 mm wide belt, this would be expressed as a tension of 28 kN/m

Note that the tension value determined is for normal belt operation, the maximum tension would have to take into account the starting tensile loads on the belt, and the effect of sudden overloads.
LUBRICATION

20.1 INTRODUCTION

Bearing lubrication has already been discussed in Chapter 10, and it would be a good idea to re-read the second half of Chapter 10, as a reminder of lubrication terminology.

It is convenient to divide them into four basic types:

- **Oils** - A general term to cover all liquid lubricants, whether they are mineral oils, natural oils, synthetics or emulsions.

- **Greases** - Technically these are oils which contain a thickening agent to make them semi-solid.

- **Dry Lubricants** - These include lubricants which are made up in a solid form, and may be bulk solids, paint-like coatings or loose powders. Popular dry lubricants are graphite and molybdenum disulphide.

- **Gases** - The gas normally used in gas bearings is air, but any gas may be used that will not attack the bearing, or that does not decompose.

The easiest way to summarise the characteristics of the three more popular lubricants is in the form of a table, as in Table 20.1.

20.2 TYPES OF LUBRICANT

The two main factors affecting the choice of a lubricant are the **speed** and the **bearing pressure**. Figure 20.1 shows the speed and pressure limits for various lubricants.

The boundaries in Figure 20.1 are drawn thick because they are rather arbitrary - the actual speed/load limit of a particular application will depend on the type of component used, as well as the nature of the lubricant.

![Fig. 20.1  Speed and Load Limits for Different Lubricants](image-url)
For example: the maximum specific bearing pressure for a grease at low speeds can vary from 2000 kN/m² for a plain soft grease to 6000 kN/m² for an EP (extra pressure) grease or a molybdenum disulphide grease.

Bearing pressure is equal to the shaft load divided by the projected area of the bearing. A 40 mm diameter bearing that is 60 mm long would have a projected area of (40 x 60) mm².

Similarly, the upper speed limit for dry lubricants is shown as about 500 mm/s because they are poor conductors of heat and tend to overheat at higher speeds.

<table>
<thead>
<tr>
<th>Lubricant Property</th>
<th>OIL</th>
<th>GREASE</th>
<th>DRY LUBRICANT</th>
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<td>Hydrodynamic lubrication</td>
<td>Excellent</td>
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<td>Nil</td>
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<tr>
<td>Boundary lubrication</td>
<td>Poor to excellent</td>
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<td>Cooling</td>
<td>Very good</td>
<td>Poor</td>
<td>Nil</td>
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<td>Low friction</td>
<td>Fair to good</td>
<td>Fair</td>
<td>Poor</td>
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<tr>
<td>Ease of feed to the bearing</td>
<td>Good</td>
<td>Fair</td>
<td>Poor</td>
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<tr>
<td>Temperature range</td>
<td>Fair to excellent</td>
<td>Good</td>
<td>Very good</td>
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<tr>
<td>Volatility</td>
<td>Very high to low</td>
<td>Generally low</td>
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<tr>
<td>Flammability</td>
<td>Very high to low</td>
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<td>Life determined by:</td>
<td>Deterioration and contamination</td>
<td>Deterioration</td>
<td>Wear</td>
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Table 20.1

As a general rule, as the shaft speed increases, the lubricant must be less viscous. With increasing speed, the lubricant changes from solid to grease to oil to gas. Conversely, as the load (or bearing pressure) increases, the lubricant's viscosity must increase.

20.3 LUBRICANT PROPERTIES

Considering oils, the most important property of a lubricating oil is its viscosity. As far as the actual lubrication is concerned the only other important factor is the boundary (or thin-film) quality, as discussed in Chapter 10.

In practice, many other factors are taken into account, to make sure the oil continues to lubricate over a long period, or to make sure that the system does not fail for some other reason.

*Thermal or temperature stability* - If an oil is in use, it is important that the heat does not make break down

*Chemical stability* - An oil can be chemically attacked by oxygen from the air, or by water, or some other substances with which it comes into contact. Chemical stability means the ability to resist chemical attack, with respect to the substances adjacent to the oil. Chemical stability is related to thermal stability, because the speed of a chemical reaction increases as the temperature increases.

*Compatibility* - interaction of the oil and surrounding materials. For example, an oil may cause the rubber of a seal to swell or shrink, to soften or harden.

*Corrosiveness* - Corrosion may occur when the oil, or at least some compound in the oil attacks a metal component in the system. An oil may be non-corrosive when new but become corrosive after a period of use, due to oxidation and the formation of acids.
Thermal conductivity - This is important where the oil is expected to conduct heat away from the bearing.

Flammability - obviously an oil should not catch fire under the conditions under which it is used. This is particularly important in some industries, such as coal-mining or aviation.

In many industrial applications, the selection of a certain hydro-carbon oil may not be enough to cope with the working conditions imposed on it. Additives are then incorporated into the oil to enhance its properties for an exceptionally arduous condition.

The additives incorporated into the oil generally go into solution in the oil, but sometimes may be present in colloidal form. The main types of additives for mineral oil are:

- oxidation inhibitors – to prolong the life of the oil,
- rust and corrosion preventives,
- load-carrying and frictional-characteristic improvers,
- pour-point improvers – to keep the oil fluid,
- viscosity-index improvers – to keep the viscosity constant over a temperature range.

Oxidation
Present day refining techniques yield oils of excellent oxidation stability, especially if they are of paraffinic origin. However, in certain applications, the equipment operates under certain conditions which impose a high demand on the oil; such as when the oil is very hot and in association with a lot of air plus exposed metal components. Copper is one metal that encourages oxidation.

As an oil oxidises it initially forms oxides, which change to ketones, alcohols and acids. The products of oxidation dissolve in the oil, but later on they precipitate as resins and a sludge. The solids settle out of the oil as well as increase the oil's viscosity. The acids cause corrosion of metallic components.

Anti-oxidation additives act to delay the onset of initial oxidation, by either slowing down the chemical process or providing a barrier between the oil and metals that would act as a catalyst for oxidation.

Note that eventually the anti-oxidants are all used up, and the rate of the subsequent oxidation is rapid. Figure 20.2 provides an indication of the relationship between oxidation and time.

![Fig. 20.2 Oxidation of a mineral oil](image)
Lubrication

Figure 20.3 indicates the relationship between oil temperature and oil life. The rates of oxidation or thermal degradation increase with temperature, so that higher temperatures may only be tolerated if the required life is shorter.

![Temperature limits for mineral oils](image)

**Fig. 20.3** Temperature limits for mineral oils

**Rusting**

It seems a paradox that rusting can occur in a lubrication system with so much oil present. Remember that some moisture exists in the oil, and the moisture has a tendency to be acidic (particularly if oxidation is happening). Rusting is common at the junction of oil and air.

Rust in the oil acts as an abrasive and also acts as an oxidation catalyst.

Rust inhibitors in the oil consist of vapour phase corrosion inhibitors, which function by possessing relatively high vapour pressure, allowing them to migrate from the oil solution into air spaces where they are absorbed onto the metal surfaces to be protected.

**Anti-foam**

Mineral oils dissolve air. The amount of air dissolved depends on the air pressure and, to a lesser extent, the temperature. If the air stayed in solution there would be no problem, but if the air pressure above the oil is suddenly reduced the dissolved air forms small bubbles - the oil foams.

This a serious problem in hydraulic systems because air bubbles in the hydraulic fluid make it compressible and the system becomes unstable.

Anti-foam additives change the surface tension value of the oil, which reduces its ability to form air bubbles.

**Load carrying**

Extreme pressure additives are included in oils when the load, temperature or velocity between two surfaces does not allow a hydrodynamic oil film to build up. There is nothing to prevent metal surfaces coming in contact, with resulting wear, unless a load-carrying additive is present in the oil.
The additives used are compounds of sulphur, chlorine, phosphorus or lead. They remain inert at normal temperatures and pressures but, as heavy loads occur, the chemicals release agents which, on reaction, yield metallic films such as chlorides and sulphides. These films prevent welding and metallic pick-up between the surfaces under heavy duty conditions.

**20.4 SELECTION OF A LUBRICANT**

As previously stated, the most important variable for a lubricant is its viscosity. Furthermore, the correct viscosity oil must be used at the *working temperature* of the oil.

For instance, if the correct viscosity for a machine is 10 centistokes (cSt), then it must have that oil whether it is operating at 20°C or at -20°C. In practice, of course, no oil can maintain a set viscosity over a range of temperatures, so the correct viscosity oil is used at the machine's normal operating temperature.

![Fig. 20.4 Viscosity selection for plain bearings](image)

Many bearing manufacturers provide charts that recommend a suitable oil for a specific bearing. Figures 20.4 and 20.5 give an indication of the type of oil viscosity required for plain bearings and rolling bearings.
Fig. 20.5 Viscosity selection for rolling bearings
MECHANICAL EQUIPMENT
Second Edition

DESCRIPTION
Topics include: Mechanical Concepts; Fluids and Thermal Concepts; Steam Measurement of Mechanical Variables; Fuels; Piston Engines; Steam and Gas Turbines; Mechanical Drive Components; Brakes and Clutches; Bearings; Belt and Chain Drives; Gears and Gearing; Boilers; Pipework and Valves; Compressors; Hydraulics And Pneumatics; Seals, Packings and Filters; Refrigeration; Materials Handling; Lubrication.

EDITION UPGRADE
Second

CATEGORY
Engineering