HAPTER – 1

THE UNITS, MASS, FORCE, WEIGHT & BASIC DEFINITIONS

S.I. Units :

Before dealing with the definitions of such fundamental terms as mass, force, weight and the principles of applied mechanisms it is essential to explain the units used in this book as they will be the units of the future not only in this country but also in other leading countries. They are known as S.I. units. At present the units in our country are of M.K.S. system ; M stands for metre (unit of length), K for kilogramme (unit of mass), and S for second (unit of time).

The letters S.I. are an abbreviation for the full name *Systems International d' units* (International System of Units). S.I. is the system of units introduced by a general conference of leading countries, including India, on weights and measures in 1960. It is based upon metric units. A conspicuous feature of S.I. units is the adoption of a unit called Newton as a unit of weight of force. Some other units in S.I. which attract attention are :

Joule (unit of work or energy).

Pascal (unit of pressure)

Watt (unit of power) : This unit is used not only for electric motors but also for internal combustion engines which used horse-power as a unit so far. In S.I. units the unit horsepower hasno place.

Six basic quantities in S.I. are :

Quantity	*Unit*	*Symbol*
Mass	kilogramme	kg
Length	meter	m
Time	second	s
Temperature	kelvin	K
Electric current	ampere	A
Luminous intensity	candela	cd

Other units in S.I. are derived from these basic units.

A "metre per second" is known as a *derived unit* because it measures one quantity, speed, in terms of bother quantities, length and time. Units of area and volume, are also derived units.

Mass. The unit of mass, *kilogramme*, is the mass of a plati-num-iridium cylinder kept at the International Bureau of Weights and Measures at Sevres, near Paris. As the basic unit of mass is kilogramme, the word gramme does not Fig. in S.I. units; e.g., density of water is 100 kg/m². One should not make the mistake of writing it as 1 Mg/m³; it should be written as 10^3 kg/m³, or 1000 kg/m³.

Length : The unit of length, the *metre,* is distance between two lines on a platinum-iridiumm bar which is kept at the International Bureau of Weights and Measures at Sevres. The measurment is made at 0°C and standard atmospheric pressure.

Time : The unit of time, the *second,* is based upon the mean solar day, this being the average time between successive transits of the sun during the year. 1 second = 1/86400 of a mean solar day.

This time is indicated fairly accurately by a "quartz" clock or watch.

Temperature : The unit of thermodynanic temperature, kelvin, is the degree interval on the thermodyanmic scale. The temperature of the ice freezing point of water on this scale is 273.15K, corresponding to 0°C on the Celsius scale. Conversion from °C to Kelvin, for all practical purposes :

Temperature in Kelvin = 273 + t °C

The temperature intervals on the Klevin and Celsius scale are equal. The sign of degree (°) is not used for Kelvin.

Note : The international definitions of the metre and the second are more complex and are of only acdemical interest to students and readers of books on engineering.

The S.I. is a coherent system of units. A system of units is coherent if the product or quotient of any two unit quantities in the system is the unit of the resultant quantity.

All other units in S.I. are derived from the six basic units mentioned here and the commonly used derived units are the following :

i. The unit of force, the Newton (N), *i.e.* that force which when applied to a mass of one kilogramme, will give it an acceleration of 1 m/s².

ii. The unit of pressure, the Pascal (Pa), *i.e.* the pressure produced by a force of one Newton applied, uniformly distributed, over an area of one m² (1 N/m²). 1 bar = 100 k Pa. The value of standard atmospheric pressure, internationally accepted, is 101, 325 N/m². This is very close to the unit bar (b) which is 10^5 N/m², *i.e.* 100,000 N/m². Meteorologists generally use as their unit the *bar,* and quite often unit *millibar,* which, as the name implies, is one-thousandth of a bar.

iii. The unit of energy, Joule (J), *i.e.* the work done when the point of application of a force of one Newton is displaced through a distance of one metre in the direction of the force.

iv. The unit of power, the Watt (W), *i.e.* the rate of doing work at one joule per second.

All the derived units in mechanics can be expressed in terms of the basic unit of length, mass and time. For example,

$$\text{Density} = \frac{\text{Mass}}{\text{Volume}} = \frac{\text{Mass}}{(\text{Length})^3} = \frac{\text{M}}{(\text{L})^3} = \left(\text{ML}^{-3}\right)$$

$$\text{Velocity} = \frac{(\text{L})}{\text{T}} = \left(\text{LT}^{-1}\right)$$

$$\text{Acceleration} = \frac{\text{Change in velocity}}{\text{Time}}$$

$$= \frac{(\text{L})}{(\text{T})(\text{T})} = \frac{(\text{L})}{(\text{T})^2} = \left(\text{LT}^{-2}\right)$$

L, M and T denote the dimensions of length, mass and time respectively.

The S.I. units are independent of g, the acceleration due to force of gravity, and are clear in statement, for example,

P (Newtons) = M (Kilogramme) × f (m/s²)

Permitted non S.I. units. The S.I. System has given a proper allowance for some non-S.I. units that are in use for a long time and have a bearing with MKS system. These units are permitted because they have certain advantages in particular fields, e.g. litre, atmosphere (for pressure), °C for temperature, are some or the permitted non S.I. units.

The prefixes attached to the unit to denote decimal multiple or submultiple of the latter are as follows :

Multiplier	Standard for	Prefix	Abbreviation
One thousand million	10^9	giga	G
One million	10^6	mega	M
One thousand	10^3	kilo	k
One hundred	10^2	hecto*	h
Ten	10^1	deca*	da
One-teeth	10^{-1}	deci*	d
One-hundredth	10^{-2}	centi*	c
One-millionth	10^{-6}	micro	μ
One-thousand millionth	10^{-9}	nano	n

The 1275000 N/m² = 1275 kN/m² = 1.275 MN/m².

In using these multiples and sub-multiples, only one prefix should be used : thus 1275000000 N/m² or 1.275 GN/m², not 1.275 kMN/m². One thousand Newtons have to be written as 1 kN or as 10^3 N.

For non-decimal figures the numerical values are written in India as follows :

91435762 is written as 9,14,35,762

since we have such units as *lakh* (also written as lac) and crore; 1 lakh = 100000; 1 crore = 10000000. The modern practice in foregin countries is to write a numerical value in groups of three digits starting from the right hand side without the use of comma. Thus, the figure 91435762 is written as 91 435 762. For writing decimal figures the practice in British and American books is to form groups of three digits commencing from the decimal point on the right hand side and also on the left hand side. For example,

15692.43 as 15 692.43

10.54737 as 10.54737

Western countries have such high units as a million (1 000 000), a billion (1 000 000 000) and a trillion which is eqvalent to a million million in America (1 000 000 000 000).

These particular prefixes are gradually falling into disuse.

In this book the practice of forming groups of three digits is not rigidly adopted.

At many places in this book, for conversion of quantities of FPS or metric system of units into S.I. units use has been made of unity brackets ans **Stroud system of conversion.** In this system, devised by Professor Stroud, *unity brackets* are used. Since 60 minutes equal one hour, if we write

$\left[\frac{60\ \text{min}}{1\text{h}}\right]$ then this is called a *unity bracket*, Unity brackets can be written with

either quantity as numerator, e.g. $\left[\frac{1\ \text{min}}{60\ \text{s}}\right], \left[\frac{60\ \text{s}}{1\ \text{min}}\right], \left[\frac{1\ \text{kN}}{1000\ \text{N}}\right], \left[\frac{1000\ \text{kN}}{1\ \text{kN}}\right]$

In most problems, unless the units of measurement are given as basic units, they must be changed into basic units. Use of Stroud system is clarified in the following example.

Example :

Change 72km/h to m/s.

Ans. $$72\text{km / h} = 72\ \frac{\cancel{\text{km}}}{\cancel{\text{h}}}\left[\frac{1000\text{m}}{1\ \cancel{\text{km}}}\right]\left[\frac{1\ \text{h}}{60\ \cancel{\text{min}}}\right]\left[\frac{1\ \cancel{\text{min}}}{60\ \text{s}}\right].$$

$$= \frac{720\ \text{m}}{36\text{s}}$$

Basic terms.

Mass and weight :

Mass is the measure of quantity of matter in a body and is a constant property of that body. The unit of mass in MKS units as well as in S.I. units is *kilogramme.*

There is a difference between *mass* and *on weight.* If a person holding a stone in his hand releases his grip on it, is falls to the earth. Clearly a force must have pulled the stone towards the earth. This force, we call a force of gravity or gravitational pull. We feel this when we hold the stone in our hands. The gravitational pull acts vertically downwards and the weight of a body indicates it. All bodies are attract towords the centre of the earth by this gravitational pull and further the body from the centre of the earth, less the pull, less is its weight. A body having a definite mass will have less weight at the top of Mount Everest than that at the sea level. If an astronaut on a flight to the moon carries with him a packet of vitamin pills which weighs one kilogramme on the earth before the flight, the packet, assuming that nothing was removed or consumed out of it, will have no weight when the astronaut reaches a point during the flight where the earth's gravitational pull is so negligible as to be zero. If at that point in the space the packet is weighed by a spring balance, the weight will be zero though the mass of the packet is the same as it was on the earth. As the astronaut approaches the moon, its gravitational pull will case the spring balance to record some weight of the packet of vitamin pills, and on the surface of the moon the weight recorded will be $1/6^{th}$ of the weight on the earth, because the gravitational pull of the moon is $1/6^{th}$ that of the earth. This should explain the difference between the terms weight and mass. The weight of a body is the force of gravity o the body. The important point to note is that mass is constant but weight is variable. The latter depends on the place where the body is situated. If at a height of say, 400km above the earth's surface, an astronaut weighs an article of mass M kg with a spring balance and the gravitational acceleration at the point is g m/s^2, the weight of the article in S.I. units, will be Mg Newtons.

As already stated, a standard piece of platinum-iridium is kept near Paris as a basis for the measurement of mass and its mass is 1 kilogramme. The masses of other bodies are found from comparison with this standard and the comparison is made by weighing. If a body has a mass double that of the standard, the force due to gravity acting on it is double the force acting on the standard. Mass is independent of gravitational pull since any variation in the latter will have an equal effect on the standard mass.

The familiar beam balance compares the masses of two bodies, a known mass in one pan and the mass to be compared in another pan. A spring balance, on the other hand, measures the force in any direction.

The unit of weight in MKS system is kilogramme-force (kgf) and in S.I. units it is Newton (N).

If a moving body suddenly stops or changes its direction, one concludes that something must have caused it; if a stationary body starts moving one concludes that there must be something that caused the motion. This "something" in the above two examples is *Force* which may be regarded as tht which produces motion, destroys it, or changes the direction and speed of the dboy. A force has magnitude, direction and a point of application. The unit of force is the same as that of weight, viz. kgf in MKS units and Newton in S.I. units. On the European continent the symbol kp or kgp (for kilopond) is sometimes written in place of kgf.

One Newton is that force which, when applied to a body having a mass of 1 kilogramme, gives it an acceleration of 1 m per $second^2$.

1 Dyne is that force which, when applied to a body having a mass of 1 gramme, gives it an acceleration of 1 cm/s^2 (1 Dyne = 10^{-5} N). It is a very small unit.

Readers accustomed to MKS units sometimes use the word "kilogramme" without being specific, whether it refers to mass or weight. When we say "a body of 1 kg", the statement is not clear and causes confusion since the mass is expressed in kilogramme and the weight is also expressed in kilogramme, in common usage. Many of you might have observed the weighing machine at railway stations or in some departmental stores which is operated by inserting a 10-paisa coin in a slot on the machine. You stand on its platform, insert a 10-paisa coin and within a second or two, emerges a card, the size of a railway ticket, on which is typed your weight stating, say, 60 kg. Actually the word kg on the card is wrong. It should be kgf. But this indicates how loosely the word kg is often used.

1kgf is the weight of a body having a mass of one kilogramme. The gravitational force on a mass of 1 kilogramme, in Newtons, is the mass of 1kg multiplied by the acceleration due to gravity (in m/s^2).

Thus 1 kgf = 1 kg × 9.81 m/s^2 = 9.81 kg m/s^2 = 9.81 Newtons.

This follows from the basic equation.

Force = Mass × acceleration

When we say that a mine car weighs 1 tonne, it means that the weight of the mine car is 1,000 kgf and expressed in S.I. units the weight is 9810 Newtons. This represents the weight of a mass of 1,000 kilogrammes. Since the word kilogramme-force (kgf) is used to express weight (or force), it is appropriate to use the word "tonne-force" or "tonnefe" to express the weight of a mass of 1,000 kilogrammes but text books rarely use the word tonnefe.

Rigid Body :

A body composed of a very large number of particles whose positions relative to one another do not vary is considered to be a rigid body. A single particle is taken to be a portion of matter whose dimensions are negligible and whose position may therefore be given as that of a mathematical point.

Speed, Velocity and acceleration :

When a body is in motion the rate at which it is moving on its parth is called its speed or velocity. A distinction is usually made between the terms *speed* and *velocity*. While speed is defined as the rate only at which a body is moving, the term velocity implies direction as well as rate. Velocity is therefore a vector quantity but speed is a scalar quantity. A body moves with uniform velocity when throughout the motion equal distances are covered in equal times, however small or however large.

$$\text{Velocity} = \frac{\text{distance moved}}{\text{time taken}}$$

Rest, motion and velocity are relative terms.

Motion of a body always means its motion with reference to other body which may be at rest or in motion. Absolute motion is impossible to conceive. All our motion is with reference to earth which is supposed to be at rest for this concept. When we say, a train is moving at 40 kmph, it means the train's speed is 40 kmph relative to the earth, or the speed with which it appears to move to an observer on the earth.

Consider a train moving at a uniform speed of 30 m/sec. If the engine puts in more power so that the speed of the train changes, at a steady rate to 38 m/sec in four seconds, the train has been accelerated by 2m/sec in each second. The acceleration is written as 2m/sec^2. Acceleration is defined as the rate of increase of velocity. When a body falls freely under the action of earth's gravitational pull, it accelerates uniformly (considering there is no air resistance) and the value of this acceleration is 9.81 m/sec^2. The acceleration due to gravity is written as g. Decceleration is the opposite of acceleration. A car moving at a speed of, say, 90 km/h, i.e. 25 m/s deccelerates and reduces its speed after application of brakes and may eventually stop. Decceleraion is negative acceleration for purposes or calculations.

The acceleration due to gravity varies from place to place on our earth but it varies so little from the sea level to Mt. Everest, the highest point on the earth, that the value is considered practically constant and is taken as 9.81 m/s^2.

A body moving in a straight line has a linear velocity and if it is accelerated, it is subjected to linear acceleration. A body moving in a circular path has an angular velocity and its acceleration is angular acceleration.

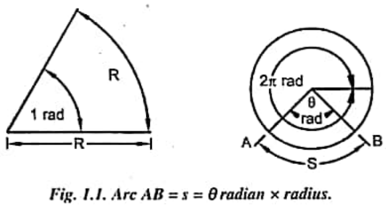

Fig. 1.1. Arc AB = s = θ radian × radius.

If a body moves in a circular path (rotational motion) the angular distance moves is measured in radians and the angular velocity is in rad/sec. The angular acceleration is measured in rad/sec^2.

There are 2π radians in a circle. (Fig. 1.1.) R.P.M. $\times \frac{\pi}{30} = \text{rad/sec.}$

Equations of linear and angular motion :

For linear motion, if

u = velocity of a body at start,

v = velocity of a body after t seconds,

f = acceleration,

s = distance covered within t seconds,

and for angular motion, if

ω_1 = angular velocity at start,

ω_2 = angular velocity after t sec.,

α = angular acceleration

θ = angle turned through in t sec.,

then the equations are as follows :

for linear motion	for angular motion
$v = u + ft$	$\omega_2 = \omega_1 + \alpha t$
$v^2 = u^2 + 2fs$	$\omega^2_2 = \omega_1^2 + 2\alpha\theta$
$s = ut + \frac{1}{2}ft^2$	$\theta = \omega_1 + \frac{1}{2}\alpha t^2$

Consider a wheel revolving at 180 r.p.m. = 3 r.p.s. 1 rev = 2π radians. Therefore angular velocity is 6π rad/sec. Angular velocity of every part of the wheel is the same, but the linear velocities of different parts differ. It is zero at the centre of the shaft and maximum at the periphery of the wheel. In Fig. 1.2 the angular velocity of particle P and Particle Q is the same but the linear velocity of Q is more.

The relation between linear velocity V and angular velocity ω is.

$v = \omega r$, where r is the radius

Example :

A train moving at 36 km/h is uniformly accelerated so that after 8 seconds its velocity is 54 m/s. Find the acceleration in m/s^2 and the distance the train goes in the interval.

Ans. :

u = 36 km/h. *i.e.* 10 m/s.

v = 54 km/h, i.e. 15 m/s.

To find acceleration :

Using $v = u + ft$

Then $15 = 10 - 8f$

$$f = \frac{5}{8} \text{ m/s}^2$$

To find distance :

Using, distance s = average velocity × time

$$s = \frac{10 + 15}{2} \times 8$$

$= 100$ m

Alternatively we could have used either $s = ut + \frac{1}{2}ft^2$

or $v^2 - u^2 = 2fs$

Example :

An electric motor starting from rest reaches a speed 2100 r.p.m. in 4 minutes. What is the angular acceleration in rad/s^2 and how many revolutions are turned in the last two of the four minutes.

Ans. :

$$\textit{Angular acceleration} = \frac{\textit{Increase in angular velocity}}{\textit{Time taken}}$$

Initial angular velocity = 0

final angular velocity = 2100 r.p.m.

$$= \frac{2100 \times 2\pi}{60} \text{ rad/s}$$

$= 219.9$ rad/s

$$\therefore \text{ angular acceleration } = \frac{219.9 - 0}{4 \times 60} \text{ rad/s}^2$$

$= 0.916$ rad/s²

Since, $w_1 = 0$, the relationship reduces to $\theta = \frac{1}{2} \propto t^2$

Angle turned through in the last two minutes can be calculated if we subtract from the total angle turned the angular travel in the first two minutes.

Let total angle turned through in 4 minutes be θ_4.

$$\therefore \ \theta = \frac{1}{2} \times 0.916 \times (4 \times 60)^2$$

$$= \frac{0.916 \times 240 \times 240}{2}$$

$= 26380.8$ radians

Let angle turned through in first two minutes = θ_2

$$\therefore \ \theta_2 = \frac{1}{2} \times (0.916) \times (2 \times 60)^2 \text{ radians}$$

$= 0.916 \times 120 \times 60$ radians

$= 6595.2$ radians

∴ Angle turned through in the last two of the 4 minutes.

$= 26380.8 - 6595.2$

$= 19785.6$ radians

$$= \frac{19785.6}{2\pi} \text{ revolutions} = 3148 \text{ rev.}$$

= 3149 revolutions.

Example :

A lift is connected to a rope which winds on to a drum of a radius 440mm. The lift accelerates from rest at the rate of 0.33 m/s^2 for 10s and then maintains constant velocity. Calculate the corresponding angular acceleration and angular velocity of the drum in units of radians per second.

Ans. :

Drum radius = 440mm = 0.44m

Acceleration of the lift = 0.33 m/s²

Let angular acceleration of the drum be α rad/s²

$\therefore \quad \propto \times$ radius = 0.33 m/s²

or $\propto = \dfrac{0.33}{0.44}$ rad / s² = 0.75 rad / s²

Now $\quad \omega_2 = \omega_1 + \propto t$

$\omega_1 = 0$ since it starts from rest

$\therefore \quad \omega_2 = \mu\, t$ where ω_2 = final angular velocity

$= 0.75 \times 10$ rad/s

$= 7.5$ rad/s.

Newton's laws of motion :

First Law : Every body continues to be in a state of rest of uniform motion in a straight line unless acted upon by an external applied force.

Second Law : Rate of change of momentum is proportional to the force producing the change and takes place in the direction in which the force acts.

Third Law : To every action there is always an equal and opposite reaction.

While the first law defines force, the second law provides us with the means for measuring force as we shall see later.

The third law can be easily understood from one or two simple examples.

A book of weight, say, 10 N rests on a level table surface. The book exerts a force of 10 N, acting vertically downwards on the surface of the table. The table opposes this force with equal and opposite reaction normal (*i.e.* at right angles) to it. The result of the vertically downward weight of the book and the normal reaction of the table is that book and the table surfaces are in equilibrium and there is no motion.

The significance of third law will be clear to any one who has tried to fire a gun. As the gun held against the shoulder is fired the momentum with which the bullet leaves the gun results in a recoil against the shoulder. An interesting and significant development based on this law is the science of rocket engines. In an aircraft equipped with rocket engines the reaction is exerted against a jet generated within the engine. The atmosphere plays no part at all in the propulsion of a rocket vehicle; on the other hand rockets work better in the vacuum of outer space.

Inertia :

Some bodies can be accelerated or retarded much more easily than others; e.g. a cricket ball can be thrown and caught much more easily than a ball of iron of the same size. This reluctance of sluggishness to motion or change of motion is referred to as *inertia* and is largely dependent on the mass of the body. Because of inertia a body will not move by itself it is at rest, no change its motion in a straight line unless compelled by external agency to do so. A body in motion will not stop or reduce its motion in quantity or direction because of inertia, unless compelled by an external force. (Aslo see under moment of inertia and radius of gyration).

Momentum :

This is the ability which a body possesses by reason of its motion to overcome resistance. It is the combined effect of inertia and velocity. The momentum of a body is meausred by the product of its mass and its velocity.

Momentum – M × v where, M = Mass

v = velocity

Rate of change of momentum = Mass × rate of change of velocity, assuing mass to be constant.

= Mass × f, where f is acceleration.

Momentum, being a product of a scalar quantity and a vector quantity, is a vector quantity and can, therefore, be treated in the same manner as other vector quantities. The basic unit of momentum is N – m/sin S.I. system.

According to Newton's second law of motion rate of change of momentum is proportional to applied force, so that

Force (P) $\propto$ rate of change of momentum

$\propto M \times f$

$= K \times M \times f$ where K is a constant.

From this relationship it will be observed that if we choose unit of weight (or force) such that when M is unity and f is unity, the value of K is one so that we get the relationship.

$P = Mf$

In S.I. units which are coherent

Force on a body = Mass × acceleration

(in Newtons) (in kg) (in m/s^2)

Force of gravity on 1 kg

i.e. 1 kgf = 1 × g Newtons = g Newtons.

Example :

A train of mass 500 tonnes, moving at 72 km/h, is stopped by application of brakes within 100 m. What is the minimum force applied by the brakes ?

Ans. :

72 km/h = 20 m/s

Since $v^2 - u^2 = 2\,fs$

$0 - 20^2 = 2 \times 100 \times f$

$$f = \frac{-400}{200} = -2\,m/s^2$$

The minus sign indicates that it is retardation.

500 te = 500,000 kg

$= 5 \times 10^5$ kg

P, Retarding force needed = M × f Newtons

$= 5 \times 10^5 \times 2$ N

$= 10^3$ kN.

Example :

An electric train weighs 800 metric tonnes force. Resistance to motion can be considered constant at 1% of the weight. The motors can provide tractive force of 20,000 kgf. How long does it take to accelerate the train to a speed of 96 km/h on level track ?

Ans. :

In an example of this nature, first convert all units into S.I. units

$$v \text{ is } 96 \text{ km/h} = 96 \times \frac{5}{18} = \frac{80}{3}\ m/s$$

Tractive force = 20,000 kgf

800 meteric tonnes force = 800 × 1,000 kgf

= 800,000 kgf

Resisting force = 1% of 800,000 kgf

= 0.01 × 800,000 kgf

= 8,000 kgf

Tractive force developed by the motors of the electric train has to perform the following functions on a level track.

(a) to overcome the force resisting motion.

(b) to cause acceleration of the entire train.

Force available for acceleration = 20,000 kgf − 8,000 kgf

= 12,000 kgf

= 12,000 × 9.81 N

A weight of 800 metric tonnes force has a mass of 800 tonnes, *i.e.* 800,000 kg.

$$P = Mf\,;\; f = \frac{P}{M} = \frac{12000 \times 9.81}{800,000} = 0.147\,15\ m/s^2$$

Using $v = u + ft$

For a body starting from rest $v = ft$

$$\therefore\quad t = \frac{v}{f} = \frac{80}{3} \times \frac{1}{0.14715}\,\frac{m}{s} \times \frac{s^2}{m} = 181.2 \text{ seconds}$$

Ans. : Time to accelerate is 181 seconds.

In example of this type it should be remembered that a body moving against a frictional resistance will move with uniform velocity so long as the applied force P is equal to the firctional resistance F. If P is greater than F the body will accelerate and the force causing acceleration is

P − F = Mf (M is mass ; f is acceleration).

Centrifugal and Centripetal force :

Tie a pebble or a small stone at the end of a string and holding the free end of the string between two fingers, whirl the pebble in a circular motion. The string, it will be observed, remains taut indicating that there is some tension in the string which keeps it tigh. This tension acts away from the centre of rotation, *i.e.* the fingers, and is known as the centrifugal force. It is a radially outward force acting on the body (in this case, on the pebble). The centrifugal force is opposed by another equal and opposite force according to Newton's third law of motion; it is called centripetal force (centripetal in Greek means centre seeking) which acts towards the centre of rotation. The fingers exert the pull and provide the centripetal force necessary to keep the pebble in a circular orbit. If the string is released, the pebble will not move along the circular orbit because the moment the string is released, the centripetal force on the body, i.e., the pebble, ceases to act.

In order to avoid effects of centrifugal force it is necessary to balance rotating parts. The rotor of an alternator, for instance, weighing many tonnes and revolving at about 1500 r.p.m. is subject to the action of a centrifugal force and it must be accurately balanced if large bending stresses on the shaft are to be avoided. In a flywheel the centrifugal force creates tension in the spokes. In the case of a train going round a curve the centrifugal force causes a pressure on the outer rail and this pressure is transmitted by the flanges of the wheels to the rail.

If a body of mass M moves in a circular path at an angular velocity of w and its distance from the centre of rotation is r, the linear velocity of the body at any instant is v = w r, and the centripetal acceleration is $\frac{v^2}{r}$.

The force that causes centripetal acceleration is

$$\text{Force} = \text{mass} \times \text{acceleration} = M \times \frac{v^2}{r}$$

Example :

A ball weighing 2 kgf is whirled in a horizontal circle at the end of a rope 1.5 m long, at 90 r.p.m. Find the force in the rope.

Ans. :

90 r.p.m. = 1.5 r.p. $\omega = 1.5 \times 2\pi = 9.42$ rad/s.

v_1 linear velocity $\omega \times r = w \times 1.5\text{ m} = 14.13$ m/s.

$$\text{Centripetal acceleration} = \frac{v^2}{r} = 133.1\ m/s^2$$

Weight of the ball is 2 kgf, i.e. weight of a mass of 2 kg.

Force = mass × acceleration = 2 × 133.1 = 266.2 N

Example :

A motor car moving at a speed of 36 km/hr weighs 1200 kgf. If it has to pass along a curve of radius 40m, what is the centrifugal force actingon the car ?

Ans. :

$$\text{Centrifugal force} = \frac{Mv^2}{r}$$

v is 36 km/hr = 10 m/s, and mass is 1200 kg.

$$\text{Centrifugal force} = \frac{1200 \times 10 \times 10}{40} = 3000\text{ N}$$

Banking of the curve : When an automobile, cycle or a vehicle travels along a curve of a road, it has a tendency to ove away from the road due to centrifugal force. But this tendency is opposed by the frictional force between the tyres and the road surface. In such a case, the tyres will soon wear out. The wear can be considerably reduced by bringing into play a cetripetal force to counter the centrifugal force on the automobile. This is achieved in practice by giving to the road surface, at its curved portion, an inclination sloping upwards towards the outer edge (circumference), because the horizontal component of the normal reaction between the tyres and road forms the desired centripetal force. This system of providing an inclinaton of the road on the curve is called *banking of the curve.*

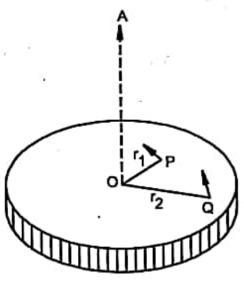

Fig. 1.2

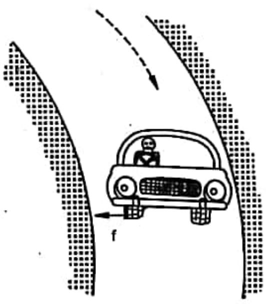

Fig. 1.2 A. Banking of a curve.

Equilibrium :

A body at rest is said to be in static equilibrium when the forces acting on it keep it steady and the system of forces is said to be in equilibrium.

You must have observed the game of tug of war in which two teams, each of 9 – 11 players, pull a rope in opposite directions. The total force of each team acts on the rope which is therefore subjected to two forces in opposite directions. When the two teams exert equal force on the rope, it does not move but remains steady. At that moment the rope is said to be in equilibrium and the two forces acting on the rope are said to be in equilibrium. The forces acting on the rope trying to stretch it, are tensile forces and the rope is in tension.

A body cannot be in equilibrium under the action of a single force, and if two forces act on a body it can only be in equilibrium if the forces are of equal magnitude but act in opposite directions along the same line of action.

Tension in a rope :

Consider the force in a rope supporting a mass of 1kg. If the mass and rope were still, the downward force on the mass due to gravity would be.

Force = Mg = 1 × 9.8 N = 9.8 N which would be the tension in the rope. (Fig. 1.3).

If the rope were raising the weight at a constant speed, the tension in it would be 9.8 N. If however, the mass is being accelerated upwards at 2 m/s², the tension in the rope has to provide two forces at once.

i. a force of 9.8 N to overcome force of gravity.

ii. a force of M f = 1 kg × 2m/s² = 2N to provide the acceleration against the inertia of mass.

Hence the total tension in the rope

= 11.8 N.

If the weight is accelerating downward at 2m/s², the tension in the rope would be (9.8 N – 2N) because part of the gravitional force (2 N) would be employed in accelerating the mass, leaving the remaining 7.8 N counteracted by the rope.

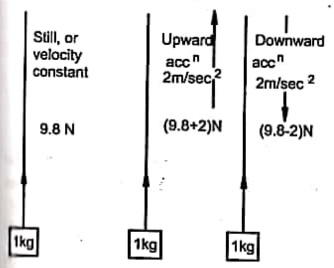

Fig. 1.3

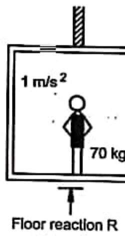

Fig. 1.3 (a)

Example :

A person of 70 kg stands in a lift which moves with an acceleration of 1m/s². What will be the pressure on the floor of the lift (a) if it is ascending, and (b) if it is descending.

Ans. :

The forces acting are shown in Fig. 1.3 (a)

(a) Upward reaction R = Force to support load + force to accelerate.

= W + P

Now $W = Mg = 70 \times 9.81 = 686.7$ N

$P = MF = 70 \times 1 = 70$ N

R, pressure on floor = W + P 686.7 + 70 = 756.7 N.

(b) R = Force to support load – force to accelerate

$= W - P = 686.7 - 70 = 616.7$N.

This example shows why a man feels himself heavy when the cage is accelerating upwards.

Example :

Ttwo masses, each 4 kg, are suspended from the ends of a cord passing over a lightt frictionless pulley. A mass of 0.5 kg is added to one of the masses. *Determine :*

(a) the resulting acceleration of the system,

(b) the tension in the cord.

Ans. :

A diagram is given in Fig. 1.4.

Method 1 :

When the additional mass is added to one side equilibrium will be destroyed and the 4.5 kg mass will accelerate downwards while the 4 kg mass will have an equal acceleration upwards.

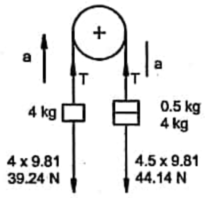

Fig. 1.4

Let the tension in the cord be T (Newtons) and the acceleration of the system be a (m/s²).

For 4.5 kg mass;

Gravitational force on mass $= 4.5 \times 9.81$

$= 44.14$ N

∴ Force producing acceleration = (44.14 – T) N

Since $P = M \times a$

$(44.14 - T) = 4.5\,a$ --- (1)

For 4 kg mass

Gravitional force on mass $= 4 \times 9.81$

≈ 39.24N

∴ Force producing acceleration = (T – 39.24) N

Thus,

$T - 39.24 = 4a$ --- (2)

Adding (1) and (2)

$4.9 = 8.5\,a$

∴ $a = 0.577$ m/s²

From equation (2) $T = 39.24 + 4 \times 0.577$

$= 41.6$ N

The acceleration is 0.577 m/s² and the tension in the cord is 41.6 N.

Method 2 :

The mass which causes the system to accelerate is 0.5 kg.

Thus, Accelerating force $= 0.5 \times 9.81 = 4.9$ N

The total mass which is accelerating

$\approx 4 + 4.5 = 8.5$ kg.

Since $= P = M \times a$

$4.9 = 8.5\,a$

∴ $a = 0.577$ m/s²

The tension in the cord is then obtained by considering the motion of one of the masses, as in Method 1.

The difference beteween the two methods should be noted. Method 1 deals with the motion of two separate masses which move as a system, whereas Method 2 considers the motion of the complete system.

Example :

A trolley (assumed frictionless) has a mass of 24.5 kg and it is accelerated by the force exerted by a hanging weight of mass 0.5 kg. Calculate its acceleration. (g = 9.8 m/s²).

Ans. :

Total mass moved = (24.5 kg + 0.5 kg)

= 25 kg.

Force causing acceleration = (0.5 × 9.8) N

= 4.9 N

Since $P = Ma$, $a = \dfrac{P}{M} = \dfrac{4.9}{25} = 0.196\ \text{m/s}^2$

∴ acceleration is 0.196 m/s².

Vector and scalar quantities :

Any quantity that has magnitude and direction is called a vector. A force has magnitude and direction; it is therefore a vector quantity. Velocity is speed in a given direction so that it has magnitude and, unlike speed, direction. Velocity, like force is, therefore, a vector quantity. Acceleration, like velocity is also a vector quantity, having both magnitude and direction.

A quantity that does not include a direction is called a scalar quantity. An area (say, $8m^2$), an amount of work done (say, 50 Joules), a speed (say, 20 m/s), mass, are all scalar quantities.

A vector quantity can be represented by a straight line, the length of the line being proportional to the magnitude of the vector quantity. The direction of the vector quantity can be indicated by the direction of the line relative to some basic direction, such as a pair of co-ordinate axes. A force F of say, 10 kgf, can be represented by a line AB, drawn to scale, and is read as "vector AB" if the arrow indicates the direction in which the force acts. If a vector is represented by the line BA it is used as "vector BA" meaning the force acts in the direction from B to A. It is written as $\vec{BA}$ and $\vec{AB} = - \vec{BA}$

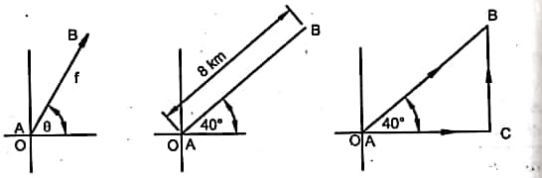

Fig. 1.5

Another convention is to write the vector as $F \angle \theta$ (omiting the letters A and B) which represents a vector of length F whose direction is indicated by angle θ referred to the four-quadrant method of designation angles. In the four-quadrant method of designating angles, an easterly direction is 0°, north is 90°, west is 180° and south is 270°. A travel of 8km from a point of origin 0 in the direction 40° north of east from A to B, as shown in Fig. 1.5, is represented by 8km $\angle 40°$. The travel from B to A can be represented by 8 km $\angle 220°$. The standard convention in $F \angle \theta$ presentation is that F is positive and measured outward from the origin so that negative values of F are nott used. A velocity of 10 m/s acting horizontally due west can be represented by 10 m/s $\angle 180°$.

The distance AB can be covered by travelling the distance from the point of origin A in the direction as indicated by the angle; alternatively the point B can be reached by travelling Easterly to point C and northerly from C to B. This is represented mathematically in the following manner.

$$\vec{AB} = \vec{AC} + \vec{CB}$$

$\vec{AC}$ and $\vec{CB}$ are the vector components of $\vec{AB}$ and $\vec{AB}$ is referred to as the resultant vector of vectors $\vec{AC}$ and $\vec{CB}$.

Vector addition illustrated in Fig. 1.6 (a) can therefore be represented matematically as follows :

(a) $\vec{OP} = \vec{OQ} + \vec{QP}$

(b) $\vec{OP} = \vec{OR} + \vec{RP}$

(c) $\vec{OP} = \vec{OS} + \vec{ST} + \vec{TU} + \vec{UP}$

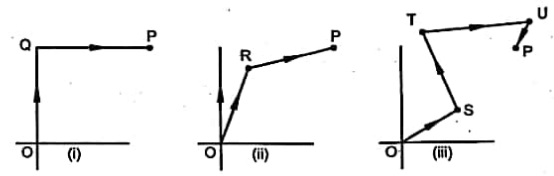

Fig. 1.6 (a)

Relative Velocity :

The relative velocity of a body A in a relation to another body B is the velocity with which B finds A moving (or an observer standing on B, finds A moving). Since the velocity is a vector quantity, the relative velocity will be the vector difference of the velocities of A and B.

If two bodies A and B are moving with the velocities Va and Vb respectively (Fig. 1.6 b) then the relative velocity of one to the other is the vector difference of Va and Vb, i.e., if vectors Oa and Ob, representing Va and Vb in magnitude, direction and sense, are drawn from the same point O, then ab represents the velocity of B relative to A and ba, the velocity of A relative to B.

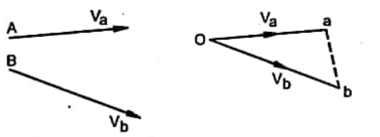

Fig. 1.6 (b)

Coplaner forces :

When a body is acted upon by a number of forces in the same plane (i.e. co-planer forces), the resultant force for this system of forces is a single force which will replace all the other forces but still have the same effect on the body.

Parallelogram of forces :

If a body is acted upon by 2 coplaner forces, P and q, their resultant is obtained by adding the forces vectorially. This is done by drawing a parallelogram with p and q as adjacent sides representing the magnitude and direction and the diagonal represents the vector of the resultant force, thereby giving the magnitude and direction of tthe resultant force. Fig. 1.7 (i).

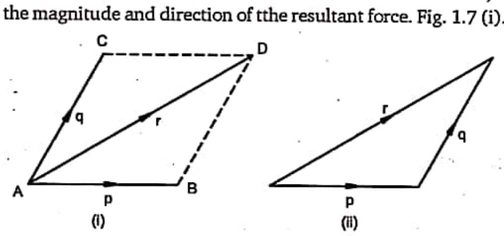

Fig. 1.7

The net effect of the vector paths p and q is the same as that for the resultant vector path. Fig. 1.7 (ii) is a vector diagram.

Resultant of two coplaner forces, not at right angles.

Forces P and Q acting in direction whic include in angle α are equivalent to a single force R where

$$R^2 = P^2 + Q^2 + 2PQ \cos \propto \text{ .... Eqn. 1}$$

in a direction making an angle θ with the direction of P where

$$\tan \theta = \frac{Q \sin \propto}{P + Q \cos \propto} \text{ .... Eqn. 2.}$$

Example :

Forces of 7 kgf and 5 kgf act on a body, and angle between them is 55°. Find their resultant and the angle between it and the two forces.

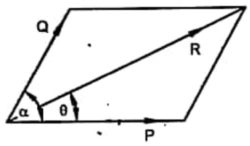

Fig. 1.8

Ans. :

Required to Find R and θ (Fig.) 1.8

1. *T find R.*

Using Eqn. 1 and substituting the given values of P, Q, and μ..

$$R^2 = 7^2 + 5^2 + 2 \times 5 \times 7 \cos 55^\circ$$

$$= 74 + 70 \times 0.5736 = 114.15$$

$$\therefore \; R = \sqrt{114.15} = 10.7 \text{ kgf approx.}$$

2. *To find θ*

Using the formula

$$\operatorname{Tan} \theta = \frac{Q \sin \alpha}{P + Q \cos \alpha}$$

Here P = 7 kgf

We have on substitution

$$\operatorname{Tan} \theta = \frac{5 \sin 55^\circ}{7 + 5 \cos 55^\circ} = \frac{5 \times 0.8192}{7 + 5 \times 0.5736}$$

$$= \frac{4.0960}{9.8680}$$

$$\therefore \; \theta = 22^\circ \, 32$$

If two coplaner forces P and Q are at right angles to each other their resultant force R is given by the relation.

$$R = \sqrt{P^2 + Q^2} \qquad \text{(From Equation 1)}$$

as cosine of the included angle 90° is zero. The angel θ between the resultant force and the force P is the such that

$$\tan\theta = \frac{Q}{P} \qquad \text{(From Equation 2.)}$$

A single force can be considered to consist of 2 coplaner component forces at right angles to each other. These components can be found for any force by drawing the parallelogram of forces using the known force as the diagonal. the parallellogram drawn to obtain the two forces is a rectangle and therefore the two component forces are called the *rectangular components.*

A force of 100 N (Fig. 1.9) acting in a direction 30°N of East is equivalent to 2 forces (rectangular components).

100 cos 30° = 86.6 N due east, say H

100 sin 30° = 50 N due north, say V

If 100 N is represented by a vector $|\underline{\theta}$.

then $R^2 = H^2 + V^2$

and $q = \theta = \arctan \dfrac{V}{H}$

$$= \tan^{-1}\frac{V}{H}$$

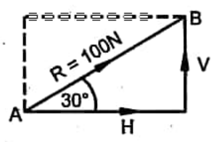

Fig. 1.9

This manner of writing $\tan^{-1}$ is now regarded as non-standard.

Just as it is convenient to indicate a resultant vector mathematically as $|\underline{\theta}$ so that $\theta = \arctan\dfrac{V}{H}$, another convenient way of expressing components of a vector mathematically is the use of column vector. The column vector for R $|\underline{\theta}$ is

$$\begin{bmatrix} R\cos\theta \\ R\sin\theta \end{bmatrix}$$

R cos θ represents the horizontal (or x-axis) component H, while R sin θ represents the vertical (or y-axis) component V.

For example, 10 km $|\underline{36°\,52'}$ can be represented by

$$\begin{vmatrix} 10\text{ km}\cos 36°\,52' \\ 10\text{ km}\sin 36°\,52' \end{vmatrix}$$

i.e. by $\begin{vmatrix} 8\text{ km} \\ 6\text{ km} \end{vmatrix}$

Example : Express, as column vectors :

(a) 5 N $|\underline{53°\,8'}$ (b) 10 m/s $|\underline{90°}$

Ans. :

Let $\begin{vmatrix} H \\ V \end{vmatrix}$ be the column vectors of R $|\underline{\theta}$. Then H = R cos θ

and V = R sin θ.

(a) H = 5 N cos 53° 8' = 5 N (0.6) = 3N

(b) V = 5 N sin 53° 8 = 5 N (0.8) = 4 N

$$5\text{ N}\,|\underline{53°\,8'} = \begin{vmatrix} 3N \\ 4N \end{vmatrix}$$

(b) H = 10 m/s cos 90° = 10 m/s (0) = 0

V = 10 m/s sin 90° = 10 m/s (1) = 10 m/s

$$10\text{ m/s}\,|\underline{90°} = \begin{vmatrix} 0 \\ 10\text{m/s} \end{vmatrix}$$

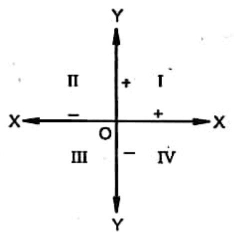

Fig. 1.10

If two or more coplaner forces act on a body and intersect at a point 0, they can be resolved so that each force has a component along the X axis and Y axis. There are four quadrants in a plane and the measurements of resolved parts along OX and OY are positive; resolved parts along the OX' and OY' are negative. (Fig. 1.10).

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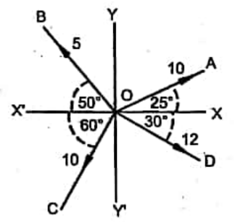
Example :

Find the resultant of forces as shown in Fig. 1.11



Viz. 10 N acting along OA
5 N acting along OB
10 N acting along OC
12 N acting along OD,

the angles made by the lines of action of the forces with XOX' and YOY' being as shown.

Fig. 1.11

Ans. :

Careful arrangement of the resolved parts is important, and some such tabulation as the following is suggested :

Components along XOX'

	Positive	Value	Negative	Value
Quad. I	10 cos 25°	9.063		
Quad. II			5 cos 50°	3.214
Quad. III			10 cos 60°	5.000
Quad. IV	12 cos 30°	10.39		
Total	+	19.45	–	8.214
	Sum = 11.24 N (approx)			

Components along YOY'

	Positive	Value	Negative	Value
Quad. I	10 sin 25°	4.226		
Quad. II	5 sin 50°	3.830		
Quad. III			10 sin 60°	8.660
Quad. IV	12 cos 30°		12 sin 30°	6.000
Total	+	8.056	–	14.660
	Sum = – 6.604 (approx)			

The negative sign shows that the total resolved part acts along OY'.

The forces now reduce to two, as shown in Fig. 1.12. Completing the rectangle, R represents the resultant of the whole system.

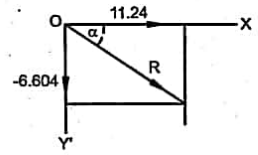


Fig. 1.12

To Find R :

$$R = \sqrt{(11.24)^2 + (-6.604)^2}$$

$$= \sqrt{170}$$

$$= 13 \text{ N nearly.}$$

Let ∝ be the angle made by the resultant with OX.

Then, taking numerical values only.

$$\tan \propto = \frac{6.604}{11.24} = 0.589 \text{ approx}$$

So, $\alpha = 30°31'$

The negative sign, if retained, would show that the angle is in the 4th quadrant.

Triangle of forces :

The theorem of the triangle of forces states that when three co-planner forces acting at a point are represented in magnitude and direction (but not position) by the sides of a traingle *taken in order* the three co-planer forces are in equilibrium.

It follows from this : If there non-parallel co-planer forces are in equilibrium they must meet at a point. It should be noted that the directions in which the forces act, as indicated by the arrowheads on the sides of the triangle, follow the same way round in order; all will be clockwise or all will be anticlockwise.

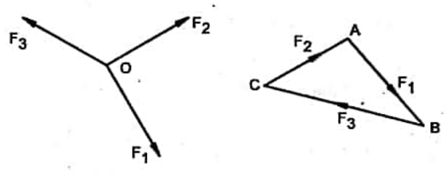


Fig. 1.13

Therefore, if we have three forces acting at a point, the equilibrium of these forces can be tested by seeing if, when they are represented in magnitude and direction, they make the 3 sides of a traingle. If this occurs then the forces will be equilibrium. Tthe triangle then drawn is called the triangle of forces. It follows from the triangle of forces tha when three forces acting at a point can be represented in magnitude and direction by the sides of a triangle taken in order, any one of the forces represents eiher the resultant or equilibrium of the other two, according to the direction in which it acts.

The equilibrant of a system of concurrent co-planar forces is that force which mus be added to the system to produce equilibrium.

Concurrency of forces :

When 3 co-planar forces i.e. forces n he same plane acting on a body, are equilibrium their lines of action intersect at one point only. these forces are then saide to be concurrent. If this is not so, two possibilities arise :

i. Two of the three forces meet at a point and have the same effect as their resultant. The third force, and "resultant" of the aforesaid two, will cause a motion of translation and the body will not be static and will not be equilibrium.

ii. The resultant of two forces which may meet at a point, and the third force may have the effect of a couple on the body which will then undergo a turning or twisting motion. In this cae also the body will not be static and will not be in equilibrium.

Lami's theorem :

If three forces acting at a point are in equilibrium each is proportional to the sine of the angle included between the lines of action of the other two.

Example :

A weight of 200 kgf suspsended by two fine light wires, 5m and 6m in length, are fastened to two points A and B, which are 8m apart on a horizontal beam (Fig. 1.14). The wires are knotted at C. The weight is also fastened on at C. Find the tension in the wires AC and CB.

Ans. :

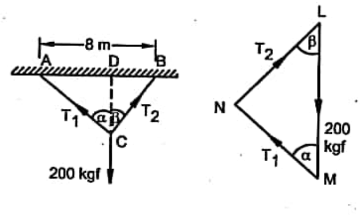
Fig. 1.14 represents the diagram of the arrangement of the forces according to the data.

CD, the perpendicular from C on AB, is the continuation of the line of action of the weight of 200 kgf. T_1 and T_2 are the tensions in CA and CB respectively.

1. Graphical Solution :

Construct the triangle of forces for the forces T_1, T_2 and the weight of 200 kgf.

Draw LM to represent the 200 kgf, on a suitable scale. From M draw a straight lines parallel to CA (force T_1). From L draw a straight line parallel to BC (force T_2). Let N be the point of intersection of these straight lines.



The Δ LMN is the triangle of forces and the sides are proportional to the three forces.

∴ on the same scale that LM represents 200 kgf, MN will represent the tension T_1 and NL will represenet the tension T_2.

Fig. 1.14

2. Trigonometrical Solution :

We must solve the Δ LMN, knowing that LM represents 200 kgf. Then we can calculate the lengths of LN and MN.

From the properties of parallel lines the angles α and β as shown in the triangle of forces are equal to α and β as shown in the space diagram.

To find α and β we must first solve Δ ABC, knowing all the sides.

Applying the cosine rule.

$$\text{Cos A} = \frac{8^2 + 6^2 - 5^2}{2 \times 8 \times 6}$$

$$\therefore \quad \angle A = 38°\ 37' \text{ and } \alpha = 90° - A = 51°\ 23'$$

$$\text{Similarly Cos B} = \frac{8^2 + 5^2 - 6^2}{2 \times 8 \times 5}$$

$$\therefore \quad \angle B = 48°\ 30' \text{ and } \beta = 90° - b = 41°\ 30'$$

$$\angle LMN = 180° - (\alpha + \beta) = 87°\ 7'$$

To find sides of Δ LMN;

Using sine rule

$$\frac{LN}{200} = \frac{\sin 51^\circ 23'}{\sin 87^\circ 7'}$$

$\therefore$ LN = 156

$$\frac{MN}{200} = \frac{\sin 41^\circ 30'}{\sin 87^\circ 7'}$$

$\therefore$ MN = 133

Thus T_2 = 156 kgf

T_1 = 133 kgf.

Similarly $\cos B = \frac{8^2 + 5^2 - 6^2}{2 \times 8 \times 5}$

Bow's notation :

Graphical solution of problems involving triangle of forces or polygon of forces are simplified by use of Bow's notation, which is named after R. H. Bow who introduced it in 1873. In Graphic Statics, this notation is universally employed because in complicated stress diagrams, it becomes extremely easy to locate a particular force in magnitude, direction and position when the notation is adopted. Bow's notation is illustrated by reference to Fig. 1.15.

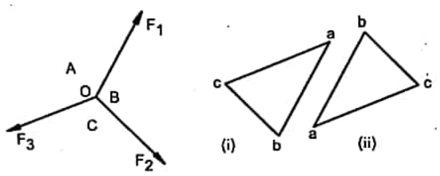


Fig. 1.15

Left – *Space diagram or position diagram or configuration diagram.*

Right– *Vector diagram or force diagram*

(i) With anticlockwise order starting with a

(ii) With clockwise order starting with a

Let us say 3 forces, F_1, F_2 and F_3 act on a body at point O and keep it n equilibrium. Draw a diagram in which each line, starting with O, is parallel to the direction of force. Write capital letters A, B, C, in a definite order (clockwise or anticlockwise) in the space on either side of the line of action of each force.Such diagram is called *space diagram, configuration diagram or position diagram.* A space diagram indicates the direction and relative position of the forces but thelength of lines is not proportional to the magnitude of the force. Draw by its side another diagram of vectors in which.

i. vector ends are given small letters a, b, c.

ii. tip of one vector joins the tail of another in a definite order (say, clockwise, as in the Fig. 1.15 (ii)

The vector ab is then parallel to force F_1 (between space letters A and B); vector be is parallel to force F_2 and vector can is parallel to force F_3. The diagram of vecors is called vector diagram or force diagram.

The merit of this method of notation is that while capital letters indiate the line of action of a force in the space diagram, the corresponding small letters in the vector diagram give the magnitude and direction. Two force diagram can be drawn for each space diagram, the one being the exact reverse of the other. In Fig. 1.15 force diagram (i) is obtained by considering the forces of the space diagram in an anticlockwise direction and force diagram (ii) is drawn by taking the forces in a clockwise direction. In either of the figures the resultant of $\vec{ab}$ and $\vec{bc}$ has the same magnitude as vector $\vec{ca}$ but the direction is opposite to that of the vector $\vec{ca}$

$$\vec{ab} + \vec{bc} = \vec{ac}$$

Force Polygon :

If more than two coplaner forces act at a point it is easy to obtain their resultant by adding the vectors as indicated above. If more than 3 coplaner forces act at a point and are in equilibrium they may be represented in magnitude and direction by the side of a polygon taken in order. If a system of forces meets at a point and that system is not in equilibrium the polygon does not close, and the force needed to bring the system into equilibrium (called equilibrant) is represented by the vector which joints the open ends of the incomplete polygon.

Draw to a suitable scale, the vectors ab, bc to represent the forces F_1 and F_2 in magnitude and direction. The vector ac represents the resultant of F_1 and F_2 (Fig. 1.16).

$\vec{ab} + \vec{bc} = \vec{ac}$ (resultant)

Now draw the vector $\vec{cd}$ to represent F_3 in magnitude and direction.

Then $\vec{ad} = \vec{ac} + \vec{cd} = \vec{ab} + \vec{bc} + \vec{cd}$, i.e. adrepresents the resultant of F_3 and F_1 and F_2. Similarly, draw the vector $\vec{de}$ to represent F_4 to scale. Join ae. Then the vector

$$\vec{ae} = \vec{ad} + \vec{de}$$

$$\vec{ac} = \vec{cd} + \vec{de}$$

$$\vec{ab} = \vec{bc} + \vec{cd} + \vec{de}$$

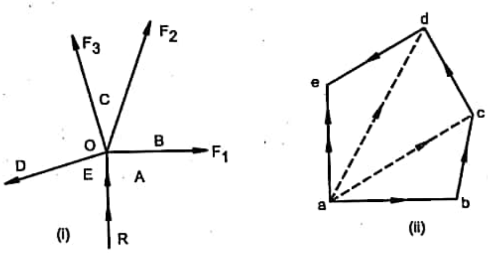


Fig. 1.16

The vector $\vec{ae}$ represents therefore, in magnitude and direction resultant R of the coplaner forces F_1, F_2, F_3, and F_4 acting at the point O. To obtain its position or line of action. through the point O in Fig. 1.16, draw a line parallel to ae, with letters A and E on either side of it. We have thus obtained the resultant R complete in magnitude, direction and position.

It should be clear from the foregoing that to obtain the resultant of a system of coplaner forces acting at a point, it is necessary to draw the vectors ab, bc, cd, etc. to represent the forces to a suitable scale, in magnitude and direction. The ling joining the first point a to the last point e in the vector diagram represents the resultant in magnitude and direction. This polygon of vectors is known as the vector or Force Polygon.

If a set of coplaner force F_1, F_2, F_3, F_4 acting at a point O is in equilibrium, their resultant R must be zero. Graphically, since the resultant is zero, in the force or vector polygon, the last point e must fall on the first point a so that the vector ae is zero, *i.e.* the force polygon must close.

An example of application of space diagram and vector diagram to find out the stresses in the bottom of a crane and its supporting members is illustrated by Fig. 1.17. The rope of the crane raises a weight, W (say 1 kN). The rope coils round a winch in the crane indicated by "effort". The boom is hinged at its base to the frame of the crane and the upper end is held in position by a member attached to the structure of the crane. The relative positions of ropes, boom and other supporting members are as shown in the figure.

To find out the forces in different members of the crane draw a space diagram as shown in the figure and mark on it the known forces W, (1 kN) and effort (1 kN) showint their directions by arrows. Draw a vector diagram starting with point a, and draw ab, bc. From a a draw a line ad parallel to force Q and from c another line cd paralle to the force P. These parallelines must meet at point to give a closed figure as the system of forces is in equilibrium. The magnitude of the forces Q and P can then be measured. All the arrows of the forces in the vector/force diagram should follow a definite order (all clockwise or all anticlockwise). The force P in the boom has, therefore to be in the direction cd and the force Q in the tie, in the direction da.

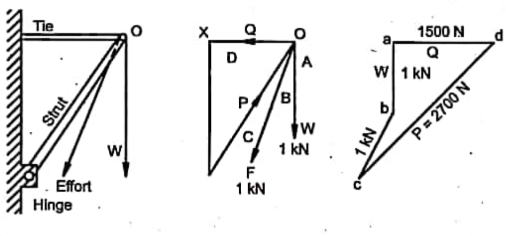


Fig. 1.17

The component of a machine or structure which is in compression is called a ***strut***. If a strut is broken whatever it supports tends to collapse inwards into the strut. The component of a machine or structure which is in tension is called a ***tie***. Whatever a tie supports tends to fall away from it if the tie is broken.

Moment of a force :

The moment of a force is defined as the product of the force multiplied by the perpendicular distance of the force from the point about which it is acting.

In Fig. 1.18 the moment of the force about the point O is F × OA where OA is the perpendicular distance of the line of action of force F from the point O. The moment of a force may be clockwise or anticlockwise as shown in the figure and it has a turning effect around the point O. The turning effect of a force is usually called the *turning moment*. Other names given to turning moment are *moment, torque and twisting moment.*

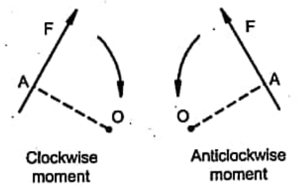


Fig. 1.18 Moment of force

The point about which the force turns is called a pivot or a fulcurm. If a force acts through the point of rotation it has no turning effect or moment. **Principle of moment** states that : If a body is at rest under the action of several forces in the same plane, then the clockwise moments about any point must be equal to the anticlockwise moments about the same point; or the resultant moment on the body about any point is zero.

Couple :

Two parallel forces, of equal magnitude, but which act in opposite directions, consitute a couple. A typical couple is shown in Fig. 1.19. A person attempting to open a screwed cap of a bottle exerts a couple on the cap. A couple is not equivalent to a simple force and cannot be replaced by a single force. The perpendicular distance between the lines of action of the constituent parallel force is called the arm of the couple. In Fig. 1.19 the forces P, P constitute a couple and d is the arm of the couple. The moment of a couple, also called turning effect of the couple, about any point in its plane is constant and equal to P × d.

Let us consider the turning effect of the couple about any 2 points, A and B, as shown in Fig. 1.19 (ii).

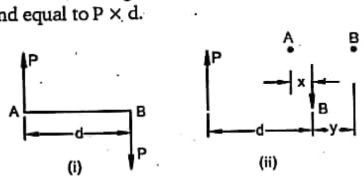


Fig. 1.19 Moment of couple

Total clockwise moment of couple about A	Total clockwise moment of couple about B
= P (x) + P (d – x)	= P (d + y) – Py
= Px + Pd – Px	= Pd + Py – Py
= Pd	= Pd

When turning a spanner, a couple is being exerted and the reaction of the nut on the spanner is equal and opposite to the force applied at the end of the spanner. (There must always be equal and opposite forces acting). In the example illustrated in the Fig. 1.20 the length of the spanner is considered to be the arm of the couple and the moment of the couple is 80 × 0.200 = 16 Nm.

It is important to remember that the arm length is the perpendicular distance. A force can have no truning moment about a point, in its own line of action.

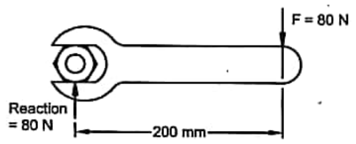


Fig. 1.20

1. Any two couples of equal moment in the same plane are equivalvent, *i.e.* a couple can be replaced by another couple of the same moment, provided it has be same direction, clockwise, or anticlockwise.

2. A couple acting on a body can only be balanced by another couple in the same plane. No single force can balance a couple.

3. Any number of couples in the same plane acting upon a body is equivalent to a single couple whose moment is equal to the algebric sum of the moments of the constituent couples.

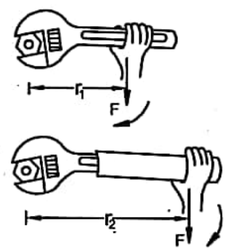


Fig. 1.21 Increasing torque of a spanner by use of a pipe on the spanner.

4. If two couples acting n one plane upon a body the equal and opposite the body must be in equilibrium.

When a couple acts upon a body there cannot be any motion in a straight line because the couple has no resultant force which will bring about a motion in a straight line. Under the influence of a couple a body can only rotate, if free, about an axis, through its centre of gravity perpendicular to the plane of the couple.

Example :

A rule 2 m long is pivoted at its centre; on the rule are 3 forces of 20 N, 50 N and 30 N acting as shown in Fig. 1.22. Find the value of force W to keep the rule in equilibrium.

Ans. :

Let us consider the moments of forces about the point O.

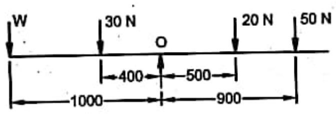


Fig. 1.22 Distances in mm

For the rule to be in a state of equilibrium, sum of clockwise moments = sum of the anticlockwise moments.

Sum of clockwise moments = (20×500) N - mm + (50×900) N-mm.

= 55000 N-mm

Sum of anticlockwise moments = (30×400) N -mm + W $\times$ 1000 N-mm

Load W should provide an anticlockwise moment equal to (55000 – 12000) N-mm = 43000 N-mm.

$\therefore$ 1m $\times$ W = 43000 N-mm = 43 N m (in an anticlockwise direction)

and W = 43 N

Example :

A bell crank lever has forces of 60 N and 20 N acting as shown in Fig. 1.23. Find the resultant turning moment on the bell crank lever.

Ans. :

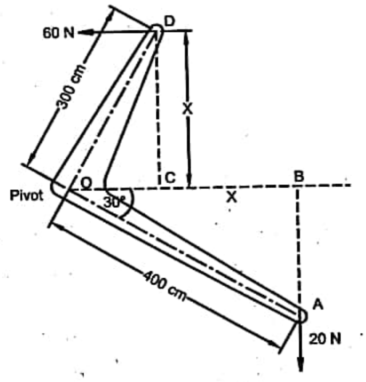


Fig. 1.23 A bell crank lever.

Before any moment can be calculated the right angled distances from the lines of action of the forces to the pivot must be found for each force. This can be done by calculation as follows :

IF OB = X

From traingle OAB, $\frac{X}{400} = \cos 30°$

$\therefore$ X = 400 $\times$ cos 30°

X = 346.4 mm

From triangle OCD, $\frac{Y}{300} = \cos 30°$

Y = 300 $\times$ cos 30°

Y = 259.8 mm

Now considering the moments about O,
clockwise moment is 20 N $\times$ X = 20 N $\times$ 346.4 mm = 6928 N.mm.

Anticlockwise moment, 60 N × Y = 60 N × 259.8 mm

= 15588 N-mm. Resultant moment on bell crank = 15588-6928

= 8660 N-mm anticlockwise.

Example :

A uniform bar of mass 100 kg rests with one end on a rough horizontal floor and the other end against a smooth vertical wall. Friction at the wall can be neglected. Find the magnitude of the reaction at the wall, and the magnitude and direction of the ground reaction when the bar makes 60° to the horizontal.

Ans. : If wall is smooth i offers no friction and reaction R_w of wall is at right angles to it.

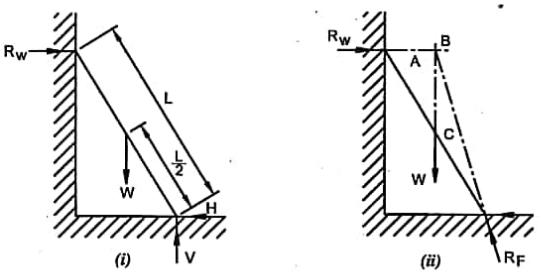


Fig. 1.24

Referring to Fig. 1.24 (ii), let the reaction R_F at the floor has a horizontal component H and a vertical component V; the horizontal component prevents the bar fro slipping Weight of bar = 981 N acting downward at its centre.

Applying the general conditions of equilibrium :

Upward forces = downward forces :

$$V = 981\text{ N} \qquad \text{--- (1)}$$

Forces to left = forces to right

$$H = R_w \qquad \text{--- (2)}$$

Taking moments about any convenient point, e.g, the foot of the ladder, since it eliminates two turning effects.

Clockwise moment = Anticlockwise moments

$$R_w \times L \sin 60° = 981\text{ N} \times \frac{L}{2} \cos 60°$$

Fig. 1.24 (iii)

Dividing by L

$$R_w \sin 60° = \frac{981\text{ N}}{2} \cos 60°$$

$$R_w \times 0.866 = \frac{981\text{N}}{2} \times 0.5$$

$$= 283.2\text{ N}$$

From Fig. 1.24 reaction at floor

$$R_F = \sqrt{981^2 + (283 \cdot 2)^2}$$

$$= \sqrt{962361 + 80202}$$

$$= 1021\text{ N}$$

Example :

ABCDE is horizontal beam of length 17m. AB = 5m, BC = 2m, CD = 7m and DE = 3m. The beam is simply supported at B and D, and there are downward loads of 400 N, 1000 N and 600 N at A, C, and E respectively. Calculate the up-ward reactions at B and D. (The weight of the beam can be neglected).

Fig. 1.25 is a diagrammatic representation of the loading.

Ans. :

If we take moments about B the supporting force at B, denoted by R_b has no turning effect, and we can form an equation with one unknown.

Taking moments about B, and working throughout in units of N and m.

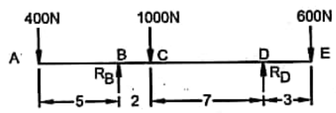


Fig. 1.25 A beam supported at two points

Clockwise moments = anticlockwise moments

$(1000 \times 2) + (600 \times 12) = (400 \times\ ) + (R_D + 9)$

$2000 + 7200 = 2000 + 9\,R_D$

$9\,R_d = 7200$

$\therefore\quad R_d = 800$ N

Taking moments about D, R_D having no turning effect.

Clockwise moments = anticlockwise moments.

$R_B\,(9) + (600 \times 3) = (400 \times 14) + (1000 \times 7)$

$9R_B = 5600 + 7000 - 1800$

$\therefore\quad R_B = 1200$ N.

Example :

Calculate the reactions R_A and R_B of the beam as shown in Fig. 1.26. The load of the beam per unit length is 300 N/m. A distributed load of 4000 N/m is placed over the beam as shown . Two concentrated loads of 3000N and 1000 N act on the beam as shown.

Ans. :

The load of the beam can be replaced by a concentrated load acting through its centre of gravity, of magnitude 3600 N. the C.G. is 6m from R_A. Similarly the distributed load can be replaced by a concentrated load of 20,000 N acting at a point 4.5 m from R_A.

Taking moments about R_A

Clockwise moments = anticlockwise moments

$(3,000 \times 1) + (20,000 \times 4.5) + (3,600 \times 6) + (1,000 \times 12)$

$= R_B \times 6$

or $3000 + 90,000 + 21,600 + 12,000 = R_2 \times 6$

or $R_B = 21,100$ N

Taking moment about R_B

Clockwise moments = anticlockwise moments

$(R_A \times 6) + (1,000 \times 6) = (3,000 \times 5) + (20,000 \times 1.5)$

or $R_A = 6,500$ N

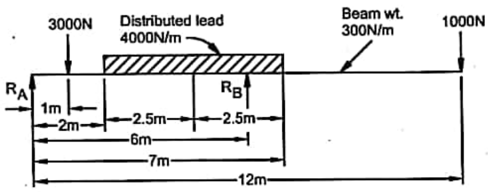


Fig. 1.26

Check upward forces $= R_B + R_A = 27,600$ N

downward forces = 27,600 N

Answer : R_B 21,100 N and $R_A = 6,500$ N

Example :

Figure 1.27 shows a shaft mounted in bearings A and B. The weight of the pulleys and shaft are given. Determine the load on each bearing.

Ans. :

Let the reactions, which will be equal to the loads on the bearings be R_A and R_B.

Taking moments about B,

$1 \times R_A = 40 \times 0.25 + 160 \times 0.5 + 32 \times 0.55 + 20 \times 0.85$

$\therefore\quad R_A = 10 + 80 + 17.6 + 17$

$\therefore\quad R_A = 124.6$ N.

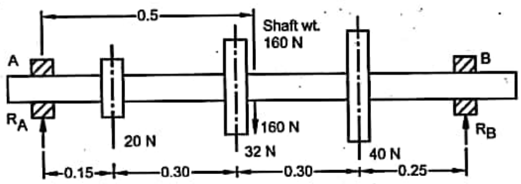


Fig. 1.27 Distances are in meters.

The value of R_B can be calcualted in a similar way by taking moments about A and we get $R_B = 127.4$ N.

The loads on the bearings are approximately equal. When designing a system involving load on bearing it should be kept in mind that the load on each bearing is approximately the same so that wear on the bearings is nearly equal.

Example :

A safety valve on a steam boiler (Fig. 1.28) is just on the point of blowing off steam. What should be the steam pressure under the following circumstances ?

(a) F is the fulcrum for the lever or arm of the safety valve.

(b) Diameter for the steam passage at P is 75mm.

(c) The lever has an effective length L of 750 mm fromthe fulcrum and weighs 12N. G is the point where its weight acts (Centre of gravity) and FG is 200mm.

(d) FV is 75mm

Weight W placed on the leverl is 240 N.

Ans. :

The area of the valve $= \frac{\pi}{4}(75)^2 = 5625 \times \frac{\pi}{4}\ \text{mm}^2$

Since Force = Pressure × area,

Force on the valve $= \text{Pressure} \times \frac{5625}{4 \times 10^6}$ N

Taking moments about pivot of lever, we have Sum of the clockwise moments = Sum of anticlockwise moments. Moment due to weight W or end of lever.

= (240 × 0.75) Nm clockwise

Moment due to weight of lever = (12 × 0.2) Nm clockwise.

$= \left(\text{Pressure} \times \frac{5625\pi}{4 \times 10^6}\right) \times 0.075$ Nm anticlockwise.

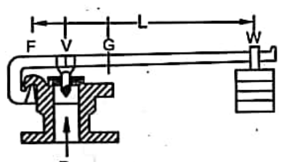


Fig. 1.28 Moment due to force of steam on valve

Therefore by the principle of moments, in units of Nm,

$$(240 \times 0.75) + (12 \times 0.2) = \left(\text{Pressure} \times \frac{5625\pi}{4 \times 10^6}\right) \times 0.075$$

$$\therefore\ (180 + 2.4)\,\text{Nm} = \left(\frac{421.875\pi}{4 \times 10^6} \times \text{pressure}\right)\text{Nm}$$

$$\text{So, Pressure} = 4 \times \frac{182.4}{421.875\pi} \times 10^6\ \text{N/m}^2 = 550.27\ \text{kN/m}^2$$

Centre of gravity :

The centre of gravity of a body is that point through which the resultant of the earth's pull upon the body passes and at which point the weight of the body can be considered as concentrated. The centre of gravity is occasionally referred to as the centre of mass or centroid. For bodies of common symmetrical shapes the centre of gravity is easily found out and is at following points.

A square or a rectangle : The intersection of diagonals.

A circle : the intersection of any two diameters.

A cube : The intersection of the centre lines, i.e. lines which join the centre of opposite square sides.

A triangle : The intersection of any two medians.

A rectangular prism : Middle point of its axes (Fig. 1.29) Axis of the prism is the straight line joining the intersections of the diagonals of the bases.

A cylinder : The centre of the axes.

A sphere : The centre.

Hemisphere : C.G. lies on the radius drawn perpendicular to the base from the centre and is 3/8th radius from the centre of the base.

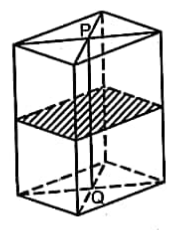


Fig. 1.29

A right pyramid and cone :

If h be the length of the axis, *i.e.* the altitude of the pyramid or cone from the centre of the base, C.G. is $\frac{h}{4}$ from the centre of the base.

Example :

On a cylinder of height 10cm and radius of base 4cm is placed a hemisphere of the same radius as that of the base of the cylinder, and made of the same material. Find the distance of the c.g. of the composite body from the centre of the base of the cylinder.

Ans. :

In Fig. 1.30 G_1 midpoint of OA, the axis of the cylinder is the c.g. of that body, G_2 is the c.g. of the hemisphere and $AG_2 = \frac{3}{8} r$ where r is the radius.

Vol. of the cylinder $= \pi r^2 h$ (h = the height)

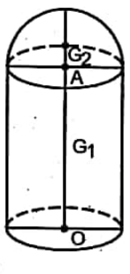


Fig. 1.30

Vol. of the hemisphere $= \frac{2}{3}\pi r^2$

$$= \frac{2\pi}{3} \times 64 = \frac{128\pi}{3}\ cm^3$$

$$\therefore \text{ Total volume } = 160\pi + \frac{128\pi}{3} = \frac{608\pi}{3}\ cm^3$$

$$OG_1 = \frac{h}{2} = 5\ cm$$

$$OG_2 = 10 = \left(\frac{3}{8} \times 4\right) = \frac{23}{2}\ cm.$$

Let $\bar{y}$ = the distance of c.g. of whole body from O. Since the bodies are of the same material, the weights are proportional to their volumes.

Taking moments about O,

$$\frac{608\pi}{3} \times \bar{y} = (160\pi \times 5) + \left(\frac{128\pi}{3} \times \frac{23}{2}\right)$$

$$\therefore\ \bar{y} = \frac{608}{3} = 800 + \frac{1472}{3} = \frac{3872}{3}$$

$$\therefore\ \bar{y} = \frac{3872}{3} = \times \frac{3}{608} = 6.37 \text{ cm approximately}$$

If a body is not of symmetrical shape the c.g. can be found out by application of the principle of moment of force. Consider a circular shaft of different diameters as shown in Fig. 1.31. It is easily appreciated that c.g. will be somewhere along the centre line of the diameter. If G is the c.g. of the whole shaft, a single force positioned at G will keep it in level. Each portion A, B and C of the shaft has its c.g. and if we take moments of the force (or weight) of A, B and C and also of the entire shaft at any convenient point then for equilibrium we should get

Clockwise moments = anticlockwise moments, and moments of separate parts = moments of whole assembly.

Example :

Find the position of the centre of gravity G of the Motor-shaft shown in Fig. 1.31.

Ans. :

Take moments of volume about line left-hand end.

Sum of moments of volume of separate parts = Moment of whole volume.

$(\pi \times 20^2 \times 60 \times 30) + (\pi + 30^2 \times 100 \times 110) + (\pi p \times 10^2 \times 160 \times 240) = \left\{(\pi \times 20^2 \times 60) + (\pi \times 30^2 \times 100) + (\pi \times 10^2 \times 160)\right\} \times \bar{X}$

π being common all through, can be discounted.

$(400 \times 60 \times 30) + (900 \times 100 \times 110) + (100 \times 160 \times 240)$

$= \{(400 \times 60) + (900 \times 100) + (100 \times 160)\} + \bar{X}$

$(24{,}000 \times 30) + (90{,}000 \times 110) + (16{,}000 \times 240)$

$= (24{,}000 + 90{,}000 + 16{,}000) + \bar{X}$

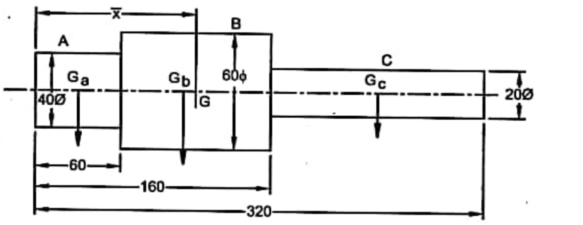


Fig. 1.31

$$720{,}000 + 9{,}900{,}000 + 3{,}840{,}000 = 130{,}000 \times \overline{X}$$

$$14{,}460{,}000 = 130{,}000 \times \overline{X}$$

$$\therefore \overline{X} = \frac{14{,}460{,}000}{130{,}000}$$

$$= 111\text{mm}$$

Answer : Centre of gravity lies 111 mm from left end edge.

Example :

Find the position of the centre of area of the eccentric shown in Fig. 1.32

Ans. :

In this particular case the 'sum' of the separate parts is in effect a subtraction. There is an axis of symmetry joining the centres of the circles, so let us take moments about line YY.

Take moments of area about line YY, and work in units of mm.

Sum of moments of area of separate parts

= Moments of whole area

$$(\pi \times 14^2)(14) - (\pi \times 7^2)(20) = (\pi \times 14^2 - \pi \times 7^2) \times \overline{X}$$

Divide all through by π

$$(14^2 \times 14) - (7^2 \times 20) = (14^2 - 7^2) \times \overline{X}$$

$$196 \times 14 - 19 \times 20 = (196 - 49) \times \overline{X}$$

$$2744 - 980 = 147 \times \overline{X}$$

$$\therefore \overline{X} = \frac{2744 - 980}{147}$$

$$\therefore \overline{X} = \frac{1764}{147}$$

$$= 12.$$

Ans. :

Centre of area is no axis of symmetry, 2 mm from centre of large circle, and 8mm from centre of small circle.

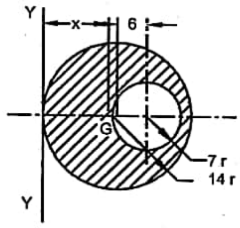


Fig. 1.32 (Figures are in mm)

QUESTIONS

1. Explain the difference between mass and weight ? What is the unit of (a) mass, and (b) weight in metric and S.I. systems ?

2. What is acceleration ? Give the equations for (a) linear acceleration (b) angular acceleration, that govern the starting velocity and final velocity after a time t and the travel covered ?

3. A car of mass 1 tonne and travelling at 90 km/h is brought to rest in 50 m by applying the brakes. What was the frictional force exerted on the tyres by the road, assuming that it was uniform ?

   (Ans. : 6250 N).

4. A rotating spindle starts from rest and attains a rotational speed of 630 rev/min in 15 seconds. Calculate the angular acceleration in rad/s² and the number of revolutions turned while accelerating to that speed.

   (Ans. 4.4. rad/s²; 78 $\frac{3}{4}$ rev.)

5. State Newton's Laws of motion ? What is momentum ? How does the second law of motion enable us to choose a proper unit of weight ?

6. What is S.I. system ? Name and define those units of S.I. system that are different from those in metric system as far as Mechanical Engineering is concerned ?

7. Write short notes on
   (a) Unity brackets
   (b) Vector
   (c) Column Vector
   (d) Radian
   (e) Centre of Gravity
   (f) Equilibrium

8. State the theorem of traingle of forces ? what is
   (a) an Equilibrant force
   (b) a resultant force

9. There forces, $F_1 = 30$ kgf, $F_2 = 40$ kgf and $F_3 = 50$ kgf act at a point. The angle between F_1 and F_2 is 45° and that between F_2 and F_3 is 60°. Find the magnitude of the resultant, graphically and by calculation.

   (Ans. : 89 kgf at 60° to F_1)

O O O

CHAPTER – 2

WORK, ENERGY & POWER

S.I. Units :

When a weight is lifted from the floor on to a table, work is done against the gravitational pull of the earth. If a weight is pulled over a rough surface, work is done. If you compress a spring, you do work against the elastic resitance of the spring. When steam pressure pushes a piston in a steam engine the steam does work on the piston.

A force is said to do work when its point of application moves in any direction not perpendicular to that of the force. If a weight of W Newtons is lifted through a vertical height of H metres, the work done on the weight is WH Newton meters. Work done is measured by the product of the force and the displacement *in the direction of the applied force.*

In Fig. 2.1 (left) if a horizontal force of P units acting on a particle moves it through a distance s from O to A in the direction inclined at θ to the axis OX, the displacement in the direction of the force is $s \times \cos\theta$ and the work done is $P \times s \cos\theta$. In the S.I. units, the unit of work done is one *Newton-metre,* which is also called *Joule* and is the work done by a force of 1 Newton when moving through a distance of 1 metre. In the C.G.S. system the unit of work is an erg, being the work done by a force of 1 dyne through a displacement of 1cm in the direction of force.

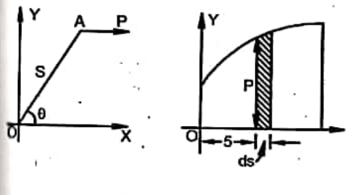


Fig. 2.1

1 Joule = $10^5 \times 10^2 = 10^7$ ergs = 1 Newton-metre

If a constant force P moves through a distance S the work done is PS units and it can be represented by a rectangle with P and S as the sides. If a force P moves a body through a small distance δs, the work done is P δs (Fig. 2.1 right). The total work done by displacement of the variable force through a distance is $\Sigma P. \delta s$ and is represented by the area under the force displacement diagram.

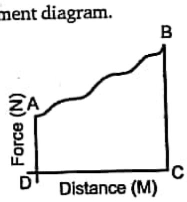


Fig. 2.2

In Fig. 2.2 the force is a varying one, shown by the line AB. The work done is represented by the area ABCD, is in Newton-meters and = distance × average force of AB.

Example :

A cage including mine car of mineral weighs 4 kN and is wounded up in a shaft 1000m deep. The winding rope weighs 10 N/metre length. Calculate the work done in raising the cage and rope. Neglect the rope length from the shaft top to winding drum.

Ans. :

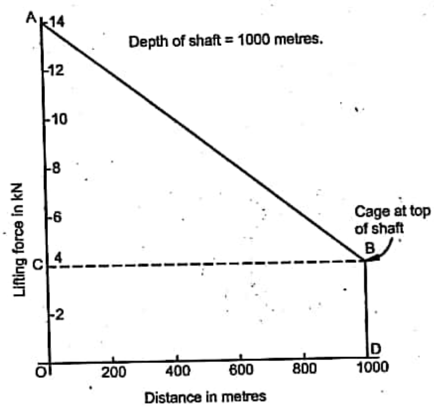


Fig. 2.3

When the cage is at the bottom of the shaft the total force required to lift the cage and the tope.

= weight of cage + weight of 1000m of rope

= 4 kN + (10 × 1000) N = 14 kN.

Total lifting force at the top of the shaft = weight of cage, 4 kN

When the cage is at the bottom of the shaft *i.e.* at A, the lifting force is 14KN. At point B, the top of the shaft, the lifting force is 4 KN.

Worke done = Area beneath the line AB

= Rectangle OCBD + triangle ABC

$= (OC + OD) + \frac{1}{3}$ (CB × CA)

$= (4 \times 1000) + \frac{1}{3} \times 1000 \times 10$ kNm

= 4000 + 5000

= 9000 MJ

Work done by a torque :

If a weight W hung by a string coiled on a drum of radius R (Fig. 24) descends as the drum completes 1 revolution, the work done by the earth's gravity on the weight is 2π RW units. Neglecting losses this is also the work done by the weight on the drum. Now O is the pivot or fulcrum and RW is the twisting moment or torque exerted on the drum and 2π is the angle turned through in radians.

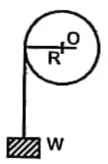


Fig. 2.4 Torque

Work done by a constant torque = Torque × angle turned through in radians.

$= T\theta$

where the torque T = force × radius

and θ = angle through which the force has moved (expressed in radians)

If T is quoted in Newton-metres the work done is also in Newtons meres, *i.e.* Joules.

If the torque is not constant, but variable,

Work done = average torque × θ

Power :

the rate of doing work is called power. The word rate brings in an element of time. Work is independent of time.

If W = the amount of work done,

t = time to do that work,

P = the power to be exerted, then

$$P = \frac{W}{t}$$

Thus if A performs some work in a given time and B performance the same quantity of work in a shorter time, B's rate of doing work is more; in other words, B is stronger and possess more power than A. In S.I. system the unit of power is watt (abbreviation W) so that

$$\text{power (in watts)} = \frac{\text{work done (in Newton-meters)}}{\text{time (in seconds)}}$$

$$= \frac{\text{Joules}}{\text{Seconds}}$$

A vehicle travelling at v m/s along a level road against a tractive resitance R newtons requires Rv watts to maintain this motion.

The watt is rather too small unit for engineering purposes and power is more conveniently expressed in kilowatts (kW); 1kW = 1000 watts. The commonly used term, horsepower, has the following relation with watt.

1 horsepower = 746 watts.

Example :

Find the power that a pump has to develope to pump 2000kg of water per min. to a height of 50 meters. Neglect losses and take $g = 9.8\ m/s^2$.

Ans. :

Work done in lifting 2000 kg

$= 2000 \times 9.8 \times 50$ Nm

= 980,000 Joules

$$\text{Power required} = \frac{980{,}000 \text{ Joules}}{60 \text{ seconds}} = 16{,}300 \text{ watts}$$

= 16.3 kW

Example :

A locomotive hauls a train of 250 tonnes on a level track at a uniform speed of 50 km/h. The frictional resistance is 6 kgf per tonne. Find the effective power expected.

Ans. :

Frictional resistance = 250×6 kgf

$= 250 \times 6 \times 9.81$ Newtons

Speed of train = 50 km/h = 13.9 m/s.

As the track is leveled and the speed is uniform, no force is required to overcome gravity or to cause acceleration, and the only resistance to overcome is that due to friction.

Force required to overcome friction = $250 \times 6 \times 9 \times 9.81$ N

Power to be expended = $250 \times 6 \times 9.81 \times 13.9$ watts.

$$\text{Power required in kW} = \frac{250 \times 6 \times 9.81 \times 13.9}{1000}$$

= 204.5 kW

Power developed by a constant torque

Work done by a constant torque = Tθ

where T is the torque in Nm and θ, the angle turned through, (expressed in radians) during t seconds.

$$\text{Power developed by a constant torque} = \frac{T\theta}{t}$$

But $\frac{\theta}{t}$ is the angular velocity of rotation, ω, and therefore, the power developed by a constant torque = Tω.

Example :

Find the constant torque provided by the shaft of an electrical motor if it developes 8.5 kW at 1400 rpm.

Ans. :

8.5 kW = 8500 W = 8500 J/s or 8500 Nm/s

$$\text{Angular velocity, } \omega \text{, rad/s} = 1400 \times \frac{2\pi}{60} = \frac{440}{3} \text{ rad/s}$$

$$P = T\omega$$

$$T = \frac{P}{\omega} = \frac{8500 \times 3}{440} \text{ Nm}$$

$$= 58 \text{ Nm}$$

Brake H.P. and indicated H.P.

Though the word horse power is not used in the S.I. systems it will continue to be used in the engineering industry and by the engineering students for some years.

1 H. P. = 746 watts.

So that 1 kW. = 1.34 H.P. (approx)

The maximum H.P. which a motor or engine is capable of developing is called its I.H.P. The complete I.H.P. is not available at the crank shaft of the reciprocating engine or the motor shaft of an electric motor as some of the power developed is utilised in overcoming losses of friction at bearings, resistance to wind, etc. in the engine or motor itself. The residual power available at the motor shaft or crank shaft and which can be used for operation of the connected machinery is called the brake H.P. of the engine. It is the output power of the engine/motor.

B.H.P. = I.H.P – H.P. lost due to friction, etc. inside the engine/motor

$$\text{Efficiency of a machine} = \frac{\text{B.H.P.}}{\text{I.H.P.}} \times 100$$

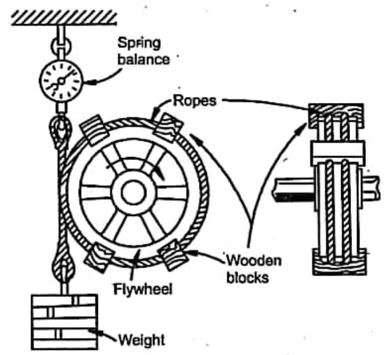


Fig. 2.5 Dynamometer

The output or breake H.P. of an engine or a motor (upto about 150 H.P.) may be determined experimentally by some form of braking system. Hence the name brake horse power or brake power. This braking system replaces the machinery which the motor or engine normally drives. Fig. 2.5 shows one simple form of friction brake.

The drum or pulley is fixed to the output shaft. Around the drum is a rope (or ropes) attached at one end to a spring balance. The rope is well lubricated manila rope of 25 mm diam. The rim of the drum should be quite smooth. If more than 1 rope is used, to keep the ropes in position a number of wooden blocks or distance pieces are attached to the rim as shown in the enlarged section.

'Weights' are suspended from the other end of the rope and hang freely. They are secured to the floor by a loose chain as a safety precaution against 'snatching'. When the drum revolves, the friction force that it exerts on the rope supports part of the suspended mass M so that the spring balance reading is reduced. The effective force opposing motion is therefore W–S where W is the down ward pull and S is the spring balance reading. The effective radius R is measured to the centre of the rope. (Because friction between the rope and drum generates heat the drum is water-cooled if the engine or motor runs for some time).

The work done against friction during one revolution.

$$= (W–S)\ 2\pi R.$$

If the speed of rotation is n rev/s,

work done per second = (W – S) 2π Rn.

This expression gives us a measure of the output.

Example :

During a test on a gas engine to determine its power, it was found that the speed of the brake drum was 350 rev/min, the suspended mass was 25 kg. and the spring balance reading 44 N. The effective radius of drum and rope was 0.6 m. Determine the brake power of the engine for this condition.

Ans. :

The work done against friction during one rev.

= (W – S) 2π R,

where W = downward pull in Newtons,

S = Spring balance reading in Newtons

R = effective radius to the centre of the rope in meters.

If the speed of rotation is n rev/s,

work done per second = (W – S) 2π Rn.

In the above example

Break power = $(25 \times 9.81 - 44)\ 2\pi \times 0.6 \times 350/60$

= 4.4 KW.

Example :

Find the brake power of an engine from the readings below :

Load M = 100 kgf; balance reading m = 15 kgf;

Speed = 1200 r.p.m., brake dia. = 30 cm.

Ans. :

Force on the brake = (100 – 15) kgf;

= 85×9.81 N

Work done per sec. = $85 \times 9.81 \times \pi \times 0.3 \times \dfrac{1200}{60}$ Joules

= 15700 J

1 Joule per sec = 1 Watt

$\therefore$ Power = 15. 7 kW.

Example :

A steam engine with a single cylinder has a cylinder dia. of 250mm and a piston stroke of 300 mm. The average steam pressure on the piston is 480 kN/m² and the piston makes 120 working strokes per min.

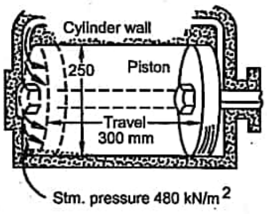


Fig. 2.6

Calculate :

(a) the average force on the piston,

(b) the work done on the piston,

(c) the power developed.

Ans :

(a) Force on the piston = Average steam pressure × crossectional area of the piston

$$= 480 \times \frac{\pi}{4}\left[\frac{250}{1000}\right]^2 \times 10^3 \text{N}$$

$$= \frac{480 \times \pi \times 62500 \times 10^3}{4 \times 10^6} \text{N}$$

= 23571.42 N

(b) The Piston moves through a stroke distance of 300mm, *i.e.* the piston force moves through a distance of 300mm per stroke.

Work done per stroke = $23571.42 \times \dfrac{300}{1000}$ Nm

= 7071.426 Nm

(c) No. of strokes per. min = 120

Worke done per min. = 7071.426×120 Nm

Work done per sec. = $\dfrac{7071.426 \times 120}{60}$ = 14142.8 J.

Power = 14.143 kW

Energy :

Energy of a body is its capacity for doring work. We have seen that work is done when a force moves through a distance; power is the rate of doing work. Work, power and energy are, therefore, closely related. A body may posses capacity for doing work because of its position, its motion, its construction or because of the process it has undergone. A spring of a watch possesses stored energy when it undergoes the process of winding and it is then capable of releasing that stored energy over 24–36 hours. A rifle bullet does work when it strike a target; a hammer does work when it hits a nail; the bullet and the hammer possess energy because of their motion. petrol, coal and explosives possess chemical energy because of their "construction", *i.e.* the manner in which they have been formed. Every person requires energy for doing his normal work and usual movements liek walking breathing, writing, etc.

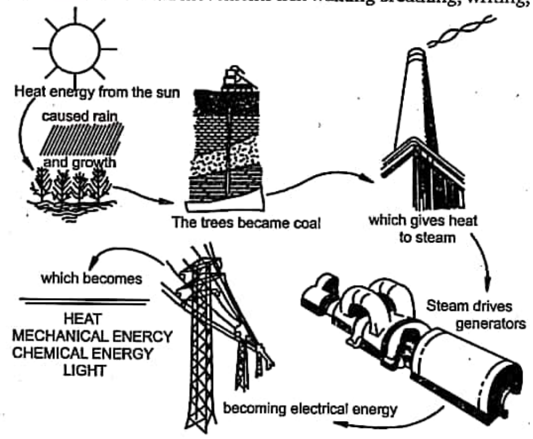


Fig. 2.7 Transformation of original solar energy into various forms.

This energy he gets from the oxygen inhaled and the food consumed. Your "automatic" wrist watch which needs no winding, keeps running by virtue of the energy it receives and stores in its main spring from the movements of the wrist. It will be realised that energy in one form can be changed into energy in another form. The chemical energy in a battery of a motor car can be changed into electrical energy to drive its starter-motor or produce an electric spark that jumps across the spark plugs. The chemical energy of an

explosive is changed to mechanical energy for breaking rocks. The potential energy in the water of a reservoir changes into kinetic energy when the water falls to a lower level and into mechanical energy of a turbine which rotates as the water impinges on it with high velocity. This mechanical energy derived from water is converted into electrical energy in a hydro-electric power station when the turbine rotates an armature in the alternator. Fig. 2.7 shows some of the ways in which energy in one form can be changed into energy into another form.

The original source of all energies is the sun and its heat energy is provided by the nuclear reactions on its surface. These nuclear reactions take place continually and the sun is, therefore, considered, according to the present thinking, an inexhaustible source of energy.

Various experiments by scientists have established one fact; no matter how good is an energy-changing device, we can never destroy or create energy and this important fact is embodied in the "Principle of conservation of energy". According to the principle of conservation of energy, energy can neither be created nor destroyed; it can only be changed from one form in to another, the energy does work, some of which is noticeable and useful but some may be less useful. For example, in an electric motor the energy input is the electrical energy but the useful output energy is the mechanical energy available from the motorshaft driving, say, a drilling machine and the less useful form of energy is the heat generated which is ultimately wasted. The heat represents a loss which is unavoidable. With any energy-changing device the energy output is always less than the energy input and its Efficiency $= \dfrac{\text{energy output}}{\text{energy input}}$ in a given time.

Potential energy :

The potential energy of a particle is the work which the forces acting on it would do if it moved from its actual position to some standard position.

Thus if a particle of mass M kg is at a height h meters above the ground, and we take the ground as the standard position, its potential energy is the work that would be done by the force of gravity Mg newtons moving through h meters, that is, Mgh Joules. The word *potential* suggests something that is not being used but is available for use.

The potential energy of a rigid body of mass M is Mgh, where h is the height of its centre of gravity above some standard position or datum.

The potential energy of a particle or a body usually involves the force of gravity but such thing as a stretched spring has a potential energy realted to internal forces of ealsticity. In the case of a compressed spring, or a stretched spring the potential energy is the work which would be done in restoring it to its natural shape.

Kinetic energy :

The kinetic energy of a particle of mass M moving with velocity v is defined as the quantity $\frac{1}{2}Mv^2$.

The kinetic energy of a rigid body of mass M moving in such a way that all particles of the body have the same velocity v, is the quantity $\frac{1}{2}Mv^2$.

If a particle moves in a straight line with constant acceleration f through a distance s and in that distance its velocity is increased from its initial value u to its final value v, we have

$$v^2 = u^2 + 2fs$$

and hence

$$\frac{1}{2}Mv^2 - \frac{1}{2}Mu^2 = Mfs$$

If P is the force causing the acceleration f P = Mf

and we have $\frac{1}{2}Mv^2 - \frac{1}{2}Mu^2 = Ps$ --- (1)

Since the force P is along the direction of motion, Ps is the work done in the displacement and hence equation (1) shows that

Change in linear kinetic energy = work done --- (2)

If a body of mass moment of inertia I, rotates with angular velocity ω it has an angular kinetic energy of $\frac{1}{2}I\omega^2 = \frac{1}{2}MK^2\omega^2$ where K is radius of gyration, and M is its mass.

If a body has both linear and angular motion, e.g. the road wheel of a moving car, then the body has total kinetic energy which is the sum of the linear and angular kinetic energies, Hence.

$$\text{Total K.E.} = \frac{1}{2}Mv^2 + \frac{1}{2}I\omega^2$$

Example :

A bullet of mass 5 gramms has a speed of 500 m/s. What is the kinetic energy ? How far wil it penetrate a wooden block if the latter offers a constant resistance of 1 kN to the motion of the bullet ?

Ans. :

Mass of the bullet is 5 grammes *i.e.* $\frac{5}{1000}$ kg.

K.E. of the bullet is $\frac{1}{2}Mv^2$

$$= \frac{1}{2} \times \frac{5}{1000} \times (500)^2 = \frac{2500}{4} = 625 \text{ J}$$

If is the distance of penetration,

Work done by the bullet against resistance of wood

$= 100 \times s$ joules

This energy for doing work has come from the K.E. of the bullet.

$625 = 1000\ s$

$s = 0.625$ m

Sum of P.E. and K.E. is constant :

Let a body of mass M be placed at a hight of H from the ground level.

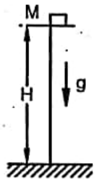


Fig. 2.8

Potential energy = MgH

The Kinetic energy of the body is zero.

P.E. + K.E. = MgH --- (1)

When the body falls freely, i.e. under the action of earth's gravity, to the ground level.

at the ground level P.E. = 0 since H is zero

and K.E. = $\frac{1}{2}Mv^2$

Now $v^2 = u^2 + 2gH = 2gH$ since $u = 0$

$$K.E. = \frac{1}{2} M \times 2gH = MgH$$

$$P.E. + K.E. = O + MgH = MgH \qquad \text{--- (2)}$$

This shows that the sum of potential energy and kinetic energy is constant for a given body with respect to any assumed datum.

Example :

Find (i) the work done in lifting a mass of 5kg to a height of 10m and (ii) the kinetic energy of a mass of 5 kg which has falls from rest through a height of 10m.

Ans. :

(i) Work = Force × distance covered in direction of force

$= (5 \times 9.81)\ N \times 10\ m$

$= 490.5J$

(ii) Velocity v after falling 10m is given by the formula

$v^2 = u^2 + 2gh$

$= 0 + 2 \times 9.81\ m/s^2 \times 10m$

$= 196.2$

$$K.E. = \frac{1}{2} Mv^2$$

$$= \frac{1}{2} \times 5\ kg \times 196.2$$

$= 490.5$ J.

If a body possesses any other from of energy in addition to potential and kinetic energy, the sum total of all such energies remain constant and this fact is stated in Bernoulli's theorem which is explained further in the chapter on Pumping.

Moment of inertia and radius of gyration :

From Newton's First Law, we know that any "body" has an inherent property to keep its state of rest or of uniform motion along a straight line unless it gets compelled by some external force to change its state. This property of the body is called inertia, as explained earlier, and often referred to as mass moment of inertia.

In the case of rotary motion also, any body by virtue of its own property tends to rotate about an axis with a uniform angular velocity and this inertial effect about the axis of rotation is called the moment of Inertia (I). It is also referred to as **mass moment of inertia**.

Mathematically, $I = \frac{T}{\propto}$

where T = Torque, $\propto$ = angular acceleration

and I = Moment of inertia.

It can also be shown that

$$I = mr^2$$

where I is the moment of inertia of a particle of mass m moving in a circle of radius r from its axis of rotation. A revolving body consists of many small particles, each at a different radius from the axis of rotation (see Fig. 1.2) and the sum of the effect of these has to be calculated by adding up all the separate mr^2 terms.

or $I = \Sigma mr^2$

$= MK^2$

Where K is called the radius of gyration and M = Total mass of the body. The radius of gyration is the radius to that point in the body where the total mass of a rotating body is supposed to be concentrated.

The moment of inertia depends not only on the masses of partciles of which the body is composed but also their distances from the axis of rotation. The moment of inertia of a body about the axis of rotation is constant and for the same body the moment of inertia is different for different axis of rotation. It has same value irrespective of the state of motion of a body. Unit for moment of inertia is kg m^2.

Moments of Inertia and radius gyration of some bodies :

Description	Axis of rotation	M.I.	Radius of gyration
Thin uniform rod of mass M, length l	Through its centre and perp. to its length	$\frac{Ml^2}{12}$	$\frac{l}{\sqrt{12}}$
do	Through its end and prep. to its length	$\frac{Ml^2}{3}$	$\frac{1}{\sqrt{3}}$
Thin ring of mass M and radius R	Through its centre and perp. to its plane	MR^2	R
Circular disc of mass M and radius e	do	$\frac{MR^2}{2}$	$\frac{R}{\sqrt{2}}$
Right cylinder	do	do	do
Hollow sphere (shell) of mass M and radius R	coinciding with its diameter	$\frac{2}{3}MR^2$	$R\sqrt{\frac{2}{3}}$
Solid sphere of mass M and radius R	do	$\frac{2}{5}MR^2$	$R\sqrt{\frac{2}{5}}$

Angular K.E. and work done by torque :

We have seen that the work done by a constant torque T is given by the relationship.

Work done = $T\theta$ = torque × angle of rotation (radius)

Let a rotating body, having moment of inertia I, be subjected to a constant torque T, and let the angular velocity increase from ω_1 to ω_2 while rotating through angle θ.

Now $T = I\propto$ and $\omega_2^2 - \omega_1^2 = 2\propto\theta$

Therefore $T = \frac{I\left(\omega_2^2 - \omega_1^2\right)}{2\theta}$

and $T\theta = \frac{1}{2}I\left(\omega_2^2 - \omega_1^2\right) = \frac{1}{2}MK^2\left(\omega_2^2 - \omega_1^2\right)$

i.e. work done on a body by torque T = increase in K.E. (angular) of the body.

Compare this with the relationship :

Work done by a force = increase in linear K. E. of a body.

Example :

A flywheel of mass 5 kg and radius 20cm, initially at rest, is subjected to constant torque so that its angular velocity after 5 sec. is 25 rad/sec. Calcualte the angular acceleration and torque.

Ans. :

In such examples of flywheel, a flywheel may be treated as a circular disc of mass M and radius R so that the moment of inertia is given by the formula $I = \frac{MR^2}{2}$, where M is its mass and R the actual radius. Some flywheels have thin spokes and the mass is concentrated at the periphery. In that case radius of gyration has to be calculated from a separate formula.

In this example M = 5 kg; R = 20 cm = 0.2m

$\omega_0 = 0$; $\omega = 25$ rad/s; t = 5 s

Moment of Inertia, $I = \frac{MR^2}{2} = \frac{5 \times 0.2 \times 0.2}{2} = 0.1\ \text{kgm}^3$

To calculate angular acceleration :

Using the first equation of motion

$\omega = \omega_0 + \propto t$

$\therefore \quad \alpha = \frac{\omega - \omega_0}{t} = \frac{25 - 0}{5} = 5\ \text{rad/s}^2$

Torque, $T = I\propto = 0.1 \times 5 = 0.5$ Nm

Example :

A solid sphere of mass 10 kg and radius 5 cm is rotating uniformly about its diameter. If it performs 120 r.p.m., calculate its (a) kinetic enegy and (b) angular momentum.

Ans. :

M = 10kg ; R = 5cm = 0.05m

120 r.p.m. = 2 r.p.s.

angular velocity $\omega = 2\pi \times 2$ r.p.s $= 4\pi$ rad/s

$$\text{Moment of Inertia, } I = \frac{2}{5} MR^2 = \frac{2 \times 5 \times 0.05 \times 0.05}{5}$$

$$= 0.005 \text{ kgm}^2$$

$$\text{(a) Kinetic energy } E = \frac{1}{2} I\omega^2 = \frac{1}{2} \times 0.005 \times (4\pi)^2 = 0.394\text{J}$$

(b) Angular momentum $I\omega = 0.005 \times 4\pi$

$= 0.0628$ kg m²/s

After an understanding of linear and rotational motions the following analogy in a summerised form should be noted:

Linear motion	Rotational motion
Linear displacement, s Velocity, v Acceleration, f Mass, M Force, F or P Force = M ×f	Angular displacement, θ Angular velocity, ω Angular acceleration, α Moment of inertia, I ($I = \Sigma mr^2$). Torque, T Torque = I× ∝
Kinetic energy = $\frac{1}{2}Mv^2$	Kinetic energy = $\frac{1}{2}I\omega^2$
Work done = Ps Momentum = MV	Work done = Tθ Angular momentum = Iω

Energy of rotation and flywheel :

A reciprocating engine or an electric motor is often connected to a load which is of a fluctuating nature, where the peak demand is much higher than the average demand. Suppose the load is such that the peak demand is 1.8 kW, the average demand is 1kW and the minimum is about 0.7kw. It would be most uneconomical to provide a motor of, sav, 2kW if this power potential is used only at intervals. It would be far more sensible to provide a motor of power slightly in excess of the average demand, say, a motor of 1.2 kW. Such motor can meet the peak power demand if some device is provided to work as a storehouse of energy so that when the energy supply exceeds the demand, the excess energy can remain stored in the device and the excess "stored" energy can be used when the energy demand exceeds the supply. Such device is a flywheel. A flywheel is a large, heavy-rimmed wheel secured to the crankshaft of a reciprocating engine or the shaft of an electric motor or prime mover. Its function is to maintain an approximately equal rate of motion among the moving parts of the engine during each complete revolution. It accomplishes this task by slightly increasing in speed and so storing up kinetic energy when the turning effort or torque is a maximum, *i.e.* when the connecting rod is approximately at right-angles to the crank, and by slightly decreasing its speed to give up its excess of energy when the torque is minimum, *i.e.* when the piston-rod, connecting rod, and crank are approximately in line. This variation in speed is invetitable if the flywheel is to serve any useful purpose. but it is only a small variation, depending on the mass of the wheel and on the energy fluctuation required in the machine concerned to maintain the speed within the desired limit.

If the shaft of a flywheel rotates at N r.p.m. the angular velocity of the flywheel, $\omega = \frac{2\pi N}{60}$ rad/s. The kinetic energy of rotation of the flywheel is then.

$$E = \frac{I\omega^2}{2} \text{ Joules}.$$

where I is the moment of inertia in kg – m²I = MK², if M is mass in kg and K is radius of gyration in meters.

It is convenient to express it in terms of the shaft r.p.m., N

$$E = \frac{I\omega^2}{2} = \frac{1}{2} \times \frac{4\pi^2}{3600} N^2 = \frac{4\pi^2 I}{7200} \times N^2 = MN^2, \text{ where}$$

$M = \frac{4\pi^2 I}{7200}$ is a constant for a given flywheel. This M should not be confused with M used for mass in this book.

From the expression $E = MN^2$, it is clear that for a given speed, the larger the "M" of the flywheel the greater is the energy stored by it. The object in fitting a heavy flywheel to an engine will now be clear. It is evident however that in the case of a rotating flywheel, the velocity varies for each particle of the wheel according to its distance from thecentre of rotation. The velocity of the outer circumference f the rim, for example, is much greater than that of the hub of the flywheel and the kinetic energy stored up by the outer parts of the wheel is thus greater than that stored up by those nearer the centre. For this reason flywheels are usually designed with a heavy rim and with their spokes or central disc joining the rim to the hub as light as possible consistent with adequate strength.

Rims of some flywheels have flat circumferencial surface : To accommodate flat belts but some have grooved rims for V-belts. Spokes of some flywheels may be in the form of blades which work like a fan and throw ambient cool air on the engine, thereby cooling it.

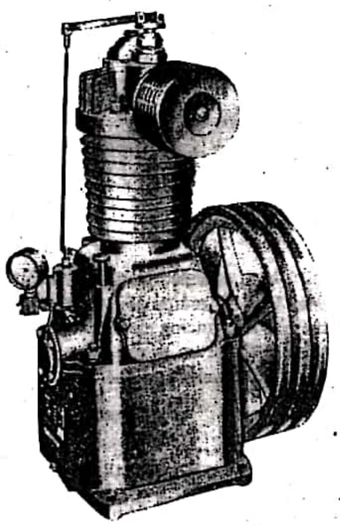


Fig. 2.8 A. The flywheel on the right has grooves to carry V-ropes for transmission (Picture of a vertical compressor model N3 of Kirloskar Pneumatic Co.).

Example :

The shaft of an electric motor rorates at 900 r.p.m. and it is brought to rest in 15 complete revolutions by a braking torque. If the constant frictional torque in the bearings is 200 Nm through out and the moment of Inertia of the shaft is 50 Kg m², what is the magnitude of the breaking torque ?

Ans. :

Angular velocity of the shaft, ω

$$= \frac{2\pi \times 900}{60} = \frac{2 \times 22}{7} \times \frac{900}{60} = 94.25 \text{ rad/s}$$

The K.E. of rotation at this speed is

$$\frac{I\omega^2}{2} = \frac{50}{2} \times (94.25)^2 \text{ Joules} = 222076 \text{ J}$$

This K.E. is absorbed by the breaking and frictional torque in 15 complete revolutions. If B is the breaking torque, the work absorbed by the two torques.

$= (B + 200) \times$ angle turned through

$= (B + 200) \times 2\pi \times 15$ Joules.

$\therefore \quad (B + 200) \times 30\pi = 50/2 \times (94.25)^2$

$$(B + 200) = \frac{222076}{30\pi} = 2356 \text{ Nm}$$

$\therefore \quad B = 2156$ Nm

Example :

A lift of mass 610 kg is attached to a rope passing over a drum of effective diameter 1.25m, radius of gyration 560mm and mass 54kg, and from there to a balancing mass of 460 kg. Find the torque that must be applied to the drum to raise the lift with an acceleration of 0.9 m/s². Neglect weight of the rope and friction in the system.

Ans. :

In an example of this nature there are three actions involved.

(a) the lift must be accelerated upwards.

(b) balancing mass must be accelerated downwards.

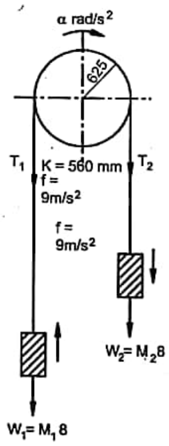
(c) The drum has to be accelerated.

(a) acceleration of the lift upwards.

Tension T_1 in the rope = Force to support lift + force to accelerate.

$= 610 \times 9.81 + 610 \times 0.9$

$= 6533$ N.



(b) downward acceleration of balancing mass.

Tension T_2 in the rope

= Force W_2 to support balancing mass – force to accelerate it.

$= 460 \times 9.81 - 460 \times 0.9$

$= 4098$ N.

Torque required on the drum for the lift and the balance mass

$= (T_1 - T_2) \times R$

$= (6533 - 4098) \times 0.625 = 1522$ Nm.

(c) Torque to accelerate drum

$= I\alpha = MK^2 \alpha$ (M is mass)

$$\alpha = \frac{0.9}{0.625} = 1.44 \text{ rad/s}^2$$

Fig. 2.9

Accelerating torque $= 54 \times 0.56^2 \times 1.44 = 24.39$ Nm.

Total torque required $= 1522 + 24.39 = 1546.4$ Nm.

Example :

A torque of 815 Nm acts on a flywheel of mass 910 kg and radius 0.9m. Find the time required for the flywheel to reach a speed of 5 rev/s from start.

Ans. :

If T = torque, M = mass,
I = moment of inertia,
α = angular acceleration

then $T = I\alpha = MK^2\alpha$

$$\therefore \quad \alpha = \frac{T}{MK^2} = \frac{815}{910 \times (0.9)^2} = 1.105 \text{ rad/s}^2$$

Now $\omega_2 = \omega_1 + \alpha t$ but $\omega_1 = 0$.

$$\therefore \quad t = \frac{\omega_2}{\alpha}$$

$$= \frac{5 \times 2\pi}{1.105} = 28.4 \text{ seconds.}$$

Governor :

A governor is a device that keeps a machine or an engine at a mean steady, speed over a long period, regardless of the changes in load.

A common type governor consists of a sleeve, a vertical spindle passing through the sleeve and connected to two weights in the form of balls through hinged arms. The sleeve carries a weight on it and is connected to the balls at their lower ends by hinged arms. The spindle is driven through gearing by the main shaft of the engine or machine. When the engine shaft and the spindle of the governor rotates at a fast speed the balls move outwards away from the spindle, because of the centrifugal force; when the engine shaft slows down, the balls drop inward. When the balls move outward, the sleeve travels up and when they move inward the sleeve travels down. This movement of the sleeve, through linkages, regulates the addmission of steam to a steam engine and admission of fuel to a petrol or diesel engine so that the speed of the engine is kept nearly constant within certain limits. Some governors use a spring in place of the weighted sleeve, e.g. Hartnell governor. When an engine is required to work against a heavy load, it slows down so that the balls drop inward and admit more fuel to the engine, just as a car driver pushes down the accelerator to maintain speed when going uphill. If the load on the engine becomes less, it tends to race, the balls move outward from the spindle and with the help of linkages reduce supply to fuel to the engine. Most governors are controlled by a hand lever that fixes the speed at which the governor will hold the engine. New cars and trucks generally have governers on their engines.

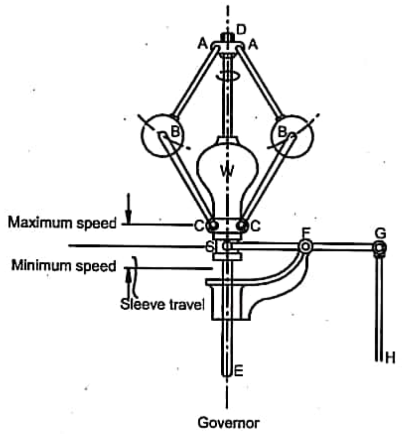


Fig. 2.10

The function of a governor is different from that of a flywheel. A flywheel does not control speed variations caused by a varying load but it controls the speed varations caused by the fluctuations of the engines turning movement during each cycle of operatins. In a 4-stroke petrol engine, for example, only one stroke, *i.e.* the power stroke exerts the maximum torque on the crankshaft but the other strokes do not exert any torque and the crankshaft slows down abruptly. This variation in the torque during a cycle momentarily increase or decreases the speed of the crankshaft resulting in uneven running and the flywheel tries to even out the fluctuations of speed during the operating cycle. It performs this function by virtue of its large inertia so that it stores energy when the torque is maximum and releases the stored energy when the torque is low. The result is that the speed fluctuations during a complete operating cycle are controlled and the speed of the crankshaft is maintained within certain specific limits.

Indicator Diagram :

A diagram in which pressure is plotted along the ordinate, and volume along the abscissa, is called the P-V diagram or indicator diagram.

For calculating the work done per stroke by the piston of a reciprocating engine, *e.g.* in a gas engine, in a steam engine, etc. use is made of indicator diagrams. In such diagram, any point on the diagram indicates the pressure on the piston and the distance travelled by the latter at any instant so that the area of the diagram gives the amount of work done in one stroke. (Fig. 2.11). In a double acting engine work is done by the piston in the forward as well as backward travel and there will be two diagrams, one for the forward travel and the other for the backward travel. For a steam engine the indicator diagram is drawn by a steam indicator shown schematically in Fig. 2.12.

The steam Indicator

A steam indicator comprises a small cylinder, C, in which there is a piston, P, whose vertical position is regulated by the steam pressure existing in the engine-cylinder, counterbalanced by a spiral spring, S. The cylinder C, is screwed into a socket on the engine-cylinder and the tap, T, enables steam to be admitted to the indicator when desired. The vertical movement of piston P is transferred via a piston-rod to the pencil, B, which thus traces a line upon the indicator card, A. This card is normally mounted on a drum pivoted vertically but here shown, for simplicity, in a horizontal side or frame, F, which is alternately drawn to right and left by the reciprocating motion of the engine-crosshead opposed by the weight, W. Their motion is transmitted to the frame via a flexible cord as at K. The vertical movement of pencil B combined with the horizontal (or rotary) movement of the card results in a closed figure being drawn, known as the indicator diagram.

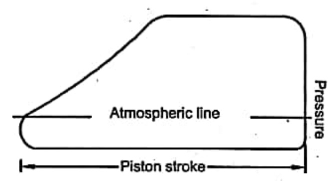


Fig. 2.11 Indicator diagram.

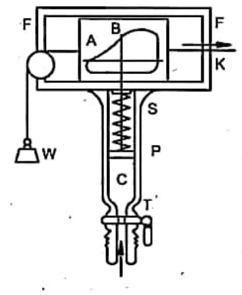


Fig. 2.12 Indicator diagram for a steam engine.

An indicator diagram gives a reliable record of the distribution of steam in an engine-cylinder at any instant during the stroke. In particular, it tells us :

1. The pressure existing at either side of the piston at any moment.
2. The effective pressure actuating the piston at any moment. From this we may calculate.
   i. The mean effective pressure on the piston throughout the stroke.
   ii. The indicated horsepower of the engine.

   In addition, the diagram gives full information as to the points of cut-off, release, and compression, and enables us to discover any defect in the setting of the valves, or any leakage of steam past the piston or through the valves.

QUESTIONS

1. Explain the terms work, energy and power and show how they are inter-related
2. Describe a dynamometer and the manner of finding out the brake-power of an electric motor ?
3. Explain the purpose of using (a) a flywheel and (b) a governor on a machine ? Describe a governer with sketch.
4. A flywheel is found to accelerate at 0.2 rad/s² under a torque of 45 Nm. Determine its mass moment of inertia. If the flywheel starts from rest, determine the energy stored in it after an accelerating torque of 68 Nm has acted for 100 seconds.

   Through how many revolutions has the flywheel turned during that time ? (Ans. : 225 kg-m²; 101250 J; 238.7 rev).

O O O

CHAPTER – 3

FRICTION, BEARINGS, LUBRICATION, INCLINED PLANE, BOLTS AND NUTS

Friction :

Friction may be described as a force which opposes movement of one surface over another. When a person attempts to slide a body, say, a book resting on the horizontal surface of a table, he has to apply a force to cause the sliding because the sliding is resisted by friction between the book and the surface of the table. When a body rests on a horizontal surface, the forces acting on it to keep it in equilibrium are :

1. Its weight, W, acting vertically downwards.
2. The normal reaction, R, of the table surface equal and opposite to the weight of the body.

Neither of these forces has any resolved part at right angles to itself. *i.e.* in a horizontal direction and the force that comes into play when sliding one body over another and which resists motion is the frictional force. A cyclist cycling on a level road has to exert effort to overcome the friction (i) between the cycle tyres and the road surface, (ii) at the axles of front and reas wheels (iii) at the axle of the pedal and (iv) between chain and sprocker wheels. The rougher the road surface, the greater is the frictional resitance to the tyres and greater is the effort needed by the cyclist. It should, however, not be considered that friction is an undesirable force or effort. When the cyclist wants to stop the cycle at a desired place, he applies the brakes which being the rubber blocks of the brakes in contact with the moving rim of the cycle and the friction between the rubber blocks and the steel rim causes the latter to slow down and eventually stop. Without friction walking and running would be difficult. A person who has tried to walk on ice which has a very smooth surface compared to the tar road must have experienced how difficult it is to walk on the smooth surface which offers very little resistance. Moving parts of machinery encounter friction during movement over stationary parts.

Limiting Friction :

In Fig. 3.1 A is a body resting on a horizontal surface. It is connected by a string passing over a smooth pulley to a weight W. The weight W acts vertically downwards and sets up tension T in the string which is transmmitted to the body A and acts on it as a horizontal force. When the body does not move the frictional force F which comes into play is equal and opposite to T. If the weight W is gradually increased by addition of smaller weights, a stage comes when the body just begins to slide. This limiting force of T is equal and opposite to the frictional force F. The frictional force has therefore a limit below which the body will be at rest or static; once a body begins to slide, the movement is still opposed by friction but the frictional force is comparatively less. This is obvious when pushing a car at rest on a road. Once than that required to push it when at rest. There is, therefore static frictional resistance (also called limiting friction) which has to overcome to commence the movement of a body from rest, and kinetic frictional resistance which opposes the movement of a body having velocity. The frictional resistance the car starts moving the force required to keep it moving is less.

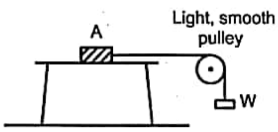


Fig. 3.1

1. is independent of the speed of sliding at low and moderate velocities.
2. is independent of the nature of the materials of the surface in contact.
3. is independent of the area of contact.
4. is proportional to the normal force between the bodies in contact.
5. acts in a direction opposite to that in which the body moves or tends to move and is always self-adjusting .

$$\text{The ratio} = \frac{\text{frictional resistance}}{\text{normal reaction between bodies}} = \text{coefficient of friction } (\mu)$$

$\therefore$ The coefficient of friction is constant for any given two surfaces and is approximately as follows :

Dry wood on wood – 0.35 (if polished) to 0.5 (if dry).

Dry metal on wood – 0.15 to 0.3.

Oiled metal on metal – 0.05 to 0.10; variable.

Wood on stone – 0.6

Leather on wood $\simeq$ 0.62

Some of the operations of machinery in a mine where friction plays a significant role are given below; without adequate friction the operations will not be effective.

(a) Brakes on haulage engines and windings systems : Friction between brake linings or brake blocks and the moving drum.

(b) Friction winders (Koepe winders) : Friction between the winding rope and winding pulley.

(c) Endless rope haulage : Friction between surge wheel and rope.

(d) Friction clutches : Friction between revolving friction clutch and stationary drum that has to be put in motion.

(e) Bet conveyor : Friction between the belt and the driving drum.

(f) Mine fans : Friction between the V-belt or flat belt and the driving pulley of motor.

(g) Friction props : Friction between the upper member and lower member of a telescopic friction prop and the clamp on it.

(h) Temporary supports during shaft sinking : Friction between laggings and shaft wall on the one side and between the laggings and the steel curb on the other side.

Some artificially made materials like asbestos brake lining have a high coefficient of friction. They are marketed under different trade names *i.e. Ferrodo* brake linings. These are used as lining material on wooden blocks for braking of haulages engines and winding engines and brake shoe lining of motor cars, trucks, etc. If such lining material is not used, the wood should posses high coefficient of friction and seasoned timber from mango tree is found suitable for this purpose, specially for brake blocks of small haulage engines. Rosin is used to increase the frictional between leather, canvas or rubber belt and the metallic pulleys.

The coefficient of friction is reduced by the use of a lubricant between two surfaces. A lubricant acts as a very thin film between the two surfaces and prevents direct contact between them. The lubricants are mostly liquid (oils). Grease (semisolid), graphite and molybdenum disulphite (solids) are other lubricants.

Suppose a body of weight W rests on an inclined surface whose Inclination is such that the body is just about to slid. (Fig. 3.2). The weight W acts vertically downwards and can be resolved into two components, one along the plane of the inclined surface, W sin θ and the other at right angles to it W cos θ which is also equal to the normal reaction to the plane. When the body is just about to slide the frictional force coming into play is equal and opposite to W sin θ and the coefficient of friction, $\mu = \frac{W \sin\theta}{W \cos\theta}$

$= \tan\theta$

Frictional force on a level surface, $F = W\mu$

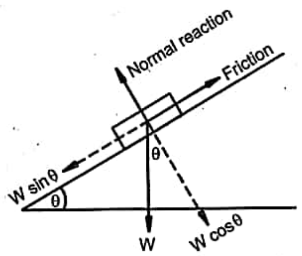


Fig. 3.2

Example :

A ladder of uniform weight is to be placed with one end on a horizontal floor and the other against a vertical wall. If the limiting coefficient of friction at each surface is 0.2, what is the least angle the ladder can make with the horizontal without slipping ?

Ans. :

Let the gravitational force acting on the ladder be F and this force acts vertically downward at the centre of gravity.

Let the length of the ladder be L, and when the ladder is on the point of slipping let it make an angle of θ with the horizontal.

The forces acting on the ladder are shown in Fig. 3.3 The normal ground reaction is R_F, and the frictional resisting force μR_F is $\frac{R_F}{5}$. The normal wall reaction is R_w, hence the frictional resisting force is μRw or $\frac{R_w}{5}$. The directions of the frictional resisting forces are so as to oppose motion.

Under the general conditions of equilibrium of the forces acting on the ladder,

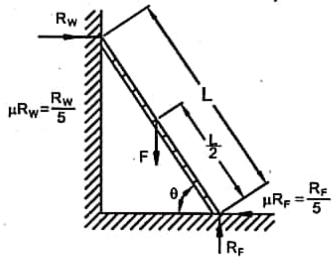


Fig. 3.3 Reaction at floor and wall by a ladder.

Upwards forces	= Downward forces	
$R_F + \frac{R_W}{5}$	= F	--- (1)
Forces to left	= Forces to right	
$\frac{R_F}{5}$	$= R_w$	--- (2)

Let us take moments about any convenient point.

Clockwise moments = Anticlockwise moments.

Taking moments about the foot of the ladder.

$$R_w \times L\sin\theta + \frac{R_W}{5} \times L\cos\theta = F \times \frac{L\cos\theta}{2} \quad \text{--- (3)}$$

From the three equations we have to eliminate all values of unknowns other than functions of θ

From equation (2) $R_w = \frac{R_F}{5}$, $R_F = 5Rw$

Substitute this value in equation (1)

$$F = R_F + \frac{R_w}{5}$$

$$= 5R_w + \frac{R_w}{5}$$

$$= \frac{26Rw}{5}$$

Substitute this value in equation (3)

$$R_W \times L \sin\theta + \frac{R_W}{5} \times L \cos\theta = \frac{13^{R_w}}{5} \times L \cos\theta$$

Divide all through by $R_w L$

$$\sin\theta + \frac{\cos\theta}{5} = \frac{13\cos\theta}{5}$$

$$\therefore \quad \sin\theta = \frac{12\cos\theta}{5}$$

Hence $\frac{\sin\theta}{\cos\theta} = \frac{12}{5}$

$\therefore \tan\theta = 2.4$

Hence $\theta = 67° 23'$

Answer : Ladder makes 67° 23' with the horizontal.

Friction in bearings :

The friction that a shaft encounters when moving in a bearing (Fig. 3.4) can be calculated as follows :

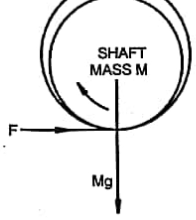


Fig. 3.4

Suppose the shaft is of mass M kg; the effect of gravity on mass M is Mg, *i.e.* the shaft has a weight of Mg. A frictional resistance F' acting tangentially at the point of contact opposes the motion of the rotating shaft and the normal reaction between the shaft and the bearing surface is equal to the force due to gravity *i.e.* Mg.

Frictional resistance $= \mu \times$ Normal reaction $= \mu$ Mg.

If the shaft has a diameter of d metres, frictional torque T, opposing

$$\text{motion} = \frac{d}{2} \times \mu\,Mg = \frac{d\mu\,Mg}{2}$$

and the power, P, absorbed in friction = T ω Watts

$$= \frac{d\,\mu\,Mg\,\omega}{2}$$

Where P = Power consumed at journal bearing (watts)

μ = coefficient of friction for given conditions (a pure number)

M = mass supported by bearings (kg).

g = acceleration due to gravity, (metres/sec^2).

d = dia. of shaft (metres)

ω = rotational speed (rad/s)

If the weight is given in Newtons as W

$$P = \frac{d\,\mu\,W\,\omega}{2}$$

When a shaft rotates the loss of work by friction is converted into heat energy.

Example :

A shaft, 50mm diam. rotating in journal bearing at 1500 r.p.m. is carrying a load of 2000 Newtons. Find the heat generated per minute if the coefficient of friction at the journal bearings is 0.03.

Ans. :

Frictional torque $= \mu \times W \times r$

$$= 0.03 \times 2000 \times \frac{0.050}{2}$$

$= 1.50$ Nm

Angle turned through in 1 min

$= 2\pi \times 1500 = 9420$ radians

work lost in friction per minute is

$T \times \theta = 1.50 \times 9.420 = 14130$ J

Since the mechanical equivalent of heat is 1 kCal = 4186.8 J, the

$$\text{heat generated per min} = \frac{14130}{4186.8}$$

$= 3.37$ kCal.

Travel of a body up or down on inclined plane :

Suppose, a body (e.g. a train of mine cars) of weight W, is being pulled by a haulage engine up a haulage plane at V m/s and that the coefficient of friction is μ. At small velocities the wind resistance is neglected. The work done per second consists of

1. Work done against gravity, and
2. Work done against friction.

The work done in (1) is force multiplied by distance and the force is W sin θ. The force of friction is μ W cos θ. the work done per sec. is = (W sin θ + μ W cos θ) $\times$ V units.

In mining problems, the inclination is generally gives as 1 in x, rather than in degrees, and $\frac{1}{x} = \tan\theta$. The discrepancy between tan θ and sin θ is usually ignored unless the inclination is steep, so that component of load due to gravity is taken as

$$W \tan\theta = \frac{W}{x}$$

Bearings :

A rotating shaft is supported or mounted on components which are called bearings, journal bearings, pedestal bearings, plummer blocks, etc. A bearing should offer as little friction or resistance to the rotation of the shaft as possible. There are various types of bearings.

A bush bearing consists of a cast iron or steel housing which houses a brass of gun metal bush and the rotating shaft rests inside the bush which is of plane cylindrical shape. A bush bearing is used for a small diameter shaft (up to 75 mm diameter) rotating at high speed. The bust is a tight driven fit into the bore of the housing. To secure it in position and to prevent it from rotatig a set screw is fitted into a hole driven partly in the bush and partly in the housing and the hole is threaded by "tap" to receive the said screw. An oil hole in the housing and extending through the bush is used for adding lubricating oil. An example is the bush bearing of the dynamo of a motor car.

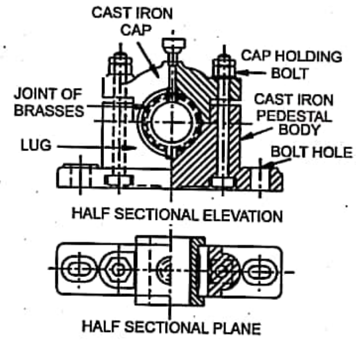


Fig. 3.5 Plummer block.

A pedestal or plummer block is a form of bearing in which the brasses are formed of two halves (Fig. 3.6) which are housed in pedestal body of cas iron as shown in Fig. 3.5. A cast iron cap is bolted on the pedestal body to cover the top brass. To prevent rotation of the brasses within the block a lug is provided in each half brass and the former fits into a recess formed in the pedestal body. The face along which the two brasses rest against each other,

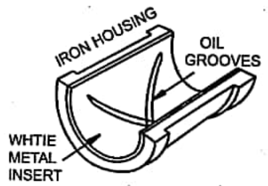


Fig. 3.6 The "brass" of a plummer block.

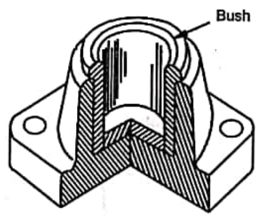


Fig. 3.7 Foot-step bearing.

i.e. the joint, is parallel to its length of the shaft. A clearance is provided between the C. I cap and C.I. pedestal body. When the brasses wear in course of use they are dismantled, after removing the C.I. cap, the joint faces filed down to compensate for the wear, and the reassembled. Lubricating oil is added through the hole in C.I. cap and goen through the hole in the upper brass.

The brasses have oil grooves which accommodate a small quantity of lubricating oil and distribute it from the central feed point tot the whole surface of the brasses. A plummer block is used for shafts of large size (usually exceeding 50 mm dia.) subjected to heavy loads and rotating at low speed. The part of the shaft which rests on the bearing known as jouranl, is usually shouldered to prevent endwise movement.

Foot-step bearing :

It is used to support the end of a vertical shaft. In the mines one may observe this type of bearing on the tension bogey of an endless rope haulage. It consists of a C.I. housing with a bushed hole to receive the shaft. The bottom of the hole is fitted with a brass or gun metal pad with a concave upper surface and the bottom end of the shaft is suitably convex shaped. Lugs are provided on the pad and the bush to prevent their rotation.

Ball and roller bearings :

These differ from the bearings described so far in that the rotating shaft is separated from the statinary portion by means of intermediate balls or rollers which roll freely between the two surfaces. In this case, no oil film is present to spearate the faces at the points or lines of contact, but as there is no slipping of continuous surfaces, wear does not normally take place. The friction of bused bearings is greater at starting than when running. In ball and roller bearings there is rolling friction between the surfaces in contact and not sliding friction so that their coefficient of friction is as low as 0.001 and only very little higher at the start than when running.

A ball bearing for a revolving shaft (Fig. 3.8) consists essentially of four main parts :

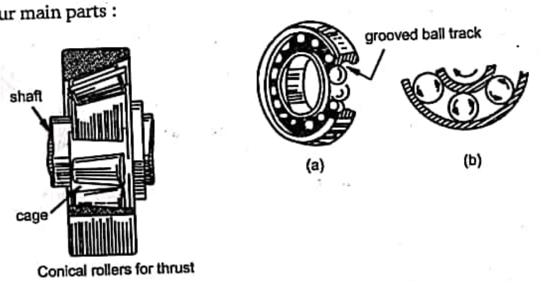


Fig. 3.8 (ii) (b) : Section of a ball bearing showin the relative motion of the parts in a ball bearing.

1. An inner race which is made a tight fit on the shaft and revolves with it. The race is held axially by a shoulder at one side and a nut at the other.
2. An outer race which is stationary and is usually made a push fit into the enclosing housing.
3. A row of balls (or sometimes two rows) running in grooves in the two races, the radius of the grooves being 5% to 10% greater than that of the balls to prevent sliding friction.
4. Cage or separator, to separate the balls and maintain them at the correct distance apart.

   The races and balls are of high-carbon chrome steel having hard surfaces. They are machined and ground to within limits of 0.0002cm, and highly polished. The cage may be of steel, or brass or a synthetic resin material.

Although there is no actual oil film betwen the points of contact of balls and races, lubrication by a pure mineral grease is necessary to protect the polished surfaces from rust, to help in the exclusion of foreign matter, and to lubricate the cage where it is in contact with the balls.

Roller bearings are similar in construction to ball bearings except that rollers take the place of the balls. A roller bearing has the advantage that it can carry about twice the load of a corresponding ball bearing, and it is able to move axially within its races to adjust for slight variations in shaft length rather more careful mounting than a ball bearing in order to maintain accurate line contact. Ball bearings are therefore generally preferred where the loads are within their capacity whilst roller berings are used for the heavier duties. Sometimes, a roller bearing is used at the driving end of a shaft and a ball bearing at the other end.

Silent block bearing : Where a shaft does not rotate but oscillates through a small angle, generally upo 20° on either side of a mid positin, silent block bearing is used, *e.g.* bearing for the ends of a leaf spring in a motor car, shaking screens *etc.* It consists of an inner steel bush and an outer steel tube with a rubber bush as a filling between the two. The outer tube is attached to the stationary member of the car, screen or machine. Such bearing needs lubrication and is noiseless in operations. The inner steel bush can twist on either side of a mid-position relative to the outer fixed steel tube. (Fig. 3.9).

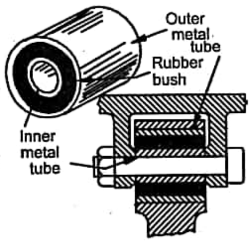


Fig. 3.9 Silent block bearing

Thrust bearing :

When a shaft is subjected to a load or pressure in a direction parallel to its axis, the ordinary journal bearing is not effective and it has to be provided with some arrangements to absorb the axial thrust. Such bearing is called a *thrust bearing* and one of the machines where it is used in mines is the turbine pump. It has been described in the chaper on pumping.

Special bearings :

Although most bearing are provided with brass or white metal lining and lubricated with oil or grease, other material are applied for certain special duties. Some bore hole pumps are made with the driving motor on the surface and the rotary pump at the bottom of the bore hole, the two being connected by a shaft running the length of the bore hold. This shaft may be run in bearings of lignum vitae, a very hard tropical wood, lubrication being effected by the water rising up the hole. Lignum vitae is also used for corrosive water which would destroy ordinary metallic bearings. Plastics are also employed for bearings in some machines and they give good services.

Lubrication :

The function of a lubricant is to maintain a continuous film of oil between the bearing surfaces, so eliminating solid friction and substituting in its place fluid-friction within throughout. The viscosity of the lubricant should be as low as possible in order to reduce the internal opposition of its particles to move over one another, but it should be sufficiently high to prevent metallic contact. The particular type to use in any given case depends on the load to be carried and the speed of the journal. High speeds and light loads require thin oils; heavy loads and low speeds require an oil or grease of high viscosity. Uniform distribution of the lubricant is essential, and oil-ways or grooves should be made where necessary to ensure This.

There are several system available for the lubrication of machinery, the simplest being hand oiling for small unimportant bearings; this is done manually with on oil can. For continuously running machines the methods of lubrication are as follows :

(a) Drip-feed oiling : in which a small amount of oil is regularly supplied to the bearing, as with siphon lubricators and sight-feed drip oilers. The Siphon lubricator consists of a small brass oil box into which a cotton wick dips. Oil rises in the wick by capailliary action Whick's other end is at a lower level touching the bearing surface in contact with the shaft. The wick is removed when the engine is stopped or otherwise the dripping oil is wasted. The sight feed drip oiler, of which the needle lubricator is an example, is shown in Fig. 3.10. The needle lubricator consists of a glass vessel filled with oil, closed by a wooden plug and inverted. A "needle" fits freely in the plug and rests on the shaft. When the latter is in motion "needle" is vibrated, so permitting oil to trickle down. The device ceases to function when the shaft is stationary.

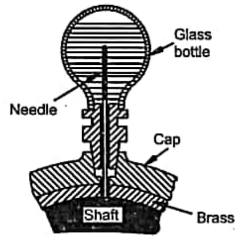


Fig. 3.10 Needle lubricator

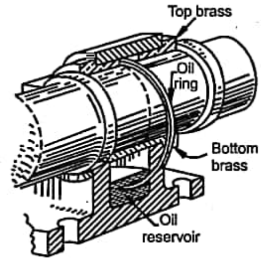


Fig. 3.11 Ring oiled bearing

(b) The Stauffer lubricator is a box filled with grease and fitted with a cap which can be screwed down as required, so pressing the grease on to the bearing. It can work in an inverted position and is suitable for low-speed work in dirty and dusty situations.

(c) Ring oiling for the bearings of large electrical generators and motors in which oil is distributed by a metal ring or rings hanging loosely from the shaft and dipping into a small oil well formed in the housing of the bearing (Fig. 3.11). To accomodate the oil rings slots are cut in the upper brass. When the shaft revolves the rings rotate with it, carrying up a little quantity of oil with them and this oil flows over the journal and back into the sump. When the shaft stops the oil supply to it also stops.

(d) Splash oiling : for groups of bearings enclosed in a casing, as in high speed steam engines, gear boxes, etc. the oil immersed gears should not be run with the gear box full of oil. Otherwise there will be a serious loss of power. The correct level of oil, is generally below the centre line of the gears and is marked on the cover of the gear box. In splash lubrication the bottom portion of the gear always remains below the oil level and when working they splash the oil round the other gears.

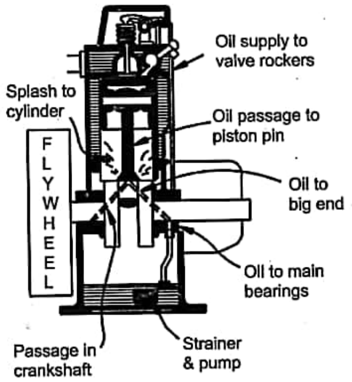


Fig. 3.11 (A) Forced feed lubrication

(e) Forced feed lubrication : for vertical, high speed enclosed type steam, gas, or oil engines, steam turbines, etc, Forced feed lubrication is a standard practice for high speed engines, the oil being pumped to the crankshaft bearings, crank and gudgeon pins, etc. through holes drilled in the parts themselves. The oil finds its way back to the sump through a strainer, and is used over and over again. A familiar example of force feed lubrication is the engine of motor car. (Fig. 3.11 A).

(f) Injection lubrication as in some types of rotary compressors.

Inclined plane :

An inclined plane or a ramp is made use of to reduce the force required to raise heavy objects. Fig. 3.12 shows a heavy drum of 100 kg to be placed in a truck body 1 m above the ground level. The mechanism advantage of the inclined plane ignoring friction is found by dividing its length (horizontal) by its height. In the Fig. 3.12 it is 3 which means that the force required to push it up the plane is 1/3 rd of that needed to lift it vertically, Longer the plane, larger is the mechanical advantage and less is the force required to push up the load.

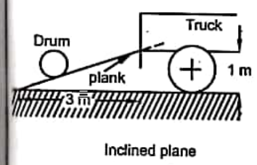


Fig. 3.12 Inclined plane

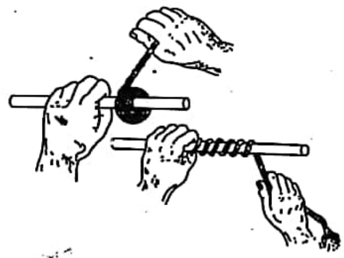


Fig. 3.13.

Screw :

The screw makes use of the principle of inclined plane. The motion of the nut relative to the thread is similar to that of a load sliding on an inclined plane. Tightening the nut is equivalent to pushing the load on the plane while loosening is equivalent to pushing it down the plane. A screw is formed by running an inclined plane around a cylinder. Make a right angled traingle ABC from a piece of paper and wrap it round a thick pencil as shown in Fig. 3.14. The spiral inclined plane so formed represents a screw. When a particle describes a circular path at a uniform speed, at the same time travelling in the axial direction at uniform rate, it traces out a curve whic is known as the *helix*. The height of this spiral path for one turn is the pitch of the screw (EF to E'F').

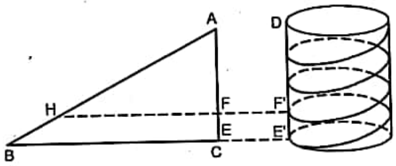


Fig. 3.14

Another example makes the understanding of a screw thread clear. If a thin rope is coiled on a thick pencil to keep the turns of the rope normal to the pencil, the turns will pile up on above the order. Instead, if the rope is coiled, not normal but an angle to the pencil, the rope will proceed forward on the pencil (See Fig. 3.13.).

If only one thread is "wrapped" round the cylinder, the screw formed has *single-start* threads an such single-start threads are common on most of the screws. Some screws have double-start and some have treble-start threads. In a double-start screw there are two continous parallel threads which is equivalent to two inclined planes wrapped round the cylinder. One may notice double-start threads on a fountain pen having a screwed cap indicated by two starting points of the threads. In one turn of the screw the thread advances an axial distance called the *lead* of the screw. (Fig. 3.15). This is not to be confused with the *pitch* of the screw which is the distance, measured axially, between two adjacent threads. With single-start threads the pitch is equal to the lead but with double-start threads the lead is equal to twice the pitch and with treble-start, the lead is equal to thrice the pitch. Heavy duty threads as used on lifting jacks and similar machines are often multi-start. The main advantage of multi-start threads is that travel of the nut (screw remaining stationary) in the axial direction is large with a single turn of the nut, as compared with the single start thread. A lifting jack can therefore be set in its place to lift a load with only a few turns.

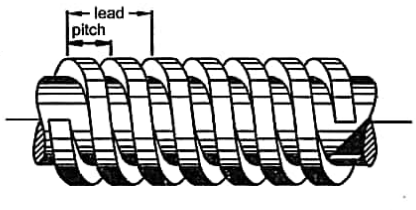


Fig. 3.15 A two-start bolt showing lead and pitch

A screw is said to be **right handed** or **left handed** depending upon the direction of threads. On a right handed screw a nut travels away from the observer if he rotates the nut clockwise keeping the screw stationary; on the left handed screw it is opposite, *i.e.* the nut travels towards the observer when he rotates the nut clockwise. (Fig. 3.16).

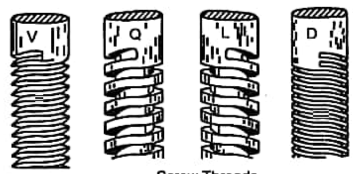


Fig. 3.16 Different types of screw threads.
V–Veetype, Rt. hand. Q–Squarethread,
Rt-hand, L – Square thread, L-H.
D-Two-start, square thread, L.H.

Forms of screw thread :

The screw threads are divided into two main classes, the angular (also called triangular) and the square thread. Other forms are either modified forms of these or a combination of the two forms. A thread cut on the surface of a screw is called an external thread while that cut in a hole, is called an internal thread. The Vee thread is stronger and grips more tightly than the square thread and is used for holding parts tightly together. The square thread works with less friction, is more efficient and is used for transmitting power, *e.g.* in a screw jack, in a lathe, etc. The commonly used threads in the industry are designated as follows :

Whitworth thread : The Whitworth screw thread which is of Vee type, is the standard form adopted in the British Industry. It is also known as British Standard Whitworth (B.S.W.) thread. It has an angle of 55°.

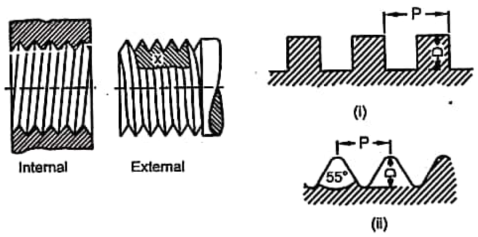


Fig. 3.17 Commonly used screw threads.

(i) Sq. thread D = 0.5 P *(b) V-thread Whitworth type = D 0.64 P.*

British Standard Find Thread (B.S.F.) :

Like the B.S.F. it has the same Whitworth profile but the pitch is finer and hence the depth smaller. B.S.P. threads are used for gas, steam, or water pipes. They are specified by the bore of the pipe and not by the outside diameter. Thus the outside diameter of the threaded pipe having a bore of a 1inch nominal diameter is 1.309". Pipes of 1" to 6" diameter have the same number of threads per inch, viz 11.

In general engineering, the hexagonal head of a bolt and the hexagonal nut bear a definite relationship to the diameter of the threaded portion of the bolt. A 1/4-inch spanner is one which fits the flats on the head of the nut of a 1/4-inch dia. bolt.

S.O./I.S.I. Metric Screw Threads :

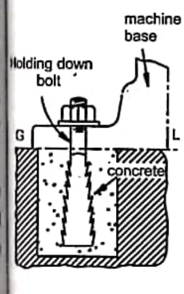
Form of thread	Outer diam. (inches)	No of threads per inch.	I.S.I. metric thread		
			designation no.	Pitch mm.	Core dia. mm.
B.S.W.	1/8	40	6	1.0	4.8
	1/4	2	8	1.25	6.5
	1/2	12	10	1.5	8.2
	1	8	16	2.0	13.5
B.S.F.	1/4	26	20	2.5	16.9
	3/8	20	24	3.0	20.3
	1/2	16	30	3.5	25.7

A screw has many uses, such as fixing, clamping, positioning devices in machine tools, raising and lowering loads in the form of a screw jack, etc. Bolt is the word used for a screw usually of large diameter and receiving nut at one end.

A holding down bolt is one which holds a part or component of a machine firm on the ground or wall. Such holding down bolt is generally embedded in concrete and clamped to the machine part by a nut (Fig. 3.18).

Prevention of loosening of nuts :

A nut becomes loose or slack on a bolt which is subject to vibration or frequent movement *e.g.* on the reciprocating screens, pumps, reciprocating engines. etc. Various devices are used to ensure that the nut remains tight on



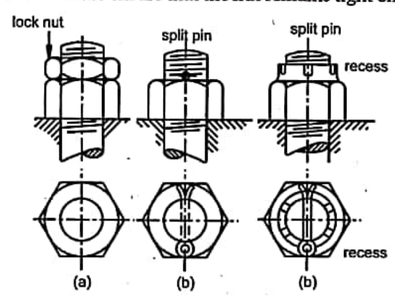


Fig. 3.18 Holding down bolt. *Fig. 3.19 Locking a nut in position.*

the bolt. Fig. 3.19 (a) shows a lock nut which is usually half the height of ordinary nut. The ordinary nut is tightened on the bolt and then the lock nut, after holding the ordinary nut firmly by a spanner. Fig. 3.19 (b) shows a split pin which is inserted into the bolt *after* drilling a small hole into the latter once the nut is firmly tightened in its place. Often, the drilling after placing the nut in position, is not possible and a bolt as well as nut with a through hole are selected. The disadvantage of this is that when the nut is tight the holds may not coincide. A small differences in alignment can be made up by placing a washer beneath the nut before tightening the latter. Fig. 3.19 (c) shows an especially formed nut used in conjunction with split pins. The special nut is called a castle nut and has a slot at the top on each of its 6 faces.The bolt has a small hole to receive the split pin. Such nut provides 3 different positions in which the split pin can be placed for the hole in the nut.

Fig. 3.20 (top) shows a self-locking nut with a fibre, plastic or nylon insert. Such nuts incorportating a special moulded nylon insert, the internal diameter of which is similar by a predetermined degree than the bolt thread, has a good gripping power. As the nut is tightened, the threads of the bolt force into the nylon insert to form a thread. At the same time this resistance to the entry of the bolt forces the nut away from the bolt head until the working sides of the nut and bolt threads are in close contact. As the nylon insert is highly compressed it grips the bolt with a sustained and considerable inward pressure, making the bolt and nut totally vibration free. Hence, unlike an ordinary nut, a nylon self-locking nut can never work loose. Once such nut in the market has the trade name *Nyloc self-locking nut.* Fig. 3.20 (middle) shows a spring washer placed below the nut. A spring washer is a small helical spring with a split. The spring washer, when placed in position on the bolt and the nut screwed on it, exerts a pressure against the nut and prevents the latter from turning and working loose. Some spring washers have hardened points formed at the position of the "split" in the washer. The lower point of the washer tends to dig, into the workpiece and the upper point in the underside of the nut. This, in addition to reaction of the washer,

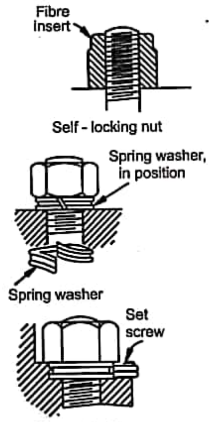


Fig. 3.20

Top – A locking nut with fibre insert
Middle – A spring washer to lock a nut
Bottom – A grooved nut with a set screw

helps to keep thenut in its position. Fig. 3.20 (bottom) shows a grooved nut. Such nuts are somewhat expensive but are used in special jobs such as, connecting rod ends of marine engines.

Fig. 3.21 shows a tab-washer. A tab-washer is a plain washer with two or more "tabs" or "lips" projecting from the outer edges. After tightening the nut over the washer one of the tabs is bent upwards and the other, downwards, and flattened against the side of the work piece. The tab washers are usually thinner than ordinary plain flat washers.

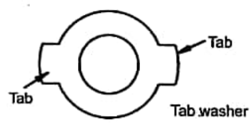


Fig.3.21

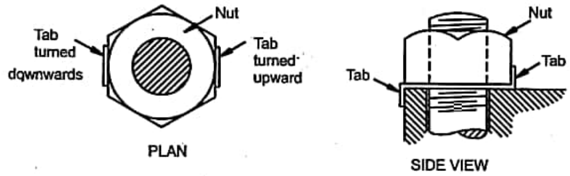


Fig. 3.21 Tab washer a lock a nut.

QUESTIONS

1. What is friction ? State three common examples in everyday life where friction is a disadvantage and other three examples where it is an advantage in the operation of machines
2. Write short notes on : ball bearings, angle of repose, needle lubricator, holding down bolt.
3. Explain the terms *lead* and *pitch* in relation to a screw thread. What is a single-start thread and a double-start thread ?

4. What are the different arrangements for preventing the loosening of nuts on bolts due to vibration ? Explain two such arrangments with sketches.

5. What is the purpose of lubrication ? Describe with sketh (i) a plummer block (ii) arrangement for splash oiling.

6. What are the different types of threads in common use on machines ? Explain the Whitworth thread, square thread and nuckle thread with sketches.

7. A ladder having a length of 8m and a mass of 30kg leans against a smooth vertical wall while its lower end rests on a rough horizontal surface. When the ladder is inclined at 58° to the horizontal it is just on the point of slipping. Assuming the gravitational force on the ladder to act at its mid-length, determine the limiting coefficient of friction.

(Ans. : 0.312)

O O O

CHAPTER – 4

SIMPLE MACHINES : LEVERS, PULLEYS, LIFTING MACHINES

A machine is any contrivance, apparatus or device in which a force or effort that is applied to one of the parts is transmitted to and overcomes a resistance or load acting at some other part of the machine. Usually the force applied is much smaller than the resistance to overcome and the ratio

$\frac{load}{effort} = or \frac{resistance\ to\ overcome}{force\ applied}$ is called the mechanical advantage.

All machines are based on the simple principle of conservation of energy which states that energy cannot be created or destroyed but can only be changed form one from into another. A machine can therefore be considered as an energy-changing device.

For a machine :

$$\text{velocity ratio} = \frac{\text{distance moved by effort}}{\text{distance moved by load}}$$

The basic machines are the lever, the simple pulley, the inclined plane and a combination of these.

Levers :

The working of the lever is based on the principle of moments, viz.

i. The turning moment of a force about a point is the product of the force and the perpendicular distance from the point to the line of action of the force.

ii. When the turning moments of two forces about the fulcrum of a lever are equal and opposite, the lever is in equilibrium.

The lever is a simple machine which uses the principle of moments in order to raise a load of large magnitude by applying a small force. The lever turns about a fixed point called fulcrum. The rod or bar is a simple example of a lever which is probably one of the earlist machines to be used by man and today, in various forms and combinations, it is the most widely used of all the machines.

The lever itself must be rigid. It may be straight, curved, angled or offset. If a lever has a sharp angle at the fulcrum, it is called a bell crank. (See pg. 39). A lever which consists of one bar having one fulcrum is known as a simple lever, *e.g.* Fig. 4.2

The small load P applied on the lever, either at one of the ends or somwhere along its length, to raise a large load W_1 is called effort.

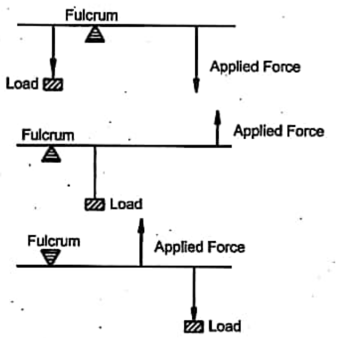


Fig. 4.1 Simple lever
Top – Lever of first order
Middle – lever of second order.
Bottom – lever of third order.

Levers are divided into 3 classes :

Class–1 :

Fulcrum between load and effort. Examples – a crowbar lifting a derailed coal tub, a beam balance, a pair of seissors, the handle of a hand pump for water, *etc.*

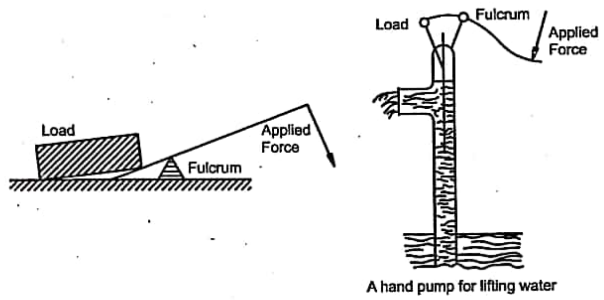


Fig. 4.2 levers of first order. A crowbar used to lift a heavy stone or load, (above)

Class–2 :

Load between fulcrum and effort. Examples – a pair of nut crackers, a wheel barrow, the safety valve (lever type) on a Lancashire boiler, sylvester prop withdrawer, *etc.*

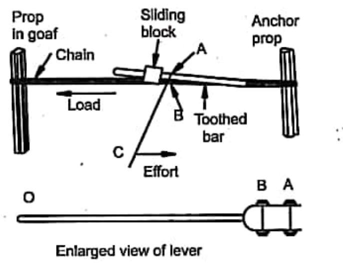


Fig. 4.3 lever of second order. A Sylvester prop withdrawer used in mines.

The load acts at B and A is the fulcrum

Class–3 :

Effort between fulcrum and load. Examples – the forearm of a human body, a pair of sugar tongs, etc.

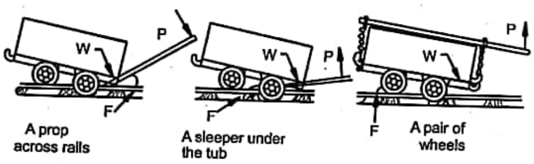


Fig. 4.4 A derailed tub being re-railed.

F is the fulcrum represented by :
left – a prop across rails (lever of first order)
middle – a sleeper under the tub (lever of second order)
right – a pair of wheels (lever of second order)

Compound levers :

A lever, which consists of a number of simple levers, is known as a compound lever.

A number of mechansims work by a combination of levers. A familiar example is the slot machine in some shops where a person stands on its platform inserts a coin and out comes a card with the weight of the person typed on it. The striking mechanism of a typewriter is an example of a combination of levers of the first and second orders.

A lever safety valve shown in Fig. 1.28 in chapter 1 is an example of a lever of second order in which the steam pressure represents the applied effort.

Fig. 4.5 shows a plier of wire cutters cutting a piece of wire. The hand exerts a force of 60 N. The resistance to cutting F can be calculated as follows.

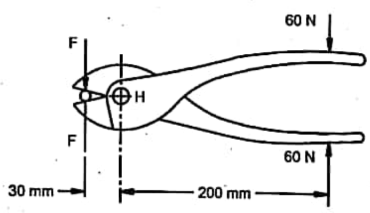


Fig. 4.5 A plier of wire cutters.

The wire cutters represent a double lever. We need to consider one half only, keeping in mined that equal and opposite forces are acting.

Taking moments about H

$$30\ F = 60 \times 200$$

$$\therefore \quad F = \frac{60 \times 200}{30} = 400$$

The resistance to cutting is a force of 400 N

If a machine is so constructed that there are two or three inter-connected levers, the mechanical advatage of the combined system of levers is :

Mech. advantage or leverage : Mech. adv. of one lever × mech. adv. of second lever × mech. adv. of third lever, and so on.

Example :

A compound lever shown in Fig. 4.6 is required to lift a heavy load W. All dimensions are in mm. Find the value of W if an effort of 100 N is applied at A.

Ans. :

From the geometry of the lever, we find that the leverage of the upper lever AB

$$= \frac{AF_1}{BF_1} = \frac{275}{25} = 11$$

Similarly leverage of the lower lever CF_2

$$= \frac{CF_2}{DF_2} = \frac{337.5 + 22.5}{22.5} = 16$$

$\therefore$ Total leverage of the compound lever

$$= 11 \times 16 = 176$$

We know that leverage

$$176 = \frac{W}{P} = \frac{W}{100}$$

$\therefore \quad W = 100 \times 176 = 17600$ N.

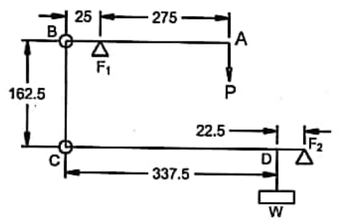


Fig. 4.6

The manual typewriter is a machine consisting of a combination of interconnected levers. Because of the large mechanical advantage available, a small effort of the finger causes the type to strike with a large force on the paper. Another example : steering gear lever in motor cars.

Pulley :

A pulley may be of metal or wood and it revolves about an axis at its centre. It may be either smooth on its circumference to receive a flat belt or it may be grooved to receive one or more ropes or V-belt.

A single fixed pulley is used to merely deflect the line of action of a pulling force (Fig. 4.7). The mechanical advantage of such a pulley is unity, as effort is equal to load, neglecting friction. The distance moved by the effort is equal to that through which the load travels so that velocity ratio, which is defined as $\frac{\text{distance moved by effort}}{\text{distance moved by load}} = 1$

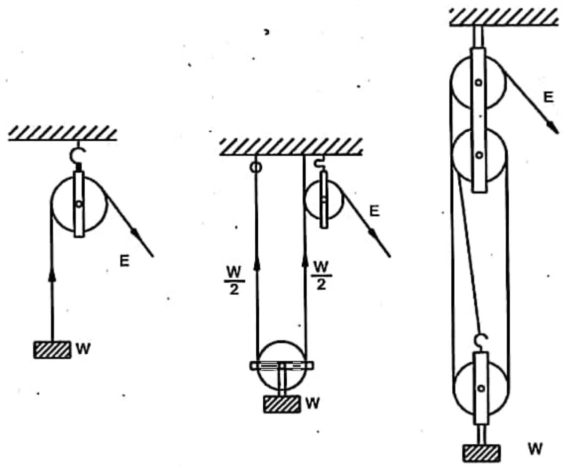


Fig. 4.7 *Fig. 4.8* *Fig. 4.9*

Fig. 4.8 shows a fixed pully to change line of action of effort and a movable pulley which supports the load W. The movable pulley is called "snatch block" and always forms part of the tackle of any pulley block system. The load W is supported by two lines of strings so that each line carries a load or tension of $\frac{W}{2}$ and efffort E applied is equal to $\frac{W}{2}$. Neglecting friction mechanical advantage = $\frac{\text{load}}{\text{effort}} = 2$.

If E travels due to a pull through a distance s, each side of the loop of the strings will be shortened by $\frac{s}{2}$ and the load is lifted vertically through $\frac{s}{2}$.

$$\text{Velocity ratio} = \frac{\text{distance moved by effort}}{\text{distance moved by load}} = \frac{s}{s/2} = 2.$$

The fixed end of the string may be attached to a fixed beam as shown in the figure or to any point on the bracket of the fixed pulley.

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In the above case, where the number of pulleys is two *i.e.* one fixed and one moving the V.R. = 2. If there are n pulleys of equal diameter the V.R. = n. Here we have assumed frictionless pulleys and weightless ropes and pulley blocks. The value of V.R. will not change when friction and weight of the pulley blocks and ropes are considered but friction will affect the mechanical advantage and therefore the mechanical efficiency of any lifting tackle.

For any machine

$$\text{Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{W \times S}{E \times s} = \frac{W}{E} \times \frac{S}{s}$$

where s is travel of effort and S is travel of load

$$\therefore \quad \text{Efficiency} = \frac{\text{Mechanical advantage}}{\text{Velocity ratio}}$$

Fig. 4.9 shows a pulley block of 3 pulleys. The load is supported by 3 lines of ropes or strings so that V.R. = 3. The mechanical advantage will however, not be 3, but somewhat less on account of friction and weight of brackets of the movable pulley.

Fig. 4.10 shows a pulley block or block and tackle used to lift heavy loads. In the Fig. the lifting tackle consists of two blocks, the upper one suspended from a beam or support and the lower, movable one, having a hook to lift the load W. The upper block has 3 pulleys which run side by side on a common axle. The lower block has also equal number of pulleys which run side by side on a common axle. The weight is, therefore, supported by six lines of one rope which has one end fixed to the top block and the other end free where the effort is applied. If the load is lifted one metre, then each of the six parts of the rope supporting the load will shorten by one metre. Thus the free end of the rope moves six meters and velocity ratio is six. The mechanical advantage will be

$$\text{M.A.} = \frac{500}{100} = 5$$

and the mechanical efficiency $= \frac{5}{6} \times 100\% = 83.3\%$

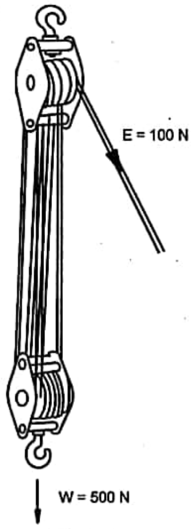
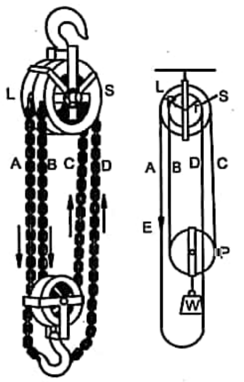


Fig. 4.10 Fig.



4.11 Westron pully block.

In most of the pulley blocks, the lower block has one pulley less than the upper one. In such case, the end of the rope must be secured to the lower block and the number of supporting lines of rope = 2p + 1 where p = number of pulleys in the lower block. The V.R. is, therefore,

V.R. = 2p + 1.

The practical limit to any system of pulley block is the large quantity of rope and the head room required to raise the load.

Weston differential pulley-block :

In this pulley-block known as the Weston pulley-block, the top pulley is made in two parts, one S, a little smaller than the other L. An endless chain passes round the larger pulley L, then round the snatch-block, P, and finally round the smaller pulley, S, of the block. The effort, E, is applied to the chain at the larger radius as shown in the Fig.

If R and r be respectively the radii of the larger and the smaller pulleys, when the pulley makes one complete revolution, E moves through $2\pi R$. A length $2\pi R$ of the chain moves up the larger pulley; at the same time a length $2\pi r$ of the chain moves down the smaller pulley. A net length $(2\pi R - 2\pi r)$ of chain is therefore obtained by W going of by half the amount is $\frac{1}{2}(2\pi R - 2\pi r)$ The velocity ratio of the Weston pulley block is, thus,

$$V.R. = \frac{2\pi R}{\frac{1}{2}(2\pi R - 2\pi r)} = \frac{2R}{(R-r)} = \frac{2D}{(D-d)} \text{ in terms of diameters.}$$

In practice the pulleys are manufactured with 'flats' or 'beds' in which the links of the chain fit so that the chain does not slip. The number of flats on each pulley is related to the diameter of the pulley. Thus it is simple to count the number of flats on each pulley to measure the radius or diameter. The number of flats on large and small pulleys are often two consecutive numbers, thus making the velocity ratio as high as possible.

Example :

A Weston differential pulley is to lift a mass of 1000 kg and the efficiency under this condition is 46%. The number of flats on the top pulleys are 12 and 11. Determine the effort required.

Ans. :

Velocity ratio $= 2 \times 12 = 24$

Mechanical advantage = velocity ratio × efficiency

$= 24 \times 0.46 = 11.04$

The mass lifted is 1000 kg, hence

$$\frac{9.81 \times 10^3 N}{effort\ (N)} = 11.04$$

Effort = 9810/11.04 = 889 N.

- The friction of Weston Pulley is very great and, in consequence, its efficiency is usually very low, about 0.4. Because of the large friction, the machine cannot run backwards. This means that, if the force E acting at A is removed, the weight W does not fall, and this is in advantage of high friction, though at the cost of efficiency. The Weston pully block is, therefore, a self-locking machine.

The wheel and axle :

If a pair of wheels of drums of different sizes and turning about a common axis are arranged as shown in Fig. 4.12 they will enable a load to be raised through any desired height with the aid of two ropes.

The load W is attached to a rope B coiled around the smaller drum or axle S, and the effort E is applied to the end of a rope A coiled in the opposite direction around the larger wheel or drum L.

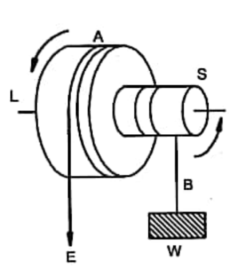


Fig. 4.12 Wheel and axle

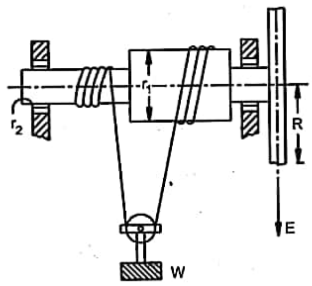


Fig. 4.13 Differential wheel and axle.

It will be clear that in one revolution, the effort E will travel through a distance which is proportional to the larger radius, R, whilst, in same time, the load W will be raised through a distance proportional to the smaller radius, r.

$$\text{Velocity ratio} = \frac{\text{distance moved by effort}}{\text{distance moved by load}}$$

$$= \frac{\text{radius of larger wheel}}{\text{radius of smaller wheel}} = \frac{R}{r}$$

If the machine were 100% efficient this would also represent the mechanical advantage, or the ratio $\frac{\text{Load}}{\text{effort}} = \frac{W}{E}$

Actual M.A. = V. R. × efficiency of the machine.

Differential Wheel and Axle :

In a simple wheel and axle where the wheel and axle are rigidly connected, when the wheel makes one revolution, the axle also makes one revolution. The velocity ratio is, therefore $\frac{R}{r}$. Using the differential principle, the axle is made in two parts of radii r_1 and r_2, r_1 being slightly bigger than r_2, and the lifting rope is so arranged that while it gets wrapped round the larger axle, it un-wraps at the same time from the smaller axle. Referring to Fig. 4.13, let the wheel of radius R makes one revolution under the effort E. The displacement of the effort is $2\pi R$. Both parts of the axle have also made one revolution. But, while a length $2\pi r_1$ of rope gets wrapped round the larger axle, a length $2\pi r_2$ gets unwrapped at the same time. A net length of $(2\pi r_1 - 2\pi r_2)$ has therefore been obtained by W going up by half the amount, i.e. by $\frac{1}{2}(2\pi r_1 - 2\pi r_2)$. The velocity ratio of this arrangement is, therefore,

$$\text{V.R.} = \frac{2\pi R}{\frac{1}{2}(2\pi r_1 - 2\pi r_2)} = \frac{2R}{r_1 - r_2} = \frac{2D}{d_1 - d_2}$$ in terms of diametrs. Evidently, the smaller $(d_1 - d_2)$, the greater is the velocity ratio.

The double purchase which or crab :

This machine commonly used in mine for lifting loads, differes from a simple "wheel and axle" in that it incorporates double reduction toothed gearing (hence the name "double purchase") between the handle and the barrel, thus greatly increasing the mechanical advantage and enabling a heavier load to be lifted by a given effort, although in a longer time. A "single purchase" which has only a single reduction gearing, but otherwise its principle is the same and will be understood from the following example.

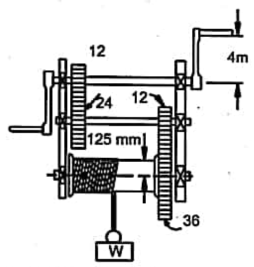
Example :

Two men exert a force of 200 Newtons each on the handles of a double purchase winch, which are 400 mm long. The driving pinions have 12 teeth each and the followers 24 and 36 teeth respectively. The diameter of barrel is 225mm and of the rope thereon, 25mm. Find the weight that can be raised, assuming efficiency of the winch 75%.

Ans. :

$$\text{Gear ratio} = \frac{T_1 \times T_2}{t_1 \times t_2} = \frac{24 \times 36}{12 \times 12} = 6 \text{ to } 1.$$

Hence for each revolution made by the barrel the handle makes 6 revolutions.



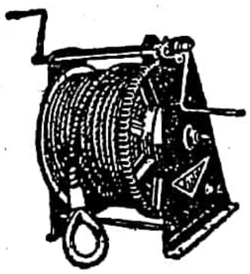


Fig. 4.14 Double purchase winch ***Fig.4.14 A crabwinch with single reduction gearing. (Single purchase)***

Let R = effective radius of barrel measured to the centre line of the rope = 125 mm.

L = Length or radius of the handles = 400mm

E = Effort or applied force = 2×200 N = 400 N

W = Load or weight raised in Newtons

Then work done on load during one revolution of barrel

$= W \times 2\pi R$ In the same time, work done by the effort

$= E \times 2\pi L \times 6$. By the principle of work, neglecting friction, we have

Work done on load = work done by effort

$$\therefore\ W \times 2\pi R = E \times 2\pi L \times 6 \ \text{ or } \ W = E \times \frac{L}{R} \times 6.$$

$$\text{Weight raised} = W = 400 \times \frac{400}{125} \times 6 \times 0.75 \ \text{(allowing for friction)}$$

$= 5760$ N.

Such example may be easily solved by utilising the following general rule.

$$\text{Mechanical advantage} = \frac{\text{Load}}{\text{Effort}}$$

$$= \frac{\text{Length of handle}}{\text{Effective radius of barrel}} \times \frac{T_1 \times T_2}{t_1 \times t_2} \times \text{efficiency}$$

$$\text{and Load} = \text{Effort} \times \frac{L}{R} \times \frac{T_1 \times T_2}{t_1 \times t_2} \times \text{efficiency}$$

The Screw Jack :

This machine uses the principle of screw to lift heavy weight by small effort. The effort is usually applied by means of an arm which increases the velocity ratio. The arm is a bar inserted into a hole on the cap or nut of the screw jack.

Consider a screw jack with a pitch of 6mm which is turned by applying an effort to a bar 15 cm long. (Fig. 4.15). Consider the screw with single-start threads.

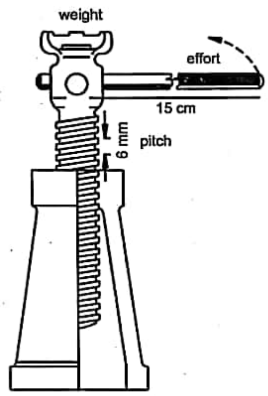
Calculation of V.R. :

When the bar turns through on revolution it moves through $2\pi \times 15$cm . Effort moves through $2\pi \times 15$cm in 1 rev.

The load is raised by 0.6 cm.

$$\text{So, V.R.} = \frac{2\pi \times 15}{0.6} = 50\pi \text{ or approx, } 150$$

Even though the friction may be high there will be a high M.A.
To reduce friction square thread are used instead of V-threads.



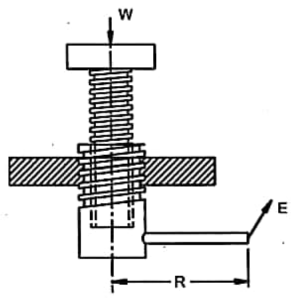


Fig. 4.15 A screw jack Fig. ***4.16 Differential screw jack***

Example :

A screw jack working with an efficiency of 25% has a velocity ratio of 60. What effort is needed to lift a load of 1 tef with the jack ?

Ans. :

$$1 \text{ tef} = 1{,}000 \text{ kgf}$$
$$= 9.81 \times 1{,}000\text{N}$$

Let the effort required be E Newtons.

$$\text{Percentage efficiency} = \frac{\text{M.A.}}{\text{V.R.}} \times 100$$

$$\therefore \quad 25 = \frac{\text{M.A.}}{60} \times 100$$

This gives M.A. = 15

$$\text{Now} \quad \text{M.A.} = \frac{\text{Load}}{\text{Effort}} = \frac{9810}{E}$$

$$\therefore \quad 15\,E = 9810 \text{ and } E = 654\text{ N}$$

Differential Screw Jack :

In the differential screw jack, the screw is in two parts, the smaller screw of pitch p can move up or down in the hollow groove of a larger screw of pitch P. The smaller screw does not rotate but moves only axially carrying the load 'W' with it (Fig. 4.16).

When the effort arm of radius R, makes on complete revolution, E moves through $2\pi R$ while the larger scrw moves up by its poitch length P. At the same time, W resting on the head of the non-rotating smaller screw moves down by the pitch length p of the smaller screw. the nett amount by which W moves up is therefore (P-p). The velocity ratio of the differential screw is, therefore

$$V.R. = \frac{2\pi R}{P-p}, \text{ where}$$

P = pitch of the larger screw

ρ = pitch of the smaller screw.

Overhauling :

In the case of a lifting machine like a screw jack if the effort is removed and the load moves down by overcoming the thread friction because of its own mass, the machine is said to overhaul. A workman in a factory operates a simple lifting machine by hand but he is not able to continuously provide a constant effort. On occasions he ceases to apply the effort, but the load does not descend. The load may be prevented from 'running back' by some safety catch, but this may not be necessary. The load can be prevented from running back by appropriate design, based on the following considerations.

If application of an effort E, which moves a distance x, causes the load L to move a distance y, then

Energy input = Ex

Energy output = Ly

and

The loss of energy, Ex – Ly, is mainly due to frictional effects within the machine.

Now let us remove the effort. The load will tend to run back and the load must overcome the frictional effects. These are reasonably constant for any given combination of load and effort. Hence for the load to run back the energy it provides in running back mut overcome the frictional effects of the machine. In which case

$$Ly > Ex - Ly$$

and dividing through by Ex,

$$\frac{Ly}{Ex} > 1 - \frac{Ly}{Ex}$$

Now $\frac{Ly}{Ex} = \frac{M.A.}{V.R.}$ = the efficiency

Hence $\eta > 1 - \eta$

and this can occur only if the value of η is in excess of 50%.

We therefore conclude that if a lifting machine has an efficiency of less than 50% the load is self-sustaining, *i.e.* it will not run back. Such machine is said to be self-locking or non-reversible.

The efficiency of a lifting machine is not constant. The efficiency varies for different combination of load and effort. In general, as the load increases, the efficiency increases, but not in direct proportion.

Example :

In a lifting machine, whose velocity ratio is 50, an effort of 10N is required to lift a load of 400N. Is the machine reversible ? If so, what effort should be applied, so that the machine is at the point of reversing ?

Ans. :

V.R. = 50, Effort, P = 10 N, Load, W = 400 N

Reversibility by the machine

We known that $M.A. = \frac{W}{P} = \frac{400}{10} = 40$

Using the reaction,

$$\eta = \frac{M.A.}{V.R.}$$

$$= \frac{40}{50} = 0.8 = 80\%$$

Since the efficiency of the machine is more than 50% therefore the machine is reversible.

Effort to be applied :

As explained earlier, that machine will be at the point of reversing, where its efficiency is 50% or 0.5

Let P = Effort required to lift a load of 400 N when the machine is at the point of reversing.

We know that $M.A. = \frac{W}{P} = \frac{400}{P}$

Again using the relation

$$\eta = \frac{M.A.}{V.R.}$$

$$0.5 = \frac{\frac{400}{P}}{50} = \frac{400}{P \times 50} = \frac{8}{P}$$

$$\therefore \quad P = \frac{8}{0.5} = 16 \text{ N.}$$

Example :

In a differential screw jack, the screw threads have pitch of 10mm and 7mm. If the efficiency of the machine is 28%, find the effort required at the end of an arm 360mm long to lift a load of 5 kN.

Ans. :

Pitch of larger screw, P_1 = 10 mm;
Pitch of smaller screw, P_2 = 7mm
efficiency = 28% = 0.28;
Length of handle l = 360 mm;
weight W = 5kN = 5000N;
Let P = Effort required to lift the load

We know that $V.R. = \frac{2\pi l}{p_1 - p_2} = \frac{2\pi \times 360}{10 - 7}$

$= 753.9$ --- (i)

and $M.A. = \frac{W}{P} = \frac{5000}{P}$ --- (ii)

Using the relation

$$\eta = \frac{M.A.}{V.R.}$$

$$0.28 = \frac{\frac{5000}{P}}{753.9} = \frac{5000}{P \times 753.9} = \frac{6632}{P}$$

$$P = \frac{6.632}{0.28} = 23.69 \text{ N.}$$

QUESTIONS

1. A wheel and compound axle lifts a mass of 68kg with an efficiency of 42%. The effective diameter of the wheel is 635 mm and the effective diameters of the axle are 150mm and 115mm. what effort is absorbed in overcoming the internal resistance of the machine ?

   (Ans. : 43.8 N; 25.4 N).

2. In a Weston differential block there are 14 flats on the smallest pulley and 15 on the larger. What load can be lifted by an effort of 220 N if the efficiency is 36% ?

   (Ans. : 2376 N).

3. Define the terms : mechanical advantage, velocity ratio, efficiency, overhauling.

   Calculate the velocity ratio of the following machines :

   (a) A differential pulley block where the effort is applied at an effective radius of 400mm, the load loop acting on effective diameters of 160 mm and 140mm.

   (b) A weston differential pully block with 50 flats on the larger wheel and 48 flats on the smaller wheel.

   (Ans. (a) 40 (b) 50)

4. A man producing 37 W is raising a mass of 1 tonne by means of a lifting tackle. If the velocity ratio is 80 and the efficiency with this load is 70% what effort is the man exerting and at what rate is the load moving ?

   (Ans. : 175 N; 158.5 mm/min).

5. A simple screw jack has a two-start thread of pitch 2.5mm. Calculate the velocity ratio if the effort is applied tangentially at a radius of 350mm.

   (Ans. : 440)

6. A screw-jack has a two-start thread with a pitch of 6mm. Determine the force required at the end of a jack handle, 0.22m long, to raise a load of 15000 N if the efficiency at this load is 28%.

   (Ans. : 465 N)

7. Derive an expression for the velocity ratio of a Weston differential pulley block.

   A differential block has 10 flats on the large pulley and 9 on the small pulley. Determine the effort required to raise a load of 1250 N if the efficiency at this load is 35%.

   (Ans. : 179 N)

8. In a screw jack one turn of the screw raises the load 13mm. A lever 914mm long is used to raise the load. It is found that 90 N applied at the end of the lever will just raise 1 tonne and that 155 N will just raise a load of 2 tonns. Calculate what load a for :e of 250 N might be expected to raise if applied at the end of a lever 1220 mm long and find the efficiency at this load

   (Ans. : 46580 N; 31.6 %)

O O O

CHAPTER – 5

MECHANICAL TRANSMISSION OF POWER

Power is transmitted mechanically by means of the following devices :

1. Shaft couplings and clutches.
2. Toothed gearing.
3. Belt.
4. Rope.
5. Chain.
6. Hydraulic pressure.
7. A combination of two or more of the above.

Lifting machines, described in earlier chapter, transmit power mechanically from a human agency for a brief period to lift. The arrangements stated above are employed for transmission of power continuously over long periods.

Shafts machines, described in earlier chapter, transmit power mechanically from a human agency for a brief period of lift. The arrangements stated above are employed for transmission of power continuously over long periods.

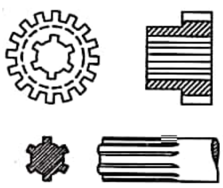


Fig. 5.1 Splined shaft

Shafts for transmitting power are generally made of forged mild steel. Large shafts are made hollow mainly to reduce the weight. The part of a shaft which rests on bearings or supports is called journal. A key is fixed between the shaft and the club and it is genrally placed in a keyway cut partly in the shaft and partly in the boss of the wheel. Where the wheel is required to slide along the shaft while still transmitting power, the shaft is generally splined and is common in gear boxes. (Fig. 5.1)

Shaft couplings and clutches :

A shaft which forms part of a prime mover (like an electric motor) or is directly connected to it, is the *driver* shaft and the shaft or pulley which is driven through gearing, belt, rope, chain, etc. is the driven shaft or pulley.

A flexible coupling is used between the driver and the driven shafts when both are in the same line, run at the same revolutions, but require slight relative movement between the two, both axially and tangentially. A flexible coupling absorbs shock of the vibrations resulting from small errors in alignment of driver and driven shafts. Where slight axial movement of the shaft is unavoidable, flexible couplings are essential *e.g.* in a turbine pump where very small axial movement of the pump shaft occurs even when the axial thrust is hydraulically balanced. Fig. 5.2 shows a flexible coupling commonly used for turbine pumps. It consists of two steel flanges known as the pin half, A, and the bush half, B. The pin half, A, usually has 6 or 8 holes to receive pins or bolts and the bush half B has the same number of holes to receive rubber bushes. The pin half A is keyed on the motor shaft and the bush half B to the pump shaft (or the driven shaft). The pins or bolts are, however, free to slide axially in the bush half B within very small limits. The bushes are loose fit in their housings. (*i.e.* in the holes on the flange) and are made of rubber lined with thin brass sleeves through which the pins or bolts can slide. The bolts carrying the bushes are recessed within the flanges so that there are no dangerous projections.

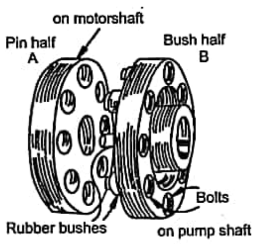


Fig. 5.2 Flexible coupling (flange and bush type)

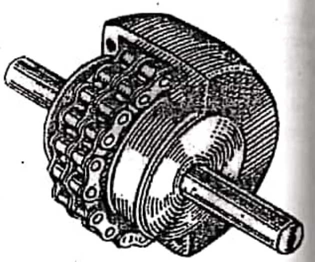


Fig. 5.3 Roller chain coupling

Another flexible coupling is the **roller chain coupling.** (Fig. 5.3). The two halves constituting the coupling are in the form of sprocket wheels. The chain is double width to encircle both the sprockets and it can be disconnected when required. The flexible coupling is enclosed in a split housing which retains the lubricant and excludes dirt.

A flexible coupling commonly used on the propeller shaft of a motor car is the universal joint (Fig. 5.4), or cross bearing. This permits the driver

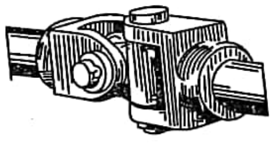


Fig. 5.4 Universal Joint

and driven shafts to be more out of alignment that what is allowed in the two couplings described above. The coupling also takes care of change in the angles between the driver and the driven members when the assembly is in motion.

Hydraulic Coupling :

A *fluid coupling* is also known as *hydraulic coupling* and the drive arrangement with the help of a fluid coupling is known as **fluid drive.** The hydraulic coupling is an important development during the last four decades and the fluid drive is being used on an increasing scale in the mechanical engineering industry. In the automobile industry large cars have mostly adopted "fluid flywheel drive". In the mines such couplings are widely used for belt conveyor drives, and locomotive drives.

A hydraulic coupling transmits engine/motor power through liquid without mechanical linkage between the driver aand the driven machinery. Most fluid drives are designed for use with engines having speeds of 1500 r.p.m. or more. If used with lower speed engines/motors a large size coupling must be used in proportion to power. It is used for motors of as low as 2 kW.

A fluid coupling consists of two main parts. (Fig. 5.5).

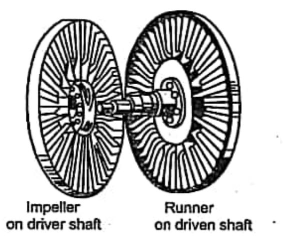


Fig. 5.5. Impeller and runner of a flui coupling

i. an impeller "A" (a disc fitted with radial vanes) which is fitted to and rotated by the driving shaft, and

ii. a runner "B" of similar construction which is secured to the driven shaft of the gear box forming part of the driven machine. The runner is also called the *turbine.*

The impeller and the turbine set are enclosed in an oil tight casing (Fig. 5.6). There is no actual contact between the two parts at any time through the gap separating them is quite small. The casing itself is of bowl type construction and is about 3/4th full of oil. The proper level is indicated by the location of the filler plug. The air space in the coupling provides for expansion of fluid as it heats in service. When the driving shaft carrying the impeller is rotated by the engine/motor, the vanes cause the oil to push by centrifugal force to the periphery. There the oil impinges on the vanes of the runner. When the impeller is driven slowly, the centrifugal force imparted to the oil, and velocity of the latter, are not powerful enough to rotate the runner but as the engine is speeded up, the oil in the impeller acquires sufficient force to cause the runner to rotate. As the shape of the impeller and runner is similar, a free path for the oil stream is provided and the oil circulates. The impeller and runner rotate in the same direction. At full engine speed the degree of "slip" *i.e.* difference between speeds of the impeller and runner, may be neary 2% or even less. The torque transmitted builds up as the square of the impeller speed and as there is no mechanical connection between the impeller and runner, but simply a vortex ring of oil, the acceleration is extremely smooth.

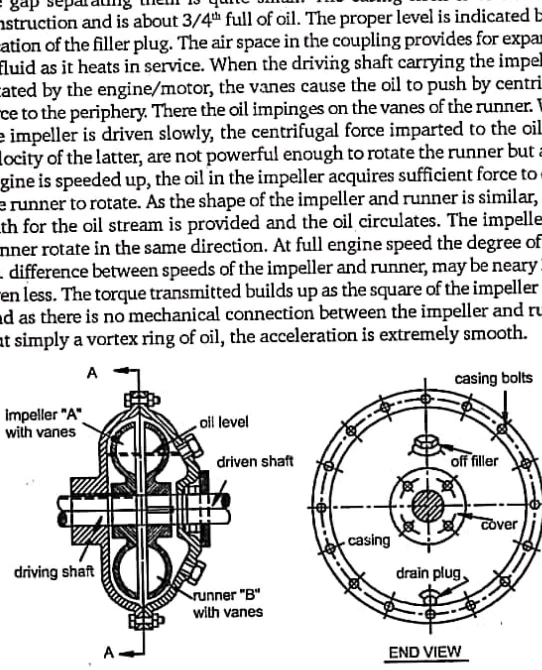


Fig. 5.6 Fluid coupling in cross-section

The hydraulic coupling has the following advantages :

1. It functions as a clutch which gives a perfectly smooth and progressive drive. Rate of acceleration can be varied over a considerable range by increasing or decreasing the amount of oil in the coupling.
2. There is no positive drive as with dog clutch and slip occurs in case of overload. This is an important safety measure. There is no danger of overheating if the driven machine is overloaded.

4. No shock is imparted to the gear box.
5. Engine speed is the only control and to stop the driven machine the speed is reduced to that level at which the impeller cannot drive the runner.

Torque converter : See at the end of this chapter.

Clutches :

A clutch is a device, by which two shafts turning on the same axis-that is, line with each other can be connected and disconnected. Clutch-like action can also be obtained by the use of a movable pulley in a belt system, by engaging and disengaging gears and in othe rways. A clutch is necessary where it is desired to keep a driving motor running while stopping the driven machine. It also enables a motor or engine to run up to speed before the load is applied. In mining, some ofthe familiar examples where the clutches are used are :

1. A direct rope haulage (two usually 100 H.P.)
2. Main and tail rope haulages.
3. Endless rope haulages where one motor serves two ropes of two different drums.
4. Coal cutting machines.

Winding engines normally do not have clutches if winding is from two fixed levels *i.e.* pit top banking level and underground pit bottom level, but windng engines which have to operate for different underground levels are provided with clutches, e.g. in metalliferous mines.

The design of a clutch depends on : whether or not it is to be engaged while running, and on whether or not slipping can be permitted. There are thus two main types, friction clutches and claw clutches. A claw clutch is also called dog clutch or jaw clutch. Winding drums and large direct haulages require a positive drive that cannot slip, and use claw clutches.

These can only be engaged or disengaged at very low speed. Fig. 5.7 illustrates a simple claw or jaw clutch suitable for haulage. It consists of two steel castings, one of which (marked F) is permanently keyed to the (right hand) driving shaft, and the other (makred L) is free to slide along the (left hand) driven shaft which is of octagonal cross section for some distance. The two halves of the clutch have jaws cut in their adjacent faces leaving three segmental projections, claws, or dogs, the width of the jaws being slightly greater than that of the claws. The sliding portion L may be moved along its shaft by the forked lever H which fits in a recess turned in the casting, and thus the two halves or jaws of the clutch may be engaged (as shown) or disengaged as required.

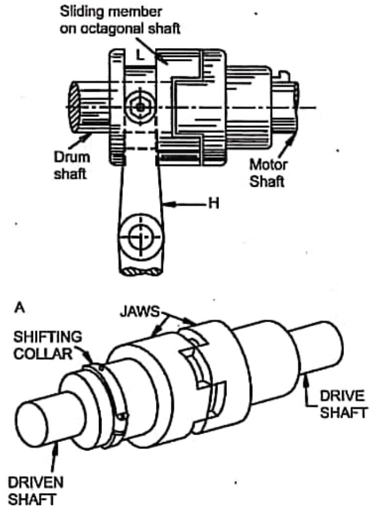


Fig. 5.7 Jaw of dog clutch

The advantages of a dog clutch are that it is simple and strong, and positive in action, but it has the disadvantage that it can only be engaged when the driving and driven shafts are nearly stationary, or are running at the same speed. It is used widely on direct haulages and main-and-tail rope haulages.

The friction clutch :

Fig. 5.8 shows a friction clutch, left being a front view, and right, a transverse section. The clutch consits of two parts :

1. the external (driven) shell E, running freely on the driving shaft S but rigidly attached to the machine to be driven and
2. the compositve internal (driving) mechanism, the boss B of which is keyed securely to the driving shaft.

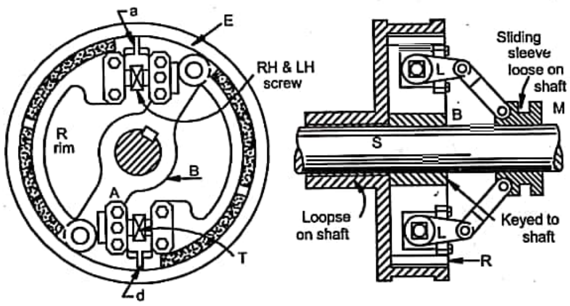


Fig. 5.8 Friction clutch

Extending from the boss B are two arms AA supporting he rim RR. The latter is divided transversely at dd, but the two havles are interconnected partly by the arms AA and partly by the two threaded rods or screws TT, each screw being provided with both right and left-hand screw-threads. Each of the screwed rod T, has a lever L keyed to it, so that the rods may be caused to rotate slightly by the levers LL and the links and sliding sleeve M. Each rod T carries also a screwed bronze nut or bush at both ends and these bushes are housed in recesses made to receive them in the faces of the driving member A. When the rods TT are rotated, therefore, the two halves of the clutch are pushed apart, and are forced into intimate contact with the inner surface of the drum of shell, E. The outer surface of the internal rim, R, is lined with suitable material, *e.g.* "Ferrodo" friction lining to increase its effectiveness and obviate seizure.

The advantage of a friction clutch is that it can be engaged or disengaged without shock when the driving shaft is revolving at full speed, and the load on the driving engine or motor can be increased gradually from zero to full load. It may also be so arranged that the clutch slips if the load becomes excessive, thus avoiding breakage of some important parts of the plant. It is much used on endless rope hauling engines.

The Centrifugal Clutch :

Fig. 5.9 shows another form of friction clutch which is known as centrifugal clutch. It consists of (a) an exterior rim, R, attached to a disc or wheel having a hub which runs loosely on the driving shaft, and (b) an interior spider, A securely keyed on the shaft.

Within the four spaces formed by the arms of the spider are loose segment-shaped shoes, B, (shown shaded) each of which is lined with Ferrodo (shown black) on its outer face. As the driving member, A, increases in speed, the shoes are thrown outwards with increasing centrifugal force and rub on the interior surface of the rim, R, thus graudally accelerating it to full speed.

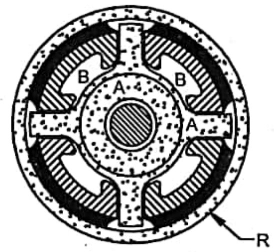


Fig. 5.9 Centrifugal Clutch

The weight of the shoes is arranged to be such that, when the driving motor is runnng at its rated speed, the power transmitted by them is equal to the power of the motor, and all slip then ceases. One of the largest standard clutches, 0.6m outside diametter and running at 800 r.p.m., will transmit 1200 kW. Such a clutch is particularly suitable for drive by electric motor because this can be run at high speed and exerts a uniform torque.

The centrifugal clutch has the advantage that the load is applied gradually and automatically to the driving motor, so reducing the starting current required. This enables a motor with a relatively low starting torque. (*e.g.* squirrel-cage a.c. induction motor) to be used and the motor's starting gear can be of a simple type. If the load exceeds a predetermined maximum, the shoes lip and so prevent overloading the motor. In a modified type, the hows are controlled by springs which prevent any engagement until the motor has attained any predetermined speed up to 75% of full speed.

The rim of the clutch may be utilised as a belt or rope-pulley, or a pulley of smaller diameter may be cast integral with the rim, and adjacent to it. when the clutch is required to carry a spur pinion or chain-sprocket wheel, this is keyed on an extended sleeve integral with the rim. The clutch may also be used as a coupling for joining two shafts together.

Ratchet and Pawl :

The ratchet and pawl, one type of which is illustrated in Fig. 5.10 permits motion of a lever one way, and resists reverse motion unless a catch is released. The ratchet is a plate with hard steel teeth in this curved upper surface. The teeth are sloped on one side but straight on the other side.

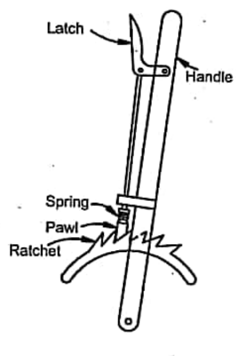


Fig. 5.10 Ratchet and pawl

The pawl is a single tooth held in a slide on the lever and pressed against the ratchet by a spring. It has the same shape as a ratchet tooth but is turned back-ward. In the arrangement of ratchet and pawl shown in Fig. 5.10 if the lever is moved to the right the pawl will move freely across the ratchet teeth, being wedged up and over by the sloped sides. But if moved to the left, its straight edge catches on the straight edge of the nearest tooth and remains so until pulled up by a latch.

Eccentric Thick. An eccentric is in reality a special form of short crank, and its eccentricity is equal to half the travel desired for the valve shown in the

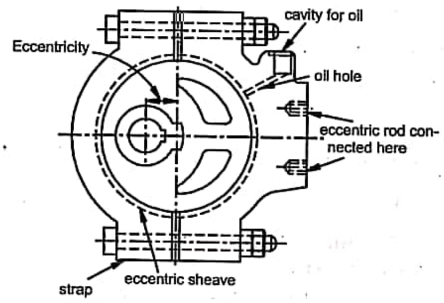


Fig. 5.11

Fig. 5.11 (a) An eccentrics is used for converting the rotary motion of a crank shaft into a small to-and-fro quick motion of a small device like the slide valve of a steam engine or a sizing screen of a coal handling plant/ore dressing plant. It cannot convert the reciprocating motion into a rotary motion.

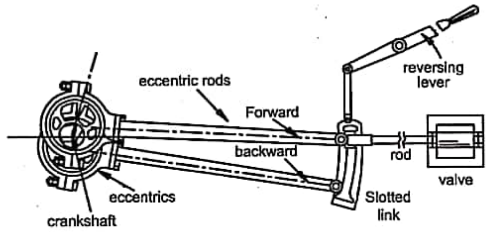


Fig. 5.11 (a) Stephenson link gear for a reversing engine

The eccentric assembly consists of a disc or sheave and a strap made of C.I. or steel. The strap is made into two halves, which are fastened together by bolts on the sheave which is bored out to fit the crankshaft (the hole being offcentre). As the scheave revolves, the straps move to and fro, conveying this motion to the eccentric rod which is connected to the small device such as the slide valve rod in a steam engine, by means of a pin. (Fig. 5.11).

The distance between the geometrical centre of the sheave and the centre of the hole is known as the *eccentricity* or *throw of the eccentric.*

An oil cup is cast integral with the strap for the purpose of oiling. In case of high speed engines, the eccentric strap is lined with white metal.

Fig. 5.11 (a) shows two eccentrics fitted to a crank shaft in Stephenson link gear for a reversing engine. One eccentric serves for forward running and the other, for reverse running, either being in operation according to the position of the slotted link.

Toothed gears :

If two plain wheels A and B (Fig. 5.11) of diameters D_1 & D_2 respectively having sufficient rough surfaces and pressed against each other, are in contact at their circumferences and have parallel axes, rotation of A will cause rotation of B by fricttion. Any point on the circumference of A will travel a linear distance πD_1 in one revolution of the wheel causing any point on the circumference of B to travel the same linear distance. A revolves with N_2 rpm and B with N_2 rpm,

$N_1 \times \pi D_1$ = distance travelled by any point on the circumference of A in 1 min. and it will be equal to the distance travelled by any point on the circumference of B in 1 min.

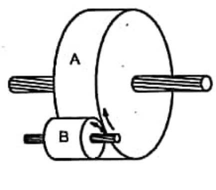


Fig. 5.11 (b)

Revolutions N_2 of B in 1min,

$$= \frac{N_1 \times \pi D_1}{\pi D_2}$$

Therefore, $\frac{N_2}{N_1} = \frac{\pi D_1}{\pi D_2} = \frac{D_1}{D_2}$

It is obvious that if A revolves in clockwise direction B will revolve in anticlockwise direction. The wheel B will be rotated by the wheel A as long as

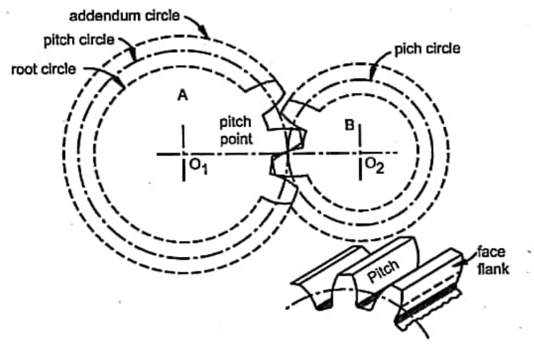


Fig. 5.11 (c) Straight toothed gear wheel

the tangential force exerted by the wheel A does not exceed the maximum frictional resistance between the two wheels. The power transmitted by such an arrangement would obviously be low. The slipping can be avoided by making the wheels toothed and the teeth on each of the wheels are formed partly above and partly below the original wheel circumferences. The original wheel circumferences are called *pitch circles* of the toothed wheels. (or toothed gear). Fig. 5.11 (c) shows the different terms used in relation to toothed gears.

Circular pitch, sometimes called simply pitch, (defined later) = p,

Addendum = height of teeth above pitch circle = 0.3 p

Dedendum = height of teeth below pitch circle = 0.4 p

Clearance = difference between addendum and dedendum.

Face = portion of working surface of teeth outside pitch circle.

Flank = portion of working surface of teeth inside pitch circle.

Pitch diameter = the diameter of the pitch circle.

Tooth thickness measured around the pitch circle is about one half the circular pitch.

Axial width of the teeth is usually from 2.5 to 4 times the circular pitch.

The number of teeth on the smallest pinion of a train of gears should not be less than 12.

Circular Pitch :

The term *circular pitch,* as applied to gear wheels, is the distance between two consecutive teeth, centre to centre, as measured along the pitch circle. Hence

$$\text{Circular pitch, } \rho = \frac{\text{Circumference of pitch circle}}{\text{Number of teeth, T}}$$

and $$\text{Number of teeth} = \frac{\text{Circumference of pitch circle}}{\text{Circular pitch}}$$

In order that two toothed wheels should gear together smoothly, it is essential that they should have the same circular pitch.

When two straight toothed gears mesh, the smaller one is called the pinion wheel or simply pinion and the large one, spur wheel or simply spur, (Fig. 5.12). The number of teeth on the smaller pinion of a set of pinion and spur wheels should not be less han 12. Toothed gearing is used for transmission of rotary motion when a positive drive is rquired between 2 parallel shafts whose centres are comparatively close together. The wheels may be of cast iron, cast steel, or phosphor bronze. If the wheel is of mild steel, the teeth are case hardened. The pinions which run at high speed may be of compressed paper, raw hide or polythene to reduce noise and vibrations. The high speed pinion should always mesh with machine-cut spur wheels, never with moulded gears.

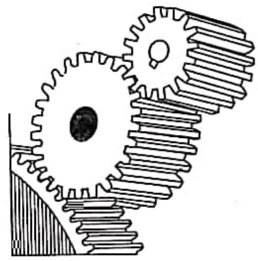


Fig. 5.12 Straight toothed gears. Bevel gears

Top wheel – pinion;
bottom wheel – spur wheel
(partly shown).
Intermediate wheel-idler gear

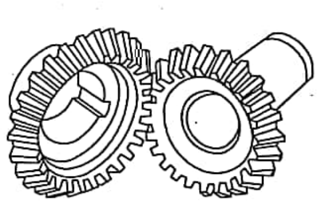


Fig. 5.13 Bevel gears

The wheels may be placed external to each other (as in Fig. 5.11(c)) or one wheel (pinion) internal to another (as in Fig. 5.15.). An internal gear has its teeth cut on the inside of the rim. Such gears are safe, exclude dirt and for a given velocity ratio they re more compact than external gears. External gears rotate in opposite direactions but with internal gears the driver and the follower rotate in the same direction. Internal gears cannot be used for large loads because the gear wheels are overhung. Wheels with internal toothing are extensively used in sun-and-planet gearing. In the rack and pinion arrangement the rack may be considered as a spur gear of infinite radius, (Fig. 5.14) *i.e.* pitch circle is a straight line. Rocks are used to convert rotary motion to a linear one.

Bevel Gears :

These are used for transmission of power between two shafts whose axes, if extended, would meet at a point. Most commonly the shafts are at 90° to each other, but any angle can be accomondated. Just as spur gears are developed from friction discs rubbing on each other, so bevel gears may be considered as derived from two cones touching. The pitch circle of a bevel is the circle forming the base of the cone and the addendum and dedendum are measured at the large end of the teeth. Fig. 5.13 illustrates a pair of bevel gears.

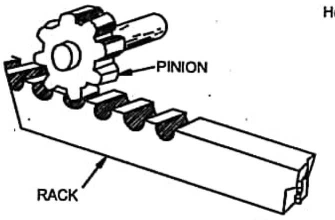


Fig. 5.14 Rack and pinion

Fig. 5.14 (a) Rack and pinion

Work Gears :

This form of gearing is specially convenient where a large speed reduction is required without the use of large or cumbersome spur wheel trains. In effect, a worm is a revolving screw with a thread of involute cross-section, engaging with a wheel whose teeth are curved to match the diameter of the worm. Shafts of the worm and the worm wheel are at right angles and do not meet (Fig. 5.16). The *pitch* of the worm is the axial distance from a point on one thread to the corresponding point on the adjacent thread. *Lead* is the axial distance the worm would move in one revolution if it were not restrained. For a single-start worm, pitch and lead are the same, but for a two-start worm the lead is twice the pitch. If a single-start worm engages a worm wheel of n teeth, then one revolution of the worm rotates the wheel 1/n revolution; or 2/n revolution if a two-start worm is used. The efficiency of a worm drive is very low, usually less than 40% and therefore "overhauling" is rarely possible with a worm drive. The efficiency can be improved to some extent by careful design.

Worm gears are made of case-hardened steel, the worm wheel commonly being bronze or gunmetal. A single worm developes and endthrust and requires some form of thrust bearing to take it up. In some forms of gears two opposed worms, of right-hand and left-hand thread, are used so that the thrusts balance out.

Because of their high velocity-ratio and compactness, worm reduction gears are often used in small haulages a single unit sufficing where two sets of spur gears would be necessary.

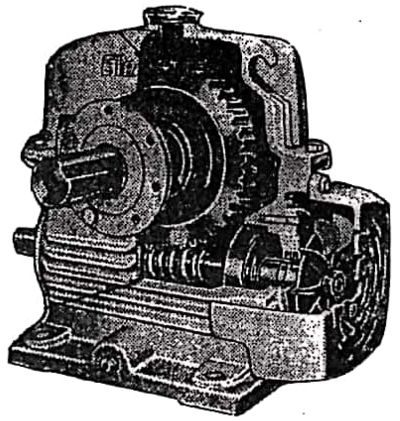


Fig. 5.15 A worm and worm wheel in a reduction gear assembly.

Helical Gearing :

Helical gears are preferred to straight gears for heavy drives. A helical gear differs from ordinary straight toothed gearing in that the teeth are not parallel to the axis of the wheel but form an angle thereto. Single helical teeth are set diagonally across the face of the wheel in single formation, whilst double helical teeth are V-shaped or in the form of a herring-bone. The advantage of the latter shape is that it is self-balancing, no end-thrust being set up on the shaft. Single helical gears are seldom used as they develope an end thrust and the commonest type used is the double helical gear with V-shaped teeth, with the point of V in the front. (Fig. 5.16).

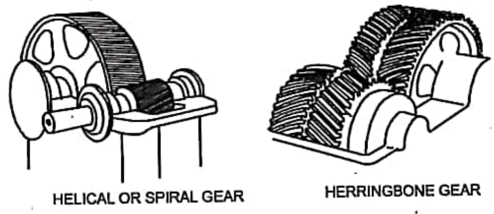


Fig. 5.16 Helical gear.

With the helical gear, the lines of contact move in a lengthwise direction along the tooth. The result is that each tooth picks up its load smoothly and gradually and is relived of its load in the same manner; wear and vibrations are reduced; and the efficiency of power transmission is greatly increased. Larger velocity ratios can be obtained with double helical gears, since the teeth are in more continuous engagement, and pinion with as few as 4 teeth have been used. Helical gears are stronger than straight gears, are smooth running and quiet. Naturally they are much more expensive to manufacture than straight gears.

Gear Trains :

$$\text{We have seen that velocity ratio} = \frac{\text{distance moved by effort}}{\text{distance moved by load}}$$

$$\text{In the case of gear wheels, V.R.} = \frac{\text{speed of the follower wheel}}{\text{speed of the driver wheel}}$$

If a larger velocity ratio has to be achieved when 2 parallel shafts are some distance apart, a combination of only one pinion and one spur wheel will make the arrangement unwieldy. Normally a velocity ratio of 10:1 is the maximum available from a set of a pinion and spur wheel with *straight teeth.* Introducing intermediate spindles fitted with spur gears and thus bridging the distance between the shafts is a suitable arrangement to obtain high velocity ratio. Such a sysem of toothed wheels is called a gear train. (See Fig. 5.12).

If the wheels in the train shown in Fig. 5.12 are numbered 1, 2, 3 having respectively T_1, T_2, T_3 number of teeh of the same pitch since 1 gears with 2,

$$\frac{N_2}{N_1} = \frac{T_1}{T_2} \text{ where N indicates r.p.m.}$$

Similarly since 2 and 3 gear together.

$$\frac{N_3}{N_2} = \frac{T_2}{T_3} \qquad \therefore \frac{N_3}{N_2} \times \frac{T_2}{N_1} \times \frac{T_1}{T_2} \text{ or } \frac{N_3}{N_1} = \frac{T_1}{T_3}$$

$$\text{So, } \frac{\text{speed of last follower}}{\text{speed of first driver}} = \frac{\text{number of teeth on first driver}}{\text{number of teeth on last follower}}$$

Thus, in a simple train of wheels, the velocity ratio transmitted is independent of intermediate shaft or shafts. Because the intermediate wheels do not affect the velocity ratio, they are known as idle wheels, or idlers. The idlers serve the purpose of bridging the space between the first driver and the last follower and they provide the arrangement to increase the velocity ratio.

Compound train of wheels :

A gear train in which each shaft carries one wheel only is caled a simple train, and one in which two wheels are mounted on a common shaft, is called a compound train. In a compound train of wheels there are three or more shafts and each intermediate shaft has, instead of one gear wheel, two wheels rigidly keyed so that both gear wheels have the same speed (r.p.m.). One wheel meshes with the driver and the other with the follower to which the motion is to be transmitted. (Fig. 5.18). If the wheels are numberd 1, 2, 3 .... with number of teeth T_1, T_2, T_3 .... respectively and speeds N_1, N_2, N_3 .... respectively the speeds are calculated as follows :

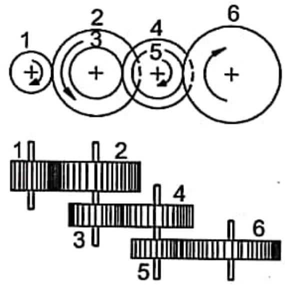


Fig. 5.17 Compound train of wheels

Wheel 1 gears with wheel 2. Therefore we have

$$\frac{N_2}{N_1} = \frac{T_1}{T_2}; \; N_1 = N_2 \text{ and since wheels 3 and 4 gear together,}$$

we have,

$$\frac{N_4}{N_3} = \frac{T_3}{T_4}; \; N_4 = N_5 \text{ and for wheels 5 and 6,}$$

we have $\frac{N_6}{N_5} = \frac{T_5}{T_6}$

$$\frac{N_6}{N_5} \times \frac{N_4}{N_3} \times \frac{N_2}{N_1} = \frac{T_1}{T_2} \times \frac{T_3}{T_4} \times \frac{T_5}{T_6} \text{ or } \frac{T_1 \cdot T_3 \cdot T_5}{T_2 \cdot T_4 \cdot T_6}$$

Therefore,

$$\frac{\text{speed of last follower}}{\text{speed of first driver}} = \frac{\text{product of the number of teeth on drivers}}{\text{product of the number of teeth on followers}}$$

With a compound train of wheels, a large velocity ratio can be transmitted by the use of small sized wheels so that the motion of the last follower is considerably intensified or reduced as desired. In this respect a compound train of wheels has a decided superiority over a simple train of wheels. For instance, let the number of teeth on the wheels in Fig. 5.17 be 15, 30, 20, 25, 20 and 40 respectively.

$$\frac{N_6}{N_1} = \frac{15}{30} \times \frac{20}{15} \times \frac{20}{40} = \frac{1}{5}$$

The last shaft runs at 1/5th of the speed of the first shaft. To obtain this speed reduction by a simple train of wheels since intermediate wheels will be idle.

$$\frac{N_6}{N_1} = \frac{T_1}{T_6} \text{ and } T_6 = 5T_1$$

If we have the smallest number of teeth as 15, since $T_1 = 15$, T_6 must be 75, almost twice as large as the largest wheel in the compound train for transmitting the same speed-reduction.

Example :

A train of wheels transmits 4 kW. The driving pinion has 20 teeth of 8mm module. Calculate the permissible speed in r.p.m. for the pinion if the pressure between the teeth exerted at the pitch points is 500 Newtons.

Ans. :

$P \times v = 4 \times 1000$

Since P is limited to 500 N, we get

$500 \times v = 4000$ or $v = 8$ m/sec.

Since $m = 8 = \frac{d}{T} = \frac{d}{20}$ $\therefore d = 8 \times 20 = 160$ mm

$$\omega = \frac{v}{r} = \frac{8}{0.08} = 100 \text{ rad/sec.}$$

$$N \text{ (r.p.m.)} = \frac{60\omega}{2\pi} = \frac{60 \times 100}{2\pi} = \frac{3000}{\pi} = 955.4 \text{ r.p.m.}$$

Epicyclic Gearing :

An epicyclic gearing s one in which there is relative motion between the axes of the shafts. It is an internal form of gearing set used for compactness and a high reduction ratio between the speeds of the driver and driven shafts. A typical arrangement is shown in Fig. 5.18. Epicyclic gear is also known as sun and planetary gear. In the arrangement of sun and planetrary gear, 1 is the driver shaft or axle shaft keyed to 2, called the sun pinion or sun gear. It is the central "sun" wheel which meshes with two or three gear wheels, called planet gears, each revolving of its own axis and carried on a spider (not shown in the Fig.). The spider itself can rotate around the "sun". The internal gear 4, is the ring gear which meshes with the planet gears. The planets are caused to rotate by the turning of the sun gear. This rotation forces them to *walk* round the toothed track from by the ring gear, pulling the carrier or spider around with them in the same direction as the axle 1 rotates. Thus when the sun pinion is rotating and the planet carrier (or spider) is free to revolve, the planets will run round the sun, turning on their own axes but not communicating any motion to ring gear. The spider which carries the shafts of the planet gear is attached to a driving flange, as in the case of rear wheel of Terex dumper.

The flange is bolted to the wheel hub. For reduction in speed and increase in torque, power is supplied to the axle shaft, , and work is performed by members attached to or driven by the flange is bolted to the wheel hub. For reduction in speed and increase in torque, power is supplied to the axle shaft, 1, and work is performed by members attached to or driven by the flange. If the ring gear has 30 teeth and the sun pinion has 10, torque multiplication is $\frac{30}{10}+1=4$

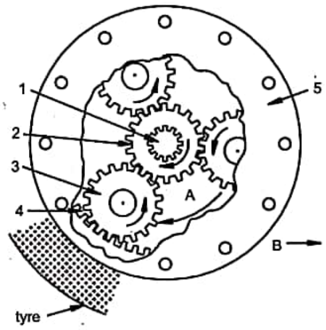


Fig. 5.18 Epicylic gear.

Cycle Speed Reducer :

The Cyclo Speed manufactured by Cyclo Enterprises, Satara, is a device of high reduction ratio upto 85:1 which is achieved in single stage only and high ratios are available by combining two or more stages. In Cyclo Speed Reductor, the speed reduction is based on cycloidal discs, which are moved by an eccentrite connected to the input shaft. The continuous cycloidal cirumference of the discs engage with a set of rollers fixed and evenly spaced in the housing. The number of rollers is one more as the number of cycloidal curves on the discs. With each revolution of the eccentric the disc is advanced by one roller in the opposite direction, giving the disc one full revolution after the eccentric has turned as many as there are curves on the disc. The rotation of the cycloidal disc is transmitted concentrically to the output shaft, through the drive pin sleeves, which thus turns at the reduced speed. (Fig. 5.19).

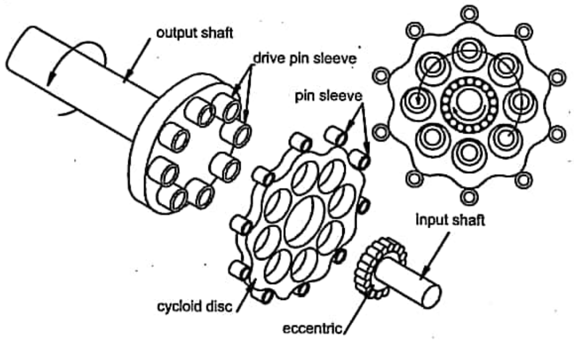


Fig. 5.19

The most critical components of the Cyclo speed reducer ar the cycloidal discs and the eccentric spaced 180 degrees apart to avoid binding stresses of the input and output shafts and to stabilise the unit dynamically. The rotation of the output shaft is absolutely uniform and compulsorily positive. All power transmitting components carry partial load at the same time. Cyclo speed reducer operates with rolling contact only, similar to roller bearings; hence it eliminates vibrations and noise produced by conventional toothed engagements.

Cyclo speed reducer can be applied virtually in any industry where low speed is required. It may be mounted in any position, contains no material subject to ageing or deterioration, and it can be furnished for electric motor or any other power sources for fraction of an horse power upto 10 horse power at present.

Cyclo speed reducer can be used in any position and turns in either direction, never self locking and can be used in any atmosphere.

The inertia effect of the cyclo is almost negligible compared to squirrel cage induction motor. This gives quick response. It is very useful for automatic control systems in machine tool, spining machinery, hoisting system, etc.

Belt and rope drives :

Belts or ropes used for drives are endless *i.e.* the two ends are joined together. Flat, flexible canvas or cotton belts are commonly employed and they are joined by patent metal fasteners or by lacing. Belts and ropes are used for transmission of power between parallel shafts. They absorb shocks, are cheaper and easier to install and service than chains, but compared to the latter, wil not last as long and will not cary as heavy loads. An overloaded belt or rope usually slips, thereby preventing damage tor driver/driven machine and providing safety in operation. Distance between pulleys of a rope or belt drive should be adjustable owing to the stretch of the rope/belt during use. A driving pulley transmits power to a belt by friction between them. So also the belt transmits power to the driven (follower) pulley by virtue of friction. The slight curvature or camber (or double coning as it is sometimes called) of the pulley face causes a flat belt to rise easily and fairly in the centre line of the pulley face without inclining to either side. Camber of the driving and driven pulleys should be the same, and ratio of pulley diameters for 2 pulleys of one belt/rope drive should not normally exceed 6:1. The distance between pulley centres of flat belts should preferably be not less than 4-6m for light belt and 6-8m for heavy belts, but this may be reduced to 2/3rd in case of V-rope drive.

Rope drives and V-belt drives require grooved pulleys. (A V-belt drive is often referred to as rope drive). V-belts are of trapezoidal cross-section (wedge-shaped) and run in V-shaped grooves of generally 45° taper. The V-belts are generally 12 mm to 50 mm at the top and are manufactured in specified lengths so that their ends cannot be joined by the user. The depth of the belt is 2/3rd to 3/4th of the top width.

The V-belt drive is-semipositive in character, noisless and highly efficient, even with short centre drives where space is limited. Under heavy overload the V-belt will slip. A great advantage of the V-belt drive is that usually 2 or more belts are employed and if one belt breaks the remainder are normally capable of carrying the load till the broken belt is replaced.

The belts are usualy made of rubberised cotton or rayon cords impregnated in rubber with an overall layer of rubber on its surface.

Power transmitted by a belt :

As already stated a driving pulley transmits power to a belt by friction between them. So also the belt transmits power to the driven (follower) pulley by virtue of friction.

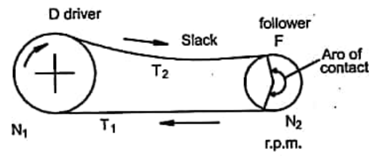


Fig. 5.20.

Fig. 5.20 shows an ordinary belt drive in which the pulley D (called the driver) is driving another pulley F (called the follower) by means of a belt. When the pulley D begins to revolve, the belt-tension on the driving side T_1 tends to stretch the belt on the side until the pulley F begins to revolve; the tension on the following or slack side T_2 is gradually diminished until the difference of the tensions $(T_1 - T_2)$ produces a uniform velocity of the belt.

This difference $(T_1 - T_2)$ is called the nett tension and is the effective tension driving the pulley F. In practice, is it usually assumed that $T_1 = 2T_2$ and therefore the net tension $(T_1 - T_2) = \frac{T_1}{2}$ = one half of the tension on the driving side. The value of T_1 can be calculated from the tables supplied by the manufacturers and taking into consideration the width of the belt and the safety factor. It will also be clear that the slack side of the belt should always run from the top side of the driving pully so that the sag of the belt will increase the arc of contact of the belt on both-pulleys. Any drive between the horizontal and 45° is considered good practice.

If the driver rotates at N_1 rpm and the follower at N_2 rpm, and there is no slip, $\frac{N_2}{N_1} = \frac{D_1}{D_2}$ neglecting belt thickness.

$\frac{N_2}{N_1}$ is called the velocity ratio of the rotating pulleys. In a belt drive the speed of a pulley is inversely proportional to its diameter.

Let T_1 = tension of tight side of a belt.

T_2 = tension of stack side of a belt.

D = distance of the driven (*i.e.* follower) pulley.

The nett tension or effective tension driving the pulley = $T_1 - T_2$

The torque producing rotation of the follower.

$$= (T_1 - T_2) \times \frac{D}{2}$$

Let us assume that the units of T_1 and T_2 are Newtons, the unit of D is meters and the pulley rotates at ω rad/s.

Work done in 1 sec. = torque × angle turned in 1 sec.

i.e. power $P = (T_1 - T_2)\frac{D}{2} \times \omega$

$$= (T_1 - T_2) \times \left(\omega \frac{D}{2}\right)$$

now $\omega \frac{D}{2} = \omega r$ which is linear speed of the belt in m/s. Let us call this v.

Hence P $(T_1 - T_2) \times v$ watts.

It should be noted that in this last formula the value of T_1 and T_2 must be in units of Newtons, while the unit of V must be m/s.

Example :

A flat belt drives a pulley 686 mm diameter at 300 r.p.m. and the tension in the tight side is 2.5 times that in the slack side. If the maximum working tension in the belt is not to exceed 14.2 N per mm width, what belt width is necessary for transmission of 11 kW ?

Ans. :

Effective torque T on pulley

$= (T_1 - T_2)r1 = (2.5\,T_2 - T_2) \times 0.343$

$= 0.515\,T_2$ Nm

Work done = $T\theta$ and $\theta = 300 \times 2\pi/60 = 10\pi$ rad/s

Work done per second = $0.515\,T_2 \times 10\pi = 5.15\pi\,T_2$ J/s

Power transmitted = 11 kW = 11000 = W = 11000 J/s

$\therefore$ $5.15\pi\,T_2 = 11000$ and $T_2 = 680$ N

$T_1 = 2.5\,T_2 = 1700$ N

Width of belt $= \frac{1700}{14.2} = 119.7$mm, or say 120mm

The ratio of the belt tension :

If the velocity of a belt is constant the power that is transmitted will depend upon $T_1 - T_2$ and if the ratio $\frac{T_1}{T_2}$ is made larger, greater power will be transmitted. However the ratio $\frac{T_1}{T_2}$ cannot exceed a certain value for a particular drive. Otherwise the belt would slip on the pulley. The value of $\frac{T_1}{T_2}$ is connected with the angle of lap round the pulley, θ, and the co-efficient of friction μ.

An increase in either value will lead to an increase in the value of $\frac{T_1}{T_2}$ as can be seen from the formula $\frac{T_1}{T_2} = e^{\mu\theta}$

Where T_1 and T_2 are belt tensions,

e is the constant 2.718 which is the base of Napierian logarithms,

μ = co-efficient of friction

θ = angle of lap round a pulley in radian.

When expressing the formula in terms of logarithms to the base 10, the formula is.

$$2.3 \log_{10} \frac{T_1}{T_2} = \mu\theta$$

The above formula holds good for normal or slow belt speeds. At high belt speeds tension due to centrifugal force comes into play. This tension, T_c, should be added to T_1 as well as T_2 so that tension on the slack side is $_2 + T_c$ and that on the tight side is $T_1 + T_c$. The pull T_c, known as the centrifugal tension in the belt, = wv^2 where w is the mass of the belt per metre length, v is in metres per second and T_c is in Newtons.

It can be shown that when the power transmitted is maximum, $\frac{1}{3}$rd of the maximum belt tension T is absorbed as centrifugal tension, or v $\sqrt{\frac{T}{3w}}$

The angle θ is the angle of lap on the smaller pulley. This angle is smaller than the angle of lap on the larger pulley and therefore slip occurs first on the smaller pulley. Once the slip occurs, the purpose of transmission is defeated. If the angle of lap is increased by use of a deflection pulley, the smaller angle should be considered.

The motion of the belt (from the driver) and the follower (from the belt) is governed by a firm grip due to friction between the belt and the pulleys. Therefore the belt is tightened up in order to keep a proper grip of the belt over the pulleys. Even when the pulleys are stationary, the belt is subjected to some tension, called initial tension. It is $T_0 = \frac{T_1 + T_2}{2}$ where T_1 and T_2 are the tight side and slack side tensions respectively.

Example :

A cord hung over a flat pulley supports a weight W_1 at one end and W_2 at the other end. If the coefficient of friction is 0.3, find the greatest ratio of W_1 to W_2 possible without the cord slipping, Angle of contact is 180°.

Ans. :

$$\frac{T_1}{T_2} = 2.718^{0.3\times3.142}$$

$$\text{Log}\frac{T_1}{T_2} = 0.9426 \log 2.718 = 0.9426\times0.4343$$

$$= 0.4.95 = \log 2.567$$

Hence w_1 can equal 2.567 w_2 before slipping occurs. This example is the basis of the Koepe system of winding.

Example :

In a conveyor belt drive the tension on the tight side is double that on the slack side. If the value of μ is 0.3, find the angle of lap required if the belt is not to slip on its driving drum.

Ans. : $\frac{T_1}{T_2} = 2.718^{\mu\theta}$. Now $\frac{T_1}{T_2} = 2.$

Taking logarithms

$$\text{Log } 2 = 0.3\theta \times \log 2.718$$

$$\therefore \quad \theta = \frac{0.3010}{0.3\times0.4343} = 2.31 \text{ radians}$$

$$= 132°$$

Example :

A conveyor belt running at 500 m/min transmits 5 kw. The angle of lap is 160° and coefficient of friction between the belt and the pulley is 0.3. If the permissible pull in the belt is 16 Newtons per mm width, calculate the tension on the driving side and the width of the belt required.

Ans. :

$\frac{T_1}{T_2} = e^{\mu\theta}$, θ is in radians and e is the base of Napierian logarithms or 2.718 (approx). Expressing the above formula in terms of logarithm to the base 10, we have

$$2.3 \text{ Log}_{10} \frac{T_1}{T_2} = \mu\theta$$

$$160° = \frac{3.14\times160}{180} = 2.79 \text{ radians}$$

$$\mu\theta = 0.3 \times 2.79 = 0.837$$

$$\text{Log}\frac{T_1}{T_2} = \frac{0.837}{2.3} = 0.364$$

$$\therefore \quad \frac{T_1}{T_2} = 2.314.$$

Work done/sec = $(T_1 - T_2) \times v = 5 \times 1000$ Joules, wher v is m/s and T_1, T_2 are in Newtons

$$\therefore \quad (T_1 - T_2)\times\frac{500}{60} = 5000$$

$$T_1 - T_2 = \frac{5000\times60}{500} = 600$$

since $T_2 = \frac{T_1}{2.314}$,

we get $T_1\left(1 - \frac{1}{2.314}\right) = 600$

Or $T_1\left(\frac{1.314}{2.314}\right) = 600$

and $T_1 = \frac{600 \times 2.314}{1.314} = 1056.6$ Newtons

If b is the width of the belt,

b × 16 = 1056.6, since permissible pull is 16 N/mm and

$$b = \frac{1056.6}{16} = 66 \text{ mm}$$

Vee belt drives :

It instead of a flat belt we have a Vee belt mating with a grooved pulley, the normal raction between the belt and the pulley is increased by the wedging action of the Vee belt in the grooved pulley. The two belt tensions provide a radial force acting toward to centre of the pulley. Fig. 5.21 shows a section through a small length of Vee belt drive, the radial force on this small length being R.

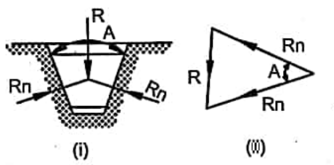


Fig. 5.21

The vector diagram is shown in Fig. 5.21 (ii) ; from the diagram

$$\frac{R_n}{\frac{R}{2}} = \text{Cosec}\frac{A}{2} \quad \text{or} \quad R_n = \frac{R}{2}\text{cosec}\frac{A}{2}$$

There are 2 friction surfaces, hence the total reaction normal to the friction surfaces is R $\text{cosec}\frac{A}{2}$. Hence the effect of replacing a flat belt by a Vee belt is to increase the total normal reaction. Since the frictional force is μ times the normal reaction, the frictional force is also increased.

The formula for flat belt drives can also be used for V belt drives provided that the value used in the formula for μ is adjusted.

With vee belt $\frac{T_1}{T_2} = e^{\mu_1 \theta}$ where $\mu_1 = \mu\,\text{cosec}\frac{A}{2}$

or $2.3\,\text{Log}_{10}\frac{T_1}{T_2} = \mu\,\text{cosec}\frac{A}{2}\,\theta$

θ = angle of lap of belt on *small pulley*, and A being the included angle of the grooved pulley.

(It should be noted that a flat belt can be considered as a special case of Vee belt, the angle A being 180°).

For V-belt drives the angle of contact or angle of lap is that on the smaller pulley but for conveyor belts, and ropes of frictional winders, the angle of lap is that on the driving pulley, *i.e.* the pully connected to the driving motor, and the angle of lap is increased by the use of snub pulleys or deflection pulleys.

Example :

A short centre multiple rope drive connects two grooved pulleys. The larger pulley has a pitch diameter of 350mm and rotates at 720 rpm. The angle of lap round the larger pulley is 240°. The Vee ropes have an included angle of 40°. There are five ropes, the maximum tension per rope is 280 N and the co-efficient of friction is 0.3. Determine the maximum power that can be transmitted by the drive.

Ans. :

The ratio of belt tensions must be calculated with reference to smaller pulley. If the belt laps the larger pulley by 240°, the lap round the smaller pulley is

$$360° - 240° = 120°$$

$$\mu_1 = \mu \operatorname{cosec}\frac{A}{2} = 0.3 \operatorname{cosec} 20^\circ$$

$$= 0.3 \times 2.9238 = 0.8771$$

$$120^\circ = \frac{2\pi}{3} \text{ rad.} = 2.094 \text{ rad.}$$

$$\mu_1\theta = 0.8771 \times 2.094 = 1.8374$$

$$e^{1.8374} = 6.280 \therefore \frac{T_1}{T_2} = 6.280$$

$$T_2 = \frac{T_1}{6.280} = \frac{280\text{ N}}{6.280} = 44.59\text{ N}$$

$$v = \pi DN = \frac{22}{7} \times 0.35 \times \frac{720}{60}$$

$= 13.2$ m/s

Power transmitted by one rope $= (T_1 - T_2) \times v$

$= (280 - 44.59) \times 13.2$

$= 235.41 \times 13.2$W

$= 3107.41$ W

Total power $= 5 \times 3107.41$ W $= 15537$ W

Example :

The usual arrangement of different methods of mechanical power transmission is illustrated in Fig. 5.23 which shows a main and tail rope haulage operating at rope speeds of nearly 8km/h. The electric motor has an r.p.m. of nearly 1450 or 950. The first reduction from such high r.p.m. of nearly 1450 or 950. The first reduction from such high r.p.m. is by V belts R on pulleys P and F, in the ratio of usually 4:1 or 5:1. The shaft of F has still high r.p.m. and the second reduction is by double helical gear wheels G and H in the ratio of usually 4:1 or 5:1. The third reduction is by straight toothed gears K and L in the ratio varying between 3:1 and 5:1. In a large number of cases, however, only two reductions suffice.

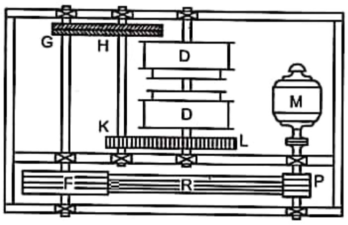


Fig. 5.22 Drive arrangement of a main and tail rope haulage

Chain Drive :

A drive involving chain for transmission of power has the following advantages.

i. Chains are, as it were, an intermediate form of drive that will give efficient service with a centre-distance too short for belts or ropes and too long for toothed gearing.

ii. The chains ensure a positive transmission, whilst at the same time affording a beneficial measure of elasticity.

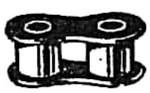
iii. The efficiency of chain drive is very high, there being no slip. There is very little tension in the slack side of a chain drive and therefore the power transmitted equals Tv watts where T is the tension (Newtons) in the tight side and v is the speed in m/sec.

iv. The initial tension required with belts or ropes is unnecessary with a chain, and the bearing friction due to this is therefore eliminated.

v. Chains are unaffected by atmospheric conditions and are compact, being much more powerful than either belts or ropes.

The chain wheels are called sprockets. The 3 chief types of power-transmission chains are : (a) block chain (b) roller chain with bush or without bush and (c) silent chain or inverted tooth type chain. The block chain is made of straight or curved blocks connected by links and pins. The silent chain is so called because it is legg noisy than the block, bush or bush-roller types of chian drives. The block chain and roller chain are used for low and medium speeds and the inverted tooth type, for the higher speeds, or where special flexibility and silence are required. Speeds usually range between 150 m and 450m per minute, and various widths of chain are obtainable. The links of the chains are made of hightensile steel, the bearing surface being case-hardened to reduce wear of the joints. Where necessary, a number of chains are mounted side by side on suitable chain-wheels, the number depending on the power to be transmitted.

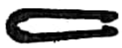
"Close Pitch" roller chain



Inner Link

Cranked link

Outer link



Spring clip

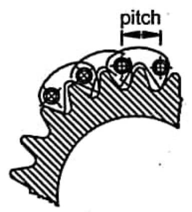


Fig. 5.23 Silent chain

Fig. 5.24. Roller chains

The pitch of a chain is the distance between corresponding points on adjacent links.

The power transmission by a chain is TCPN watts if

P = pitch (meters)

t = effective tension in chain (Newtons).

N = speed of the sprocket (rev/s)

C = no. of teeth of the sprocket

Chain drives run best if totally enclosed and in an oil bath, when they have a very high efficiency and are almost noiseless.

Torque converter :

A torque converter, which is an advance over the fluid coupling, is a device used to increase the torque while reducing the speed. In this respect it performs a function similar to that of a gear box, but whereas a gear box provides only a small number of fixed ratios, the torque converter provides a continuous variaton of ration from the lowest to the highest upto a certain limit. (Fig. 5.25)

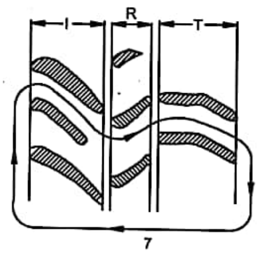


Fig. 5.25 Elements of torque converter.

I – Impeller stage. *R – Reaction stage.*

T – Turbine stage. *F – Fluid circulation.*

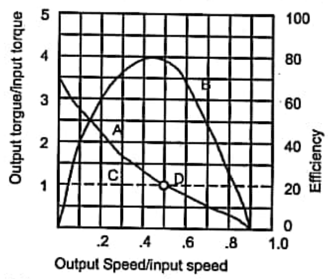


Fig. 5.27 Typical characteristic curve of a single-stage torque converter.

A–Torque *B–Efficiency*

C–Constant input torque *D–Clutch point.*

The construction of a torque converter is essentially like that of a fluid coupling except that an additional reaction member with a set of guide vanes is introduced between the impeller and turbine. The elements of a torque-inverter are, therefore,

1. An impeller keyed to the shaft of the prime mover. It has radial vanes or blades on a ring.
2. The fixed element or stator or reaction member with guide vanes on a ring. The purpose of the fixed guide vanes is to change the direction of the fluid that leaves the impeller at a high speed.
3. The driven element or rotor (also called turbine) having guide vanes on a ring. It is connected to the driven machinery *e.g.* to the propeller shaft in the case of dumper.

All these elements are placed in the housing of the torque converter which is filled with fluid.

The action of the torque converter is somewhat similar to that of a fluid coupling, described earlier, and is as follows :

As the impeller is driven by the engine, the fluid flung out of the periphery of the impeller by centrigugal force, enters the guide vanes of the stator and is defected by them. The fluid then impinges on the vanes of the turbine, causing it to rotate in the same direction as the impeller. The change of direction of the fluid in the stator results in torque multiplication. Under normal running conditions the action of the impeller causes the turbine blades to turn almost freely and fluid passes through the converter easily and quickly, striking each blade on a very slight angle. But when a load is encounterd, the turbine slows down since it is directly connected to the driven shaft. The fluid then strikes the turbine blades at a sharper of larger torque to the driven element. Thus the torque converter selects the proper output torque for any load. In operation selects the proper output torque for any load. In operation the engine runs essentially at constant speed driving the impeller also at constant speed whereas the turbine runs fast or slow justlike an electric D.C. series motor adjusting its speed automatically to the load. The turbine shaft speed varies inversely as the torque so that with increase in torque, the turbine shaft speed decreases until a limit is reached beyond which the shaft simply stalls without causing any overload of the engine.

In a single stage torque converter having only the three elements stated above, the torque multiplication is of the order of 3 to 3.5 and seldom exceeds 4.0 except in special designs. In a 3-stage torque converter, the torque multiplication is nearly six. The elements in a 3-stage torque converter are : one impeller, two stators (reaction members) and three turbines.

Fig. 5.26 indicates typical characteristic curves of a single stage torque converter. The important point on the characteristic curve for output torque is the *clutch point* or speed ratio at which the output torque equals input torque. Clutch point is referred to as specific torque by some manufactures.

Operating beyond the clutch point, *i.e.* high speed ratio, both the torque ratio and the efficiency fall rapidly. Torque converters have a reasonably good efficiency over only a narrow range of rotor speeds. The loss of efficiency arises mainly due to slippage inside the fluid drive and results in heat which has to be dissipated by an outside cooling system, commonly a radiator. A good operating range for a converter fluid is between 83° and 104°C.

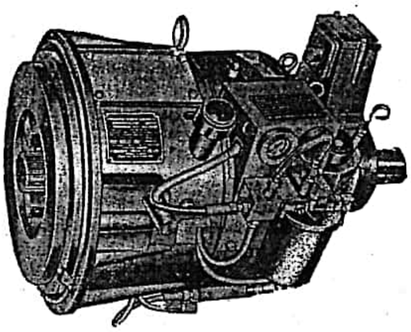


Fig. 5.27 3-stage hydraulic torque converter manufactured by Kirloskar Pneumatic Co. Ltd.

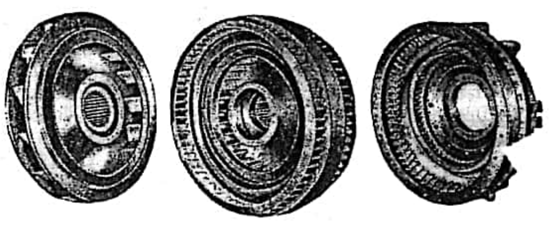


Fig. 5.28 Can exploded view.

The maximum torque multiplication occurs when the turbine is stationary (speed ratio-zero) and the value of torque multiplication at that stage is referred to as *stall torque ratio*.

If a torque converter is used, it is not necessary to employ a fluid coupling as a torque converter serves the purpose of a coupling as well as of a gear box. Some torque converters have twin turbines with rotating housing. At the time of start, only one of the turbines which is designed for high torque and low speed becomes effective, whereas during the run, the other turbine designed for high speed and low torque, comes into play automatically without any attention from the operator.

Torque converters are not used for small engines/motors and prime moves of 30kW or more power are suitable for their utilisation.

QUESTIONS

1. What is a clutch ? Name the different types of clutches used on the common machines in the mines and describe a jaw clutch with a sketch.
2. Describe a hydraulic coupling and explain the principle of its working.
3. A worm-reduction gearbox has a reduction ratio of 40, implying that if the input shaft turns 40 revolutions the output shaft turns one revolution. It is found that an input torque of 50Nm results in an output torque of 560Nm. Calculate the efficiency of the gearbox at this loading. (Ans. 28%).
4. Write notes on : Epicyclic gearing, helical gearing, rack and pinion arrangement, universal joint. Circular pitch.

O O O

CHAPTER – 6

STRENGTH AND PROPERTIES OF MATERIALS

The physical and mechanical properties of metals and other materials which interest most of the engineers while selecting a material are given below, but an engineer is very often concerned with only two or three of the properties which assume importance when the metal or material is put to use.

The properties are :

1. Specific gravity.
2. Tensile strength.
3. Compressive strength.
4. Shear or torsional strength.
5. Hardness.
6. Brittleness.
7. Fatigue resistance.
8. Toughness.
9. Damping capacity.
10. Wear resistance.
11. Malleability.
12. Ductility.
13. Weldability.
14. Machineability.
15. Friction (for bearings)
16. Corrosion resistance.
17. Impact resistance.
18. Moulding ability.
19. Rigidity.
20. Magnetic property.
21. High temperature resistance.
22. Low temperature
23. Effect of increase or decrease of temperature on the above properties.

Engineers dealing with a specific problem may have to consider some special properties that are not listed here *e.g.* permeability, radio-activity, etc. In this chapter we shall concentrate only on those physical properties which have a bearing on the strength of materials.

Specific gravity of a substance or body is too familiar a term to need any clarification here.

$$\text{Sp. gravity} = \frac{\text{Weight of a given volume of a substance}}{\text{Weight of same volume of water}}$$

It will be noticed that the sp. gr. is a ratio and it has no units.

Elasticity : It is the property by which a material regains its original size, shape and strength after the removal of the force which caused the change of form. A rubber band is a common example of an elastic material. All metals possess elasticity, though to a very small degree.

Plasticity : If a lump of plasticine or moist clay is pulled out or squeezed, it retains its new shape when released. This property of the material i.e., retaining the shape when the external force is removed, is called plasticity and the material is said to be plastic. The property is opposite of elasticity. This property of the material is utilised for forging, stamping, etc.

Ductility and mallebility : The ductility of a material is that property which enables is to be drawn into wire or tube. Ductility is associated with plastic deformation of metals and it requires both plasticity and tensile strength. Ductility may be defined as the ability of a material to undergo deformation beyound elastic limit. Lack of ductility is brittleness.

Ductility is a quality which consiitutes some measure of protection againstt suden shock load *e.g.* the loads developed at the start of winding or hauling. Generally speaking a high tensile strength is accompained by low ductility and vice versa. A ductile material shows considerable elongation before fracture occurs and this can provide warning of impending fracture.

Some common metals in order to decreasing ductility are : W, Au, Ag, Pt, Fe, Ni, Cu, Al, Zn, Sn.

Malleability is a property which enables a material (1) to be drawn into thin leaves by hammering or (2) to be worked into various shapes by the process of hammering or rolling. Malleability is dependent on plasticity and not on tensile strength.

Toughness : It may be defined as resistance to impact loading. It is a Property which combines high elastic limit with ductility. A material is a said to be tough when considerable energy is required to fracture it and the fracture is associated with local deformation.

Hardness : One has only to see a dictionary to know the various meanings attached to the term *hardness,* which may mean resistance to indentattion, abrasion, wear, scratching, cutting or machining. In geology hardness denotes the ability to resist scratching and a material which can scratch another is said to possess more hardness than the latter. In engineering, hardness of a material refers to its ability to withstand scratching or indentation or its ability to scratch or cut other material. The relative hardness of material may be tested in several ways, one of which is the Brinell test. In this test a 10mm steel ball is pressed into the material under a standard load (3000kgf for ferrous metals) and the depression caused is measured.

The Brinell number is the ratio of the force applied through the standard steel ball to the area of indentation which is thereby produced in the sample tested. *Softness* is the opposite of hardness.

Brittleness :

A material is said to be brittle when it fails to yield and change its shape if subjected to a stress but breaks suddenly instead. Glass is hard but at the same time, brittle. Brittleness may be regarded as reverse of ductility.

Resilience :

It is a measure of ductility of a material and hence its ability to withstand shock load or suddenly applied loads. It is work required to produce a given stress. The stress set up in a material by a given load depends not only on its magnitude but also one the way in which it is applied. A suddenly applied load causes more stresses than a steady and gradually applied load. A material subjected to load within elastic limit has some energy stored in it and behaves like a stretched spring; the energy stored in equal to the work done in stretching it. The stored energy is called *resilience* and it is represented by the area of the traingle OAM in Fig. 6.6 Where A is the elastic limit.

Stress and Strain :

The combination of all the forces acting on a body is called the *load.* The change in dimensions which occur in a body due to application of load or force is called *strain.* When strain is produced in a body, a system of internal forces, called stresses, come into play, so as to resist the strain, and the exact form of stress and strain which occur depends on how the load is applied. The strain may be visible or it may be so small as to measured by only precise instruments. It is obvious that the degree of stress will depend on the load applied and on the area of cross-section of the body that is resisting the load. Stress, is therefore, measured in terms of Newtons per sq. metre (N/m^2) or N/cm^2. N/m^2 is the basic S.I. unit of stress.

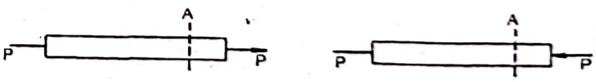


Fig. A rod of cross-section A subjected to a force P (left) tensile (right) compressive.

Strain is defined as the extension, shortening or deformation of unit length. Thus, if a steel rod, originally 3m long is stretched by 2mm by a given load, the strain is 2 ÷ 3000 or 0.66×10^{-3}. The strain is, therefore, a ratio and is denoted by a number; it has no units.

There are 3 kinds of stresses and strains; tension, compression and .ear. Bending and torsion are combination of these, though sometimes they e classed separately.

ension and tensile stress :

A weight hanging from a rope tends to pull its fibres apart and the pe tends to strech. The rope is said to be in tension.

When a force is applied to a body in such a way as to cause its length, easured in the direction of the force, to increase, the body is said to be in a age of tension and the force producing tension is called tensile force. The ress and strain produced by application of external force are referred to as nsile stress and tensile strain respectively.

Tenacity is the resistance which a body offers to being called as under y a tensile force or load.

ompression and compressive stress :

When a body is subjected to a load or force which tends to squeeze it nd shorten its length or size, it is said to be under compression and the force known as compressive force. The stress set up is the compressive stress and ie corresponding strain produced is known as compressive strain. A beam aded at its upper surface has its upper surface under compression and lower urface under tension. A tensile as well as a compressive force acts at right ngles to the cross-sectional area of the body subjected to such force.

.ateral strain and Poisson's ratio :

When a body is subjected to a direct load (i.e. tensile or compressive), undergoes a strian in the direction of the applied force and an opposite kind f strain in every direction at right angle to it. All axial strain, tensile or ompressive, is accompanied by lateral strain. For example, the length of a tie ar will increase under an axial pull but its cross-section will decrease. column under axial load will shorten but its section will slightly increase. his lateral strain, or strain of the sides, is a fraction of the linear strain and ears a constant ratio to the linear strain (within the elastic limit).

Thus, $\dfrac{-\text{lateral strain}}{\text{linear strain}}$ = a constant within the elastic limit.

This constant is known as *Poisson's ratio* and is usually denoted y $\dfrac{1}{m}$. The value of m lies between 3 and 4 for most metals so the Poisson's atio varies from 0.34 to 0.25.

Shear Stress :

Shear stress is transverse stress across a plane of the material. When a material is under the action of a shear stress, there is a force tending to make one surface slide over the other. Its effect on the material can be understood by laying a thick book flat on the table, placing your hand on top of it, and with the hand pressing, simultaneously pushing the book pages horizontally (Fig. 6.1). The effect is to make adjacent pages slide over one another so that the whole book becomes a parallelogram in end view instead of a rectangle.

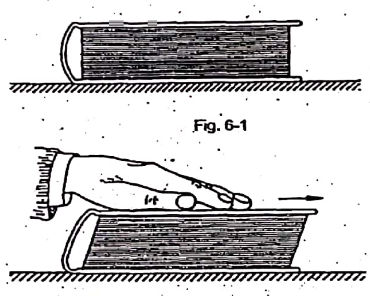


Fig. 6.1

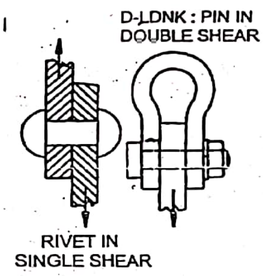


Fig. 6.2

When a body is subjected to two equal and opposite forces P acting tangentially across its section, as in the case of section of a rivet holding together two plates in tension, (Fig. 6.2) the internal resistance set up is called shear stress. Shear stress acts parallel to the cross-section area.

Measurement of stress and strain :

If P Newtons is the tensile (or compressive) force applied to a bar and the area of the bar at right angles to the direction of force in m² is A, the intensity of tensile (or compressive) force, p, is $\dfrac{P}{A}N/m^2$. The tensile (or compressive stress) set up within the bar is also $\dfrac{P}{A}N/m^2$. If the original length of the bar is L and deformation in length of bar is *l*, the bar will have a length of L + on the application of tensile force and L-*l* on the application of compressive force. In either case strain $e = \frac{l}{L}$ which is only a ratio.

The value of $\frac{\text{stress}}{\text{strain}}$ is constant for a given material and it is called Young's modulus or modulus of elasticity, denoted by E.

$$\text{Young's modulus}: E = \frac{P}{A} \times \frac{L}{l} \times \frac{P}{e}$$

Although modern thought now links Young's Modulus of elasticity with tensile loads, it is found that the ratio $\frac{\text{compressive stress}}{\text{compressive strain}}$ for many materials is little different from the ratio $\frac{\text{tensile stress}}{\text{tensile strain}}$ and unless informed specifically to the contrary, E can be presumed to have the same value for a given material whether it be loaded in tension or in compression, and this is a fundamental assumption in the strength of beams and columns. Typical values of the tensile strength and modulus of elasticity of different materials is given in table. The value of $\frac{\text{stress}}{\text{strain}}$ is constant only upto elastic limit, explained later.

The intensity of shear stress $q = \frac{R}{A}$ where R is the resistance to external force P set up by section X X, having A as the area of cross-section. R = P and $q = \frac{P}{A}$. The corresponding deformation or shear strain is measured by the angular deformation accompanying shear stress. A rectangular block A B C D, (Fig. 6.3) when subjected to shearing forces as shown, may be distorted to assume the position ABC_1D_1 and the shear strain is measured as the angle CBC_1 in radians. Since it is always very small, it may be taken equal to the tangent $CC_1 \div BC$.

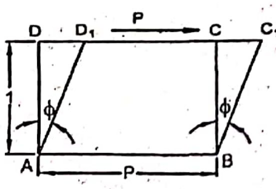


Fig. 6.3

Shear stress is also produced by torsion and has a bearing on the strength of revolving shafts transmitting power. Shear stress is proportional to shear strain for a material within certain limit of stress, and the ratio $\frac{\text{Shear stress}}{\text{Shear strain}}$ is constant. This constant ratio is called the modulus of elasticity in shear (symbol G or C) or modulus of rigidity.

$$\text{Thus } G = \frac{q}{\phi}$$

For steel G is nearly 80×10^9 N/m² and for rubber it is 10^6 N/m².

Consider the action of a punch. If it is hammered into a metal plate of thickness t mm, it pushes out of the metal a small disc or cylinder having a height of t mm. During the process the surface of the cylinder which slides over the surface of the metal plate is in line with (*i.e.*, parallel to) the applied force P and it is equivalent to the circumferential surface of the *cylinder*. This area = πDt if D is the diameter of the punch and it is the area which resists shear force.

$$\text{Shear stress} = \frac{\text{Load}}{\text{Shear area}} = \frac{P}{\pi Dt}$$

If the compressive stress is required, then the area under compression is the area of the top of the cylinder at right angles to the load *i.e.* $\frac{\pi D^2}{4}$

$$\text{The compressive stress} = \frac{\text{Load}}{\text{Area}}$$

$$= P \frac{\pi D^2}{4}$$

$$= \frac{4p}{\pi D^2}$$

Example :

The cover of a cylinder is to be secured by bolts having a shank diameter of 10mm. The cylinder has a diameter 80mm, and the pressure inside the cylinder is 7×10^5 N/m², Calculate the minimum number of bolts to be used if the stress in their shanks is not to exceed 14×10^6 N/m².

ns. :

Force = pressure × area

∴ Force due to pressure on cylinder cover

$$\frac{\pi}{4}\times\frac{80}{1000}\times\frac{80}{1000}\times 7\times 10^5\ \text{N} = 3518.6\ \text{N}$$

$$\text{Permissible stress} = 14\times 10^6\ \text{N/m}^2 = \frac{\text{Load}}{\text{area of shank of all bolts}}$$

Cross-sectional area of shank of each bolt

$$= \frac{\pi}{4}\times 0.1\times .01\ \text{m}^2$$

Let number of bolts b n

Area of cross-section of n bolts = .0000785 n m²

$$14\times 10^6\ \text{N/m}^2 = \frac{3518.6\text{N}}{n\times .0000785\ \text{m}^2}$$

solving, we get n = 3.20

Thus minimum no. of bolts is 4.

Example :

A tensile load of 100 kN is applied to the end of steel bar having a cross-section of 5cm × 2cm. What is the tensile stress in the bar ? If the bar is 1.5mlong when unloaded, calculate the increase in length when the load is applied. The value of Young's modulus of steel is 205 × 10^6 kN/m².

Ans. :

$$\text{Tensile stress}\ \ P = \frac{P}{A} = \frac{100\ \text{kN}}{5\times 2} = 10\ \text{kN/cm}^2$$

$$= 10^5\ \text{kN/m}^2$$

$$\frac{P}{e} = E \qquad \therefore\ e = \frac{P}{E}$$

$$\text{Hence strain}\ \ e = \frac{10^5\ \text{kN/m}^2}{205\times 10^6\ \text{kN/m}^2}$$

$$= 0.488\times 10^{-3}$$

$$\text{but}\ \ e = \frac{\text{elongation}\ l}{\text{origina length, L}}$$

∴ elongation $l = e\times L = 0.488\times 10^{-3}\times 1.5\times 10^3$ mm

= 0.73 mm.

Example :

A cast iron column has a hollow circular cross-section, outside diameter being 380mm. It is to withstand a force of 1.8MN. When loaded the column has a length of 1220mm. If the working stress is not to exceed 83 MN/m², find the minimum thickness of material and the initial length of the column, assuming Young's modulus for cast iron 120 GN/m².

Ans :

Area of the column under load

$$= \frac{\pi}{4}\left(380^2 - d^2\right)\text{mm}^2$$

Stress p = 83 MN/m² = 83 N/mm²

$$\therefore\ \text{Area} = \frac{1.8\times 10^6}{83} = 21690\ \text{mm}^2$$

$$21690 = \frac{\pi}{4}\left(380^2 - d^2\right)$$

Solving this, d = 342mm and minimum thickness

$$= \frac{380-342}{2} = 19\text{mm}$$

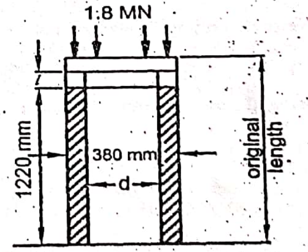


Fig. 6.4

If initial length is L and deformation is l strain $= \frac{l}{L}$

$$E = \frac{\text{stress}}{\text{strain}} = 120\ \text{GN/m}^2 = 120\times 10^3\ \text{N/mm}^2$$

$$\text{strain} = \frac{83}{120\times10^3} \text{ but strain} = \frac{l}{l+1220}$$

$$\frac{l}{l+1220} = \frac{83}{120\times10^3}$$; From this we get l = 0.85 mm Initial length

1220.85mm.

xample :

Calculate the shear stress in a 1cm diameter steel rivet if a shear force 5kN is distributed uniformly over its cross-section. If the shear modulus of eel is 80 GN/m², calculate the shear strain.

ns. :

$$\text{Intensity of shear stress, } q = \frac{5kN}{\pi\times0.5^2\times10^{-4}\,m^2}$$

$= 63.7 \times 10^3\ kN/m^2$

$= 63.7\ MN/m^2$

$$\text{Shear strain } \phi = \frac{q}{G} = \frac{63.7\times10^6 N/m^2}{80\times10^9\,N/m^2} = 0.796\times10^{-3}$$

Example :

A hole 25mm diam. is to be punched out in a metal plate 12.5 mm thick by a punching force of 10kN. Calculate (a) the shear stress set up inthe material, and (b) the comprssive stress in the punch.

Ans. :

Area resisting shear = circumference of hole × thickness of plate

$= \pi \times$ diam × thickness

$= \pi \times 25 \times 12.5$

$= 981.75mm^2$

$$\text{Shear stress} = \frac{\text{Shear load}}{\text{Shear Area}} = \frac{10{,}000\,N}{981.75mm^2} = 10.186\ N/mm^2$$

The area resisting the compressive force is the area of the punch at right angles to the direction of applied force

$$= \frac{\pi}{4}\times(\text{diam. of punch})^2 = \frac{\pi}{4}\times(25)^2$$

$= 490mm^2$

$$\text{Compressive stress} = \frac{\text{Load}}{\text{Area}} = \frac{10{,}000\ N}{490\ mm^2} = 20.41\ N/mm^2$$

Typical values for the tensile strength and modulus of elasticity of materials.

Material	Tensile Strength MN/m2	Modulus of Elasticity; GN/m²
Aluminium alloy	320–480	70
Brass	420	100
Bronze	540	120
Carbon Steel	590	205
Magnesium alloy	340	45
Mild steel	450	205
Nickel steel	1220	205
Polythene	15	[illegible]
Stainless steel	1275	195
Timber	125	12.5
Titanium alloy	1020	105

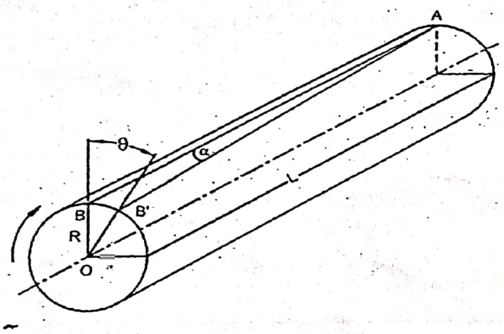


Fig. 6.5 Illustrating torsion.

Torsion :

When a shaft is subjected to a twisting moment or torque, the stress set up in it is pure shear stress and the shaft is said to be under torsion. The torsion equation of a circular shaft (solid on hollow), in its simplest form is,

$$\frac{q}{R} = \frac{G\theta}{L} \qquad \text{(Fig. 6.5)}$$

Where q = shear stress in N/m² at a radius R meters from axis

G = Modulus of rigidity (shear or torsion modulus) in N/m².

θ = angle of twist in radians in a length of L meters.

This expression holds at any radius in the shaft and hence their stress at any point in a shaft is proportional to its distance from the axis, the maximum being at the outside edge, and zero at the centre. The shaft axis is therefore a neutral plane analogous to the neutral plane in a loaded beam subject to tensile and compressive stresses.

Stress-strain diagram for ductile material in tensile test :

If a ductile material is subjected gradually to increasing tensile load, the resulting strain increases proportionately to the applied load (and the equivalent internal stress) upto certain limit. The relation of stress and strain is represented graphically in Fig. 6.6 for a short test piece (usually 200 mm long) of mild steel when conducting the tensile test.

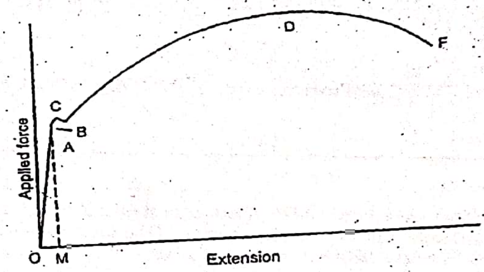


Fig. 6.6 Stress-strain curve for mild steel

The limit upto which elongation or strain is directly proportional to the load applied is called *elastic limit* or limit of proportionalitty. Upto this point, A, the material is truly elastic, and the line OA is straight. Beyond A the strain increases more quickly with the increase of stress upto the point B, known as yield point, where the material suddenly elongates faster with very little increase in stress. Beyond the yield point the material becomes plastic and elongates rapidly. The stress represented by the point B is called the yield stress and is the maximum which a material can withstand without suffering permanent damage or permanent set. That part of the strain whichdoes not entirely disappear on the removal of the stress, is termed *permanent set*. Further increase of load and the resultant strain is depicted by the curve CD where D incidates the highest stress which the material can withstand. The stress at D, calculated on the original diameter of the test piece, is called the maximum stress or ultimate tensile stress or simly the tensile strength. At D local yielding takes place and the material begins to form a waist; with the formation of waist, the load necessary to break the material is less than the ultimate stress, as shown at F. The graph OABCDF is with *nominal stress* calculated as the ratio of load ÷ original area of cross-section.

For all commercial pruposes, the elastic limit is taken as being the same as yield point.

The region OA where the strain is directly proportional to stress conforms with Hooke's *Law* which states : the stress produced in a body under tension is proportional to the strain provided a certain maximum stress is not exceeded. Hook'es Law is applicable only upto limit of proportionally or elastic limit A so that $\frac{\text{stress}}{\text{strain}}$ = a constant. In other words, $\frac{P}{e} = E$, a constant Robert Hooks after whom the law is named was a scientist of the 17[th] century. With the materials discovered since the time of Hooke, and using more sophisticated appartus which enables measurements to be made very precisely, it has now been found that Hooke's Law is not very accurate for a number of engineering materials many of which are quite elastic beyond the point at which stress is proportional to strain.

The largest load, repeatedly applied, which a piece will carry without taking permanent set is called the *proof load* and the corresponding stress is the *proof stress* or proof strength.

Fatigue resistance and endurance limit :

When metals are subjected to rapid reversal of stresses, *i.e.*, alternate compression and tension, failure occurs at much lower stress than the corresponding static stress. The phenomenon is called fatigue failure which may even occur at a stress lower than the elastic limit. The fatigure strength is the maximum stress a metal can withstand without breaking or without failure for a specific number of stress reversals. The lower the applied stress, the greater is the number of reversals that a metal can withstand.

Creep :

This is the gradual plastic deformation of metal over a long period of time when subjected to a stress below the yield point. Metals generally creep at high temperature, (100°C to 500°C) but lead creeps even at room temperature. It is measured as percentage elongation when subjected to static stress for a specified number of hours, usually 1000 or 2000 hours.

Factor of Safety :

The greatest resistance or stress which a material is called upon to put up-in other words, the stress which a material will be worked to-is called the working stress. It is naturally much less than the ultimate stress and is kpet well within the elastic limit to avoid the risk of permanent deformation.

The ratio $\frac{\text{ultimate stress}}{\text{working stress}}$ is called the *factor of safety*. The factor of safety of any component of a machine is decided by the designer ...ing into consideration

i. strength, uniformly and reliability of the material,

ii. kind of stress and character of loading.

iii. possible consequences of a breakdown.

iv. certainly with which amount of load is known

v. character and life of structure.

Thus winding engine components and the winding rope have a higher factor of safety than those of haulage engines or stationary machines whose breakdown does not involve loss of life or expensive damage. The factor of safety is less a temporary structure than for permanent ones. It is more for alternative stresses than for direct stresses. For dead loads the factor of safety may be as low as 4 but for live or fluctuating loads or where the load cannot be estimated correctly or where corrosion may ultimately weaken the structure, it may be as a high as 20. A high factor of safety results in bulky or unwidely components of a machine. For the ropes used in cities for hoisting lifts and winding ropes of men-winding engines, the factor of safety is between 8 and 10. In cranes or other hoisting/lifting equipment, it is 4. For the haulage ropes in mines the factor of safety is usually 6 or 7.

The factor of safety is sometimes facetiously reffered to as a factor of ignorance to cover up the ignorance in the estimation of stress, in the imperfections of the material of construction, etc. Modern tendencies favour a list of safe working stresses for a given material in terms of typical applications.

Stress in hollow cylinders :

The stress produced in the walls of a cylindrical vessel by an internal pressure of steam, compressed, it or gas or by external pressure is calculated as follows :

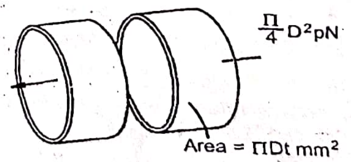


Fig. 6.7

Let the diameter be D mm, and the walls bet mm thick and let the pressure be P Newtons per mm². The force tending to burst the vessel is equal to the total pressure on the ends or $\frac{\pi}{4}D^2p$ Newtons. This force is resisted by an area of the material πDt m². (Fig. 6.7)

$$\text{Stress} = \frac{\text{Load}}{\text{Area}} = \frac{\pi}{4} \times D^2 \times p \times \frac{1}{\pi D_t}$$

$$= \frac{Dp}{4t} \text{ N/mm}^2$$

The stress tending to burst a shell, such as a boiler or a com-pressed air receiver along a longitudinal joint can be shown by calculations to be $= \frac{Dp}{2t}$ Newtons per mm²; in other words, double that around a circumferential joint. For this reason it is essential to make longitudinal joints stronger than circumferential ones.

QUESTIONS

1. Explain meaning of the following terms : tensile stress, ductility, hardness, elastic limit, resilience, Poisson's ratio.

   A rod of metal, 11.3mm dia. was subjected to a test load of 7.6 metric tonnes to find the tensile strength. The rod withstood the load. Calculate the tensile strength of the metal in MN/m².

   (Ans. 746 MN/m²)

2. What is Young's modulus ? A bar of steel, 40mm × 20mm in cross-section and 4m long, is subjected to a tensile load of 32000 N. How much will the rod extend if the modulus of elasticity of steel is 200×10^9 N/m². Give the answer as a decimal fraction of millimeter.

   (Ans. 0.8 mm).

3. A mild steel column has an outer diameter of 0.22m and is 3m long. When subjected to axial compressive load of 895 kN there is a shortening of 2mm. Calculate the thickness of wall of the column. Take E = 203 GN/m2.

   (Ans. : 0.01 m)

4. A pipe 2.5m long having outside dia. 80mm and inside diameter of 60mm is used as a strut and subjected to a compressive stress of 16 MN/m². Calcualte.

   (a) the amount of shortening of the strut if E = 200×10^9 N/m². Give the answer in fraction of a millimeter.

   (b) the load on the strut in kgf.

   Ans. : (a) 0.2mm (b) 3590 kgf.

5. Explain shear stress and shear strain with diagram. It is required to punch holes of 45mm dia. in a mild steel plate of thickness 3mm. If the shear yield stress of the steel is 250 MN/m², calculate the punching force required.

   (Ans. : 106 kN).

6. Define safety factor and state why it should be considered in designing machine components.

   What is the value of safety factor for ropes used :

   (a) for lifts in cities

   (b) for ropes used to wind men in mines

   (c) for haulage ropes.

O O O

CHAPTER – 7

ENGINEERING MATERIALS; METALS

The materials used for engineering purposes are :

1. Iron and Steel : These are also called ferrous metals (from Latin *ferrum* = Iron) and are used on a large scale for engineering requirements.
2. Non-ferrous metals such as, brass, copper, aluminium, tin, etc.
3. Wood and timber, plastics, rubber, etc.
4. Building materials like bricks, cement, concrete, etc.

This chapter will be devoted only to metals and alloys.

A metal in its pure state is or soft and relatively weak. It is, therefore rarely used in the pure from for engineering purpose. To enhance some of its properties like hardness, tensile strength, corrosion resistance, etc. a pure metal is melted with small additions of other metal or substances and the resultant metal is called an alloy. An alloy is defined as a metal which is a mixture of two or more elements (at least one of which is a metal) that mix together when molten, and do not separate upon cooling. The properties of an alloy are much different from those of the elements which make it. Generally speaking alloys are harder and stronger than the elements they contain. Brass is an alloy of copper and zinc. The addition of zinc to copper gives an alloy harder than either of the elements, copper or zinc. Aluminium, *e.g.*, is light in weight and has poor tensile strength; by adding about 4% copper to it an alloy called. Duralumin is produced which has a tensile strength of 385 MN/m^2 to 510 MN/m^2. If thus has nearly the same tensile strength at mild steel, but is only 1/3rd in weight. One of the effects of impurities (and also of the deliberately added metal or substance in an alloy) in a metal is the lowering of the melting point. Addition of carbon to iron proressively lower the melting point (from 1530°C for pure iron) by approx 100°C for each 1% carbon.

The melting point of lead is 327°C; that of tin is 232°C. Yes when a solder contains about 66% tin and 34% lead, its melting point is as low as 180°C.

FERROUS METALS :

Iron is the most important material in engineering industry. Ordinary iron, as we understand it for engineering and construction uses, contains not only the element Fe, but other non-metallic materials, chiefly carbon, silicon, phosphorus and sulphur and also some metals such as manganese, etc. in small quantities. It is produced by heating the ores of iron like hematite, magnetite, etc, with hard coke and limestone in a kiln called blast furnace. Coke serves as the fuel to generate heat for smelting and it also provides a reducing atmosphere so that the oxide ore is deprived of its oxygen and reduced to the state of molten metallic iron mixed with impurities derived from the ore, coke and limestone. The limestone acts as a flux which combines with the impurities in the ore to form a slag that floats on the molten iron. The molten iron and slag are tapped separately at intervals from the bottom of the blast furnace through tap hole. The tapped molten metal is allowed to cool in moulds or blocks and is called *pig iron*. It contains 3-4% carbon and other impurities like sulphur, phosphorous, silicon, manganese, etc. Sulphur and phosphorous are undesirable impurities in ferrous metals. These pigs are slabs of iron in its crude form which is further processed to get cast iron, wrought iron, or steel.

There are three chief kinds of ferrous metals, viz.

1. **Cast iron :** containing between 1.8% and 5% of carbon.
2. **Wrought iron :** containing less than 0.15% of carbon and differing from low carbon steel in the method of manufacture.
3. **Steel :** containing less than 1.8% of carbon and rarely more than 1.4 %.

Cast iron : Articles of cast iron are made by melting pig iron in cupolas or small furnace with coke and air blast and then pouring into moulds of the desired shape. In cast iron, the carbon is present partly in mechaical mixture as free flakes of graphite and partly combined as iron carbide (Fe_3C). Cast iron is easily melted and can be cast in moulds of any required shape and is, therefore, often used for casting to intricate shape. It is fairly hard, but brittle. It has low tensile strength, 100 MN/m^2 to 180 MN/m^2 but high compressive strength, 615 MN/m^2 to 750 MN/m^2. It cannot be forged but is machineable. It is not easily amenable to welding and is unsuitable for resisting high tensile or alternating stresses and is mainly used where weight is not objectionable, but rigidity is essential, *e.g.* bed plates of machines, flywheels, pulleys, steam and air engine cylinders, columns, stator frames of generators or motors, low speed gear wheels, fire grates, *etc.* Phosphorous renders cast iron more fluid but reduces the strength of castings and it should not exceed 0.75% in important parts.

Malleable cast iron : is produced by heating a special kind (white cast iron) of iron casting in some oxidising material for several days and retaining it at a certain temperature. It is cooled slowly over similar period. Compared to cast iron the malleable cast iron is less brittle, more ductile and has more tenacity as the carbon percentage is reduced. Tensile strength is 300 MN/m^2.

A special kind of cast iron is known as spheroidal graphite iron. In it, the carbon (graphite) is present in the spheroidal form and not in the "flake" form. Compared to cast iron, the spheroidal graphite iron has more tensile strength and ductility. It is much less brittle than cast iron and deforms before breaking. Like cast iron, it also can be easily moulded into intricate shapes by casting.

Wrought iron : It is the purest commercial form of iron and consists of almost pure iron in chemical combination with less than 0.15% carbon. It is manufactured from pig iron in an open hearth furnace, called a puddling furnace, by heating in contact with iron ore where nearly all the carbon and other impurities are eliminated. Wrought iron is ductile, malleable, has high tensile strength (325-390 MN/m^2), easily weldable and machineable, and withstands repeated hammerings, shocks and forgings without deterioration. It resists corrosion better than mild steel. It is used chiefly for out-door ornamentation of buildings, for hauling and lifting gear and manufacture of chains, tub coupling, drawbars, bolts, etc. It requires annealing to remove the hardened skin that tends to develope on surfaces exposed to wear. Wrought iron cannot be hardened. It has high magnetic permeability and finds large application in the manufacture of electromagnetic apparatus such as pole pieces of the field magnets in motors and generators. For many uses, wrought iron has been replaced by mild steel which is easy and cheap to manufacture and stronger than wrought iron.

Steel : Steel contains upto 1.8% carbon and is manufactured from pig iron by remelting the latter in open hearth furnace where excess carbon is removed by blowing hot air and gases over the molten surface. Steel is produced in the open hearth furnace by either the *acid process* or the *basic process*. In

steel as well as in wrought iron, the percentage of sulphur should not exceed about 0.06 as it causes red shortness *i.e.* that metal cannot be worked at or above red heat and when it is red hot it becomes brittle and cracks under hammer. Sulphur also renders the metal more difficult to weld. Presence of phosphorus in wrought iron and steel causes cold shortness *i.e.* the metal is brittle and cannot be worked when cold, although it works well when heated. In steel, the percentage of phosphours should not exceed 0.06 though in wrought iron a higher limit, not exceeding 0.2%, is permissible.

The word *steel* when used without qualification, always refers to plain carbon steel containing carbon in chemical combination upto 1.8% with small percentages of silicon, manganese, sulphur and phosphours. It differs from wrought iron in having no slag inclusions, has high carbon content and the processes for manufacture are different Mild steel, also called structural steel, contains less than about 0.5% carbon (generally 0.15 to 0.25%). It is ductile, has high tensile strength (460 MN/m^2) and high cmpressive strength, can be forged and welded but m.s. with low carbon % cannot be hardend. Mild steel is used for manufacture of boiler plates, rolled joists, rivets, tin plates, C.G.I. sheets, wires and a variety of other products. Unlike wrought iron steel is easily rusted.

Cast steel (carbon steel) : Cast steel is sometimes referred to as carbon steel although all steels contain some carbon percentage. According to one broad classification steel containing more than 0.5% carbon is called cast steel, hard steel or simply carbon steel. It is extremely strong and hard but with lower toughness and ductility and a tensile strength of 700-850 MN/m^2. In steel as the carbon content increases hardness, brittleness, fusibility and tensile strength are increased but ductility, toughness and welding power are diminished. Hardness however, may be modified by heat treatment with corresponding changes in tensile strength and ductility. Cast steel is usually made by what are termed the *electric* furnace or crucible processes which differ from the method of manufacture for structured or mild steel. It is unsuitable for forging and cannot be easily welded. Steel casting are usually *annealed* prior to use, *i.e.*, they are rendered softer by heat treatment. Steel with 1.25% or more carbon is unweldable. High carbon steel (0.45% to 0.65% carbon) is used for railways tyres, rails, laminated springs, keys and wire for winding and hauling ropes. Steel used for making tools. (tool steel) like hammers, axes, chisels, taps, dies, lathe tools, saws, files, surgical instruments, generally contains 0.65 tto 1.5% carbon.

Heat treatment of iron and steel :

The object of heat treatment of iron or steel materials is to develope in the material a structure most suited to the purpose desired. The processes of heat treatment include :

1. Forging.
2. Welding.
3. Annealing.
4. Normalising.
5. Hardening.
6. Tempering

Forging : This is the process of heating steel to a temperature of 700C–1100C and hammering it, when it is still hot, on an anvil to shape it do desired size or shape. Below the temperature mentioned here the metal is liable to become brittle and is not in a fit state for working. If heated beyond 1100°C the metal gets burnt.

Welding : Welding is union of metal parts. Welding by forging and hammering of white hot iron or steel materials is carried out at about 1200°C for wrought iron and 1100C for mild steel and is a common process in a blacksmith's shop. If the welding by forging is done properly, new crystals are developed across the line of contact of the two pieces and the joint is not visible. The hammering to forge two steel pieces is continued down to a temperature of about 700C as indicated by cherry red colour. Smith welding or forge welding is not applicable to cat iron. It is used only for iron and steel parts of simple shapes and small cross-sections *e.g.* chains, drawbacks, pipes, etc. and it is largely used for small repair jobs at the mines. Wrought iron welds more readily than mild steel. A flux of sand (or sometimes of borax is used during such welding.

Annealing : It is a process of heating the metal to normalising temperature and then allowing it to cool slowly and uniformly not in air, but in the furnace or in a bed of hot sand, lime or ashes. The object of annealing is to remove internal strains caused by previous heat treatment or by mechanical working when cold, *e.g.* annealing of work-hardened surface layers of cage chains. Annelaing restors the material to a soft state and restores its ductility and this facilitates easy machining and mechanical working. For some jobs low temperature annealing is adopted. It consists of heating the material to a temperature lower than the critical temperature (viz 650° C for mild steel and 700°C for wrought iron), keeping it at the temperature for 15–20 minutes and then cooling it slowly in a bed of hot sand, ashes or line. Low temperature annealing restores softness, ductility and removes internal stresses but it does not refine the crystal size. In mining, cage chains and drawbacks have to annealed at intervals but if they are made of mild steel containing about 1.5% manganese such periodical heat treatment is not necessary.

Normalising : It is a process of heating steel to a temperature slightly above the upper critical temperature for about 15–20 minues and allowing it to cool *naturally in air.* The critical temperature referred to here is about 900°C for pure iron and about 700°C or 0.80% carbon steel and depends upon carbon content in the steel. More carbon percentage means lower temperature. The objectof normalising is to reduce the size of crystal grains. The process restors the material to a soft, ductile condition and relieves it of internal stresses.

Hardening and tempering : In the process of hardening the steel is heated gradually and uniformly to a temperature *higher* than the critical temperature (normalising temperature) and then cooled *rapidly* by quenching in water or oil. Hardening is possible in steel containing carbon so that pure iron, wrought iron or low carbon steel cannot be hardened. The degree of hardness obtainable depends upon :

1. Percentage of carbon. Higher percentage gives higher degree of hardness.
2. Rate of cooling. Oil quenching is slower compared to quenching in water and more hardness is possible by water quenching. Hardness, however, introduces some degree of brittleness.

The process of hardening is applied to (a) tips of coal cutting picks and othe tools used for cutting where hard cutting edge is required. (b) Working parts of machinery subject to high stresses *e.g.* piston rods, axles, crank shafts, etc. The steel hardened by quenching is in a state of severe strain and to render it fit for use, it should be *tempered.* The purpose of tempering is to remove strains caused by hardening and to soften and toughen the steel to the desired extent. The process of tempering consists in re-heating hardened steel to a temperature *below* the critical temperature and then cooling it more or less rapidly in either water or oil, depending upon the degree of hardness required.

As the tempering temperature is gradually raised (but kept below the critical temperature) the ductility and toughness increases but hardness and tensile strength decrease. Engineering steels are tempered to a temperature ranging from 400 C to 650 C depending on the material and hardness required. Drill steels and miner's picks, after they have been reshaped by forging at about 1000 C and allowed to cool slowly in air, are generally hardened and tempered in one operation. These tools generally are of plain carbon steel with 0.60 to 0.75% carbon.

Case hardening : Some parts of machines and their components should have a surface which is hard to wear but the core or the interior should be tough to resist shocks. Such parts are gear wheels, crank shafts, ball and roller bearings, hammer drills, pistons gudgeon pins etc. The wearing surface only are, therefore rendered hard by suitable processes and are then said to be case-hardened. Case hardened parts are made from mild steel or low carbon steel (0.1 to 0.15% carbon). Sometimes small percentage of nickel or chromium may be added to the steel. For case hardening the parts, aftter machining to the desired shape, are completely covered in a carburising material rich in carbon and then heated in a box for 8–12 hours to a temperature of 900–950°C. The surface of the steel part gradually absorbs carbon to a depth of 0.25–3.5mm and the high carbon (nearly 0.9%) in the skin renders it wear resistant. The piece of mild steel case-hardened in this way, is observed to posses a very hard wear resistant surface while the centre core retains its original tenacity and ductility. The carburising medium is bone charcoal, leather, petroleum coke mixed with a little barium carbonate, potassium cyanide or sodium cyanide. If a case-hardened object is reheated to a temperature not exceeding 300°C and then quenched, the hardness of the outer layer is hardly affected but its brittleness is considerably reduced.

Alloy Steel :

These are steels to which small quantities of other elements, outer than carbon, have been added to increase the strength or to enhance some other desired property. Carbon, however, remains and essential and most important ingredient in most cases. The elements commonly used for making alloy steels are manganese, chromium, nickel, molybdenum, tungston, silicon and cobalt. To a lesser extent lead, copper, vanadium and other metals are also used. Some of the typical alloy steels and their uses are given below :

1. Manganese Steel : Manganese in small percentage is present in all steels, but the term 'manganese steel' refers to steel having 11-14% manganese and 1 to 1.5% carbon. Such steel has high tenacity, ductility, high degree of hardness and resistance to wear and abrasion. It is much used for tube wheels, railway points and crossing, jaws of stone crushers, etc. Mild steel containing 1.5% manganese is used for cage chains, tub couplings, drawbars and does not require periodical heat treatment. Manganese steel is practically non-magnetic and can be easily cast, resembling cast iron in this respect. It is almost unmachineable and holes cannot be drilled in it by normal methods. Therefore, holes formed in it are usually of the "cored" type *i.e.*. they are formed during casting of the metal.

2. Nickel Steel : This is an alloy steel containing 2–4% nickel with 0.25 to 0.4% carbon. It is stronger and more ductile than mild steel and is used for machine parts requiring high tenacity and toughness, *e.g.* driving shafts of turbine pumps, coal cutters, etc. Nickel steel is also used for turbine blades (5% Ni), tubes of water tube boilers (3%Ni), electromagnetic apparatus

(75% Ni), gears (2.4%Ni). Nickel steel (36% Ni) has very low coefficient of expansion and is used for steel bands, used by surveyors for precise measurement. Steels having over 50% Ni have a high permeability and are employed for electrical purpose and electromagnetic apparatus.

3. High speed steel : This is used for cutting tools and it retains its cutting qualities even at red heat. The essential constituent is tungsten (14–20%); other constituents of a good cutting tool steel are generally chromium upto 4% and small percentage of molybdenum. Tungsten steel is now extensively used for cutting tools in connection with lathe works.

4. Silicon steel : This is an alloy steel with silicon upto 4%. It has high magnetic permeability and is widely used for cores of electromagnetic apparatus, transformers, etc. Springs are made of silicon steel (upto 2% silicon). It has great strength, toughness and resilience. Steel with 15% silicon is used for chemical plants as it has high corrosion resistance against acids.

5. Chromium steel : In this alloy steel, chromimum percentage is 1–2%. Chromium increase the tensile strength and renders the metal hard and resistant to wear and abrasion. It is used for making ball bearings, roller bearings, chisels, jaws of stone crushers, etc.

6. Stainless steel : contains 18–20% chromium and is highly corrosion resistant. It is used for cutlery, machinery parts subject to high stresses, steam turbine blades, exhaust valves of internal combustion engines and other purposes. Chromium is a hardening agent and steel centaining 1% chromum is used for dies and ball race.

7. Nickel chromium steel : This is the most important alloy steel for most engineering purpose. Inclusion of nickel and chromium makes the steel exceptionally strong and resistant to wear, yet easily machineable. The steel is used for parts subject to dynamic loads and shocks.

NON-FERROUS METALS :

The commonly used non-ferrous metals are aluminium (Al), antimony (Sb), bismuth (Bi), copper (Cu), lead (Pb), nickel (Ni), tin (Sn), Zinc (Zn), manganese (Mn), chromium (Cr), and magnesium (Mg).

Aluminimum : It is manufactured from the raw material known as bauxite. Melting point 658°C, sp. gr. 2.70, white in colour, good conductor of heat and electricity; mechanically a weak metal.

Antimony : Melting point 630° C, sp. gr. 6.7, bluish white metal, very brittle; used for hardening lead and tin alloys, *e.g.* white metals.

Bismuth : Melting point 264°C, sp. gr. 9.82 white; brittle; used in some fusible alloys. Bismuth expands on solidifying.

Copper : Melting point 1050°C, sp, gr. 8.8; tough, soft, ductile, malleable with fine reddish colour; very good conductor of electricity and heat and is used extensively in electrical industry. After iron and steel, copper is probably the most important metal in engineering. It can be cast but not easily machined. It has high resistance to corrosion. Copper may be cold-worked readily. All the copper alloys may be annealed by heating the metal and cooling it rapidly in water, this being the exact opposite to that required in the case of steel.

Lead : Melting point 327°C, sp. gr. 11.36; very soft with bluish grey colour, malleable, ductile and tough, but has low tenacity, (less than 30 MN/m^2); used for making pipes, sheets, sheath in paper insulated electric cables, electrodes etc.

Nickel : Melting point 1452°C, sp gr. 8.8, white, malleble, ductile, tenacious, corrosion resistant; much used in chemical engineering.

Tin : Melting point 232°C, sp gr. 7.4; silver white, malleable, ductile, low tenacity (about 30 MN/m^2); used in canning industry as it is resistant to vegetable acids.

Zinc : Melting point 420°C, sp. gr. 7.1; bluish grey metal, brittle, low tensile strength (20 MN/m^2 when cast); harder than tin but softer than copper; resistant to atmospheric influence and therefore, used for galvanising iron articles. It is readily attacked by vegetable acids and is unsuitable for canning purposes.

The non-ferrous alloys in common use in the mining and engineering industry are as follows :

Brass : Alloy of copper and zinc in the proportion of nearly 2 to 1. It can be easily cast into intricate shapes and can be machined, forged and brazed. It is softer than steel and has low coefficient of friction and is, therefore, a useful metal for bearing bushes. It does not corrode but tanishes easily.

Bronze : Alloy of copper (80 to 90%) and tin, the latter being 8 to 10%. Harder bronzes that can withstand wear contain about10% tin; bell metal contains about 23% tin. Bronze is largely used for bearings.

Gunmetal : Alloy of 90% coper, 10% tin. Addition of zinc up to 2% increases fluidity and resistance to corrosion. It can be cast and was formerly used for casting cannons. It is used for bearings, valves, boiler mountings, etc. Church bells are mostly made of gunmetal.

Phosphor bronze : It contains 0.3 to 1% phosphorus in a 90/10 bronze which increases hardness, toughness and resistance to corrosion. It is unaffected by sea water. Phosphor bronze is a good bearing material. It is used for pump impellers, valves springs, wires, steam turbine blades, *etc.* Muntz metal is an alloy of 3 parts copper, with 2 parts zinc and is very resistant to corrosion.

White metal : It is an alloy of lead, antimony, copper and tin in varying proportions. It can be easily melted and is used for bearing mainly for its low coefficient of friction. It will melt and flow out of the bearing housing if the oil supply ceases, so that seizure and damage to the shaft is avoided and timely warning of danger is available. A typical composition of white metal for general purposes is : tin 40–60%, antimony 8–9%, copper 4–4.5%, lead 40–50%. White metal used for filling in wire rope sockets consists of lead 80%, antimony 15% and tin 5%, with a melting point of 260°C.

Plumber's solder : It consists of 2 parts lead and I part of tin; solidifies between 240°C and 180°C.

Babbit Metal : It is a white metal with tin base. Suitable for bearing purposes, its composition is about 80% tin, 3% copper and 10% antimony.

Duralumin : It is an alloy of aluminium containing 4% copper with small percentages of manganese, magnesium and silicon (upto 0.7% each). Its tensile strength is 385–510 MN/m^2 when heat treated.

Monel metal : It is a nickel – copper – iron – manganese alloy (67, 30, 1.25, 1.25 percent respectively of each metal). It does not tarnish, takes a good polish and possesses good heat and acid resisting qualities. It is highly resistant to corrosion. In many respect it is similar to stainless steel and to a large extent is used for similar purposes. It has considerable ductility and a tensile strength of 460 – 620 MN/m^2. It is used mainly for pump-parts, turbine blades and chemical plants.

Widia : Widia is a trade name. It consists essentially of about 94% tungsten carbide, about 5% cobalt and small percentages of manganese, silicon and nickel. After moulding to the desired shape, the material is heated in an electric furnace having a hydrogen atmosphere and it then assumes a natural hardness which approaches that of a diamond, and is not subsequently affected, even at bright red heat. Widia is used for the tips of drill bits of coal cutting picks of special shape, cutter picks of continuous miners, *etc.*

QUESTIONS

1. Give the approximate composition of cast iron, wrought iron and mild steel and describe in brief the made of their manufacture.

2. What is an alloy ? Mention the typical alloys used for (i) cutting tools in lathe work (ii) bearing materials (iii) fusible plug (iv) plumber's solder (v) white metal for sockets of wire ropes, (vi) impellers of turbine pumps, (vii) springs.

3. Write short note on (a) annealing (b) hardening (c) tempering (d) case hardening.

4. Give the compositions of the following metals and state their uses : carbon steel, brass, bronze, permalloy, stainless steel, duralumin, monel metal, widia.

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CHAPTER – 8

ENGINEERING MATERIALS: WIRE ROPES & THEIR ATTACHMENT

A wire rope is an important item of engineering materials in mining and many other engineering industries. Wire ropes are made from steel wires of plain carbon steel having high tensile strength. The wire rope factory receives from rolling mills coils of steel rods of 6 to 13 mm dia. formed from steel in gots by repeated hot rolling. these rods may be made from carbon steel (0.5 to 0.8% carbon) made by open hearth acid or open hearth basic process and a typical analysis of the steel is as follows : (by weight, percentage).

Carbon 0.5, silicon 0.11, manganese 0.48, sulphur 0.033, phosphorus 0.014; iron rest. According to I.S. Specification No. 1835 of 1961 neither sulphur nor phosphorus content in the steel for wire rope should exceed 0.050 percent.

The ultimate tensile stength or breaking strength of the steel rods is about 65 kgf/mm² (638 MN/m²). These rods undergo the process of patenting (or normalising), pickling and coating as well as wiredrawing. The ultimate tensile strength (breaking strength) of wires used for haulage/winding ropes is generally between 140 kgf/mm² and 170 kg/mm². (160 kgf/mm² is equivalent to 1570 MN/m²).

If the wire rope is to be used in a wet shaft the wires are galvanised, *i.e.*, coated with molten zinc. Ropes of stainless steel are not used as the material has low tensile strength.

The wire is subjected to the following tests carried out according to the standards prescribed by I.S. specifications.

1. Tensile test.
2. Torsion test.
3. Bending test.
4. Wrapping test.
5. Looping test.

Types and construction of wire ropes :

Some wire ropes are required to carry the burden or load but are more or less stationary, *e.g.* guy ropes, guide ropes in shafts bucket-supporting ropes in bicable aerials ropeways. Such ropes are classified as *standing ropes.* Other types of ropes, classified as *running ropes,* have to undergo frequent movement, running or coiling often with varying loads and they have, therefore, to be flexible. The ropes used for winding, haulage, coal-cutting machines, winches, excavators, cranes are running ropes and compared to them the wires in the standing ropes are usually more robust in section and fewer in number as flexibility is not a major consideration with the latter.

On the basis of construction wire ropes are classified as :

(a) stranded ropes, and

(b) non-stranded ropes.

A stranded rope is built up of strands and each strand consists of a number of concentrically twisted wires laid in the form of a helix round a central steel wire. A seve-wire strand consists of a single centre wire, called king wire, covered by 6 concentrically laid wires and is common in the ropes used for haulages. If the wires of such strand are covered with a second layer of 12 wires the resultant strand is a 19–wire strand which is normally used for windng ropes. In the flattened strand rope, described later, the king wire in the strand is of triangular cross-section.

Wire ropes are designated usually by stating the number of strands, followed by the number of wires in each strand. For instance 6×7wire rope means that it is made up of 6 strands, and each strand is made up of 7 wires. Such rope is the simplest construction and is used mainly for haulage purposes. For winding and hoisting 6×19 or 6×37 construction is preferred.

The typical description of a rope used in heavy earth moving machinery will be as under :

"ungalvanised round strand steel wire rope, basic grade, preformed, 45mm dia., 6×36 Seale Warrington construction wires laid as (14/7 + 7/7/1)", right hand Lang's lay, 157® MN/m^2, IWRC).

Here, as stated earlier, the first number indicates the number of strands and second number, the number of wires in each strand. The figure in the paranthesis, viz. 14/7 + 77/1 indicates the number of wires layerwise in each strand. Thus in this case 14 outer wires are laid around 7 + 7 = 14 wires, which in turn are laid around 7 wires and one central wires.

The traditional method of clarifying the arrangement of wires in strand is to use the term like *straight, Seale, Warrington, Filler, Seale Filler, etc.*

The flexibility of a strand depends upon,

(a) *type of core :* a stand with a flexible core is more flexible than one with steel wire at the centre ;

(b) *thickness of individual wires :* thinner the wires, more is the flexibility;

(c) *number of wires :* larger the number of wires, more is the flexibility.

In a wire rope of the stranded type the strands are laid concentrically round a core which may be of the following type :

1. Fibre Core.
2. Steel Strand Core abbreviated as W.M.C.
3. Independent Wire Rope Core abbreviated as I.W.R.C.

This itself is a small wire rope consisting of 7 strands, each with 7 wires *e.g.*, ropes used for coal cutting machines have an independent wire rope core.

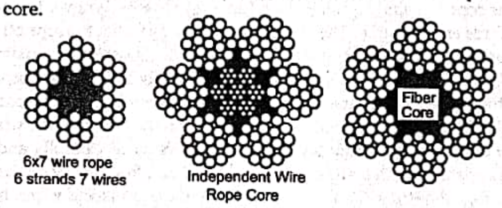

Fig. 8.1 A cross-section of round strand ropes.

Typical use : Left – for haulage; middle – for coal cutting machine; right – for winding.

Ropes for winding and haulage purposes have a central core of fibre when forms a soft bed for the strands and preserves the shape of rope under strain. During manufacture the fibre core is heavily lubricated. A steel wire stranded core is preferred in ropes operating in conditions of high temperature (*e.g. in steel melting shop of a steel plant*) or in ropes subjected to shock loads (*e.g. coal-cutting machines*).

Wire rope lays :

The term lay used in relation to a strand indicates the direction of laying of wires, in the strand. There are two types of lays, the right hand lay, and the left hand lay. In a right hand lay the wires sprial round the core in the same direction as the threads of a right hand screw. The opposite is known as the left hand lay. The length of lay is the distance measured along a straight line parallel to the strand in which the individual wire forms one complete spiral round the strand.

In the right hand lay the rope resembles a multi-start right hand screw thread. The right hand lay is a standared construction unless ordered otherwise by the user. The left hand lay construction is not common for haulage ropes or winding ropes used on drum winders but is sometimes adopted for ropes on multi-rope Koepe winders, where adjacent ropes are of opposite lays, *i.e.* one rope with right hand lay and the adjacent rope of left hand lay. This prevent untwisting of strands.

Lang's lay and ordinary lay :

A rope is of *ordinary lay* construction if the wires in the strand and the strands in the rope are laid in *opposite directions.* Ordinary lay is also known as *regular lay.*

A rope is of *Long's* lay construction if the wires in the strand are laid in the *same* direction as the strands are laid in the rope. Such construction causes the rope to spin. For this reason Lang' lay rope must never be used if there is a free end to rotate. The advantage of this lay is that the rope offers a better wearing surface than one of ordinary lay and it is also more resistant to bendine fatigue. Lang's lay ropes are favoured for windng and haulage purposes, not only because they present a much greater wearing surface than the ordiany lay but also because they are not liable to break on the crown of the strands unlike in ordinary lay. Lang's lay ropes have to be used carefully and are preferably operated with both ends of ropes anchored. Regular lay ropes have the advantage that they are non-rotating since the strands and wires, being laid in opposite directions, tend to balance each other's rotating tendency. They are ideal for use in places where they are freely suspended, such as in cranes.

Fig. 8.3 shows wire ropes with different construction of aly.

For the past few years, wire ropes of *equal lay* construction are being manufactured. In equal lay (or parallel lay) construction ropes, all layers of wires have the same pitch or length of lay. Each wire therefore lies either in a bed formed by the interstices or vally between the wires of an underlay or alternatively along the crown of an underlying wire. Such ropes can be made in Lang's lay on ordinary lay. An equal lay construction increases wear resistance. It also avoids deformation, internal wear and secondary bending which results from the point of contact between the wires in a conventionally laid strands. In most fields of application therefore equal lay ropes have given longer life than conventional ropes. The following are examples of equal lay ropes : (1) Warrington pattern (2) Seale pattern (3) Filler pattern (4) Warrington-Seale pattern.

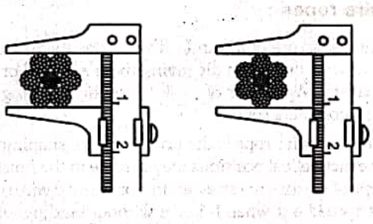

Fig. 8.2.

To measure diam. of a rope

Left : wrong method Right : correct method.

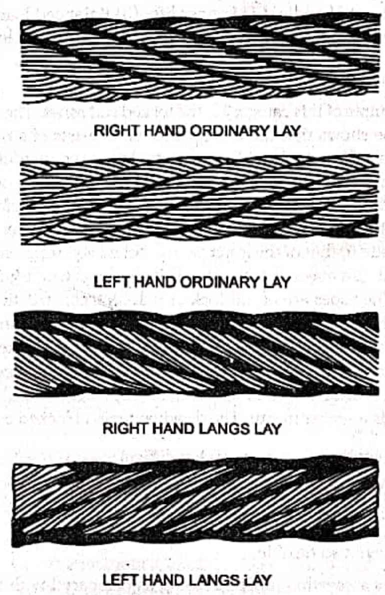

Fig. 8.3 Different lays of stranded ropes.

Preformed wire ropes :

In the manufacture of a standard wire rope the first operation is to draw a number of wire through a die giving them a helical or spiral pattern thus forming a strand. A number of strands are then laid together in spiral form to produce a complete rope.

Pre-forming a wire rope is the process of pre-shaping the wires and strands into the exact helical positions they assume in the finished rope, thus relieveing the rope of the internal stresses. In a preformed wire rope the strands on wires do not spread out when it is cut without binding with wire at the point of cutting. It is therefore easier to handle. Ropes used for haulage, winding, coal cutting machines, cranes, excavators, etc. are now-a-days of preformed construction and they are available with Lang's lay or Regular lay.

Use of preformed wire ropes is recommended for the following reasons : (1) Easy to handle, (2) Longer life, (3) Balanced load on strands, (4) Broken wires lie flat, (5) More easily spliced, (6) Less liable to kink.

Non-stranded ropes :

An example of this category is the locked coil ropes. The cross-section of lock coil rope shows that the central portion consists of strands of thick round wires ; only the outer layer (or two outer layers) consists of round wires placed between specially shaped wirs of I section, rail section or trapezoidal section so that the wires lock with one another and the rope surface is smooth and plane compared to stranded ropes. By laying up the outer wires in the direction opposite to that of the inner wires, locked coil ropes are made non-spinning and this is a major advantage in sinking shafts where guide ropes are not installed. The ropes are of full-lock or half-lock construction (Fig. 8.4). The locked coil ropes are heavier and stronger but less flexible than the stranded ropes of the same dia. For winding and hoisting purposes a locked coil rope is sometimes preferred because of its high capacity factor which permits a high factor of safety. A broken wire in a locked coil rope lies flat and it is possible to braze the ends together in situ. The disadvantages of locked coil ropes are :

1. its construction is somewhat difficult,
2. its interior cannot be lubricated from outside.
3. it cannot be spliced,
4. it is not so flexible,
5. it is somewhat difficult to cap as compared with the stranded ropes.

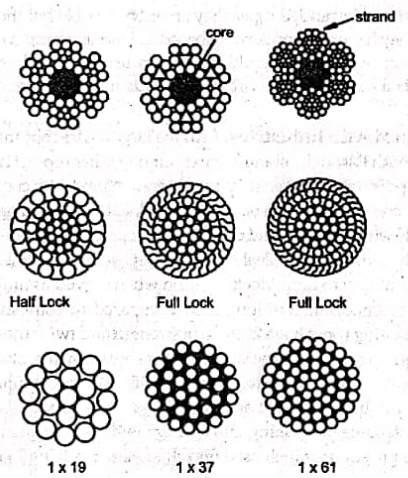

Fig. 8.4

Top row : Flattened strand ropes. Left- 6× 8 (7/Δ). 6 is the number of strands; 8 is the number of wires in one strand and consists of 7 wires round + 1 core wire of triangular section. Middle– 6 × 8 (7/Δ). Right 6× 25 (12/12Δ), denoting 6 strands each of 25 wires.

Middle row : Locked coil ropes.

Bottom row : Spiral strands. These are used as ropes for certain applications, mostly suspension bridges.

Locked coil ropes do not stretch as much as the stranded ropes and their smooth exterior causes less abrasion and wear of the surface in contact.

Locked coil ropes were not preferred for Koepe winders until a few years ago because of smooth surface and low coeff. of friction but there are no a number of installation in Britain and other European countries using locked coil ropes.

Flat ropes are not much used in this country, the exceptional use being on the reels of Koepe winders at Sudamdih Colliery in its early stage. The rope is made by stitching several strands side by side with M.S. wires.

Compared to round stranded ropes they are more flexible but the life is much shorter and they have been preferred abroad as balancing ropes on the Koepe winding system. A flat rope should not be confused with a flattened strand rope, which is a stranded rope with the strands having a core of triangular cross section.

Usha Martin Industries, Ltd. the largest wire rope manufacturing company is South East Asia, manufactures, among other ropes, HYFLEX ropes. The Hyflex rope is a rotation resistant rope manufactured with compact strands in multi layer construction with outside stands laid in direction opposite to the inside strands with a wire strand core. Hyflex rope offers superior resistance to rotation which makes it an ideal rope for a single-part hoist line. In addition, it also has the ability to limit block rotation when reeved as multi-part hoist line. This is of importance in long lifts. Compared to conventional ropes, Hyflex non-rotating ropes have 20% extra strength and twice the resistance to bending fatigue. They have large strand surface area which provides excellent resistance to abrasion, thus extending rope life. The smoothness of Hyflex rope strands minimises sheave and drum wear. Higher steel content gives maximum resistance to crushing due to drum winding and permits of multi-reeving. These ropes are much used for tall towers involving long lifts.

Selection of wire ropes :

A wire rope is to be selected on the following considerations :

1. **Watery place and corrosive atmosphere :** A galvanised rope should be used in such places to prevent rusting and effect of corrosive fumes.
2. **High temperature :** Rope with fibre core should be avoided in such places *e.g.* in foundries, steel melting shops.
3. **Stationary or running coiling rope :** Stationary ropes can be of large diameter rods or strands *e.g.* guide ropes in a shaft. Running or coiling ropes require flexibility and smaller the drum/pulley, more is the flexibility required, *e.g.* rope of a coal cutting machine which has to coil on a smal drum should consist of a large number of thin wires and the lay of rope should be "regular" as it give more flexibility.
4. **Splining or rotating quality :** In a crane rope, one end is free to rotate and a non-spinning rope or one with ordinary lay should be used. In a sinking shaft, the sinking bucket is not travelling on guides; therefore, a non-spinning rope of locked coil construction or a rope with ordinary lay should be used.

5. **Shock loads :** Where a rope has to withstand shock loads, the core should be of steel *e.g.* coal cutting machine rope.
6. **Resistance to wear :** Ropes for haulages and winders have to be flexible and resistance to abrasive wear. Such ropes should be of Lang's lay construction as they offer more wearing surface.

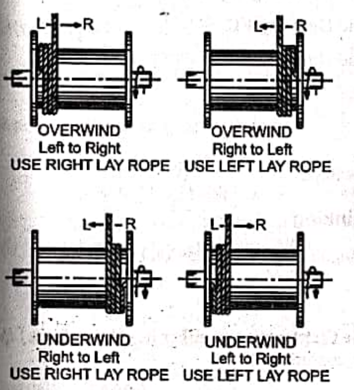

Fig. 8.5 Winding on drum

7. **Tensile strength and factor of safety :** Ropes used for winding of men should have high tensile strength and high factor of safety than those used for winding of materials only. Rope of the Lang's lay construction stretches under load more than the rope of regular lay construction.
8. **Bending fatigue :** Repeated bending of a wire rope over sheaves or drums cases fatigue failure of the wires. The rope should be flexible which is possible in a rope having large number of smaller wires.
9. **Groove size :** The rope should not be loose or too tight in the groove of the pulley or drum.
10. **Crushing and distortion :** A flattened strand rope and locked coil rope is better able to withstand crushing than a round strand rope. The core should be of steel wire.

Once the construction, lay and other characteristics of the rope are decided upon, one has to decide its size after calculating the stresses that the rope may have to withstand.

Ropes used for different purposes :

Generally ropes of the following construction are used in mines for different jobs.

Winding ropes :

6 × 7 Lang Lay, F.C.

6 × 19 Seale Regular or Lung Lay, F.C.

6 × 21 Filler Wire Regular or Lang Lay, F.C.

6 × 25 Filler Wire Regular of Lang Lay F.C.

6 × 27 Flattened Strand Lang Lay, F.C.

6 × 30 Flattened Strand Lang Lay, F.C.

Locked coil hoist Rope.

Guide Ropes :

Half Locked Coil Guide Rope.

Winding ropes for shaft sinking :

19 × 7 Non-rotating Regular lay or Locked Coil Hoist Rope.

Haulage Ropes :

6 × 7 and 6 × 19 Seale Construction in either Regular or Lang Lay, F.C., depending upon operating conditions.

Coal Cutting machine ropes :

6 × 37 Regular Lay with IWRC or 6 × 31 regular lay, IWRC,

Dipper Shovel Ropes :

(a) Dipper Hoist Ropes :

For 32mm and smaller size, 6 × 25 Filler Lang Lay with IWRC.

For 35mm to 68mm size, 6 × 41 Seale Filler Lang Lay with IWRC.

(b) Crowd and Retract Ropes :

For 58mm and Smaller size, 6 × 41 Seale Filler Wire Lany Lay with IWRC.

(c) Boom Hoist Ropes :

For 30mm size, 6 × 25 Filler Wire Lang Lay with IWRC.

Dragline Hoist Ropes :

For 32mm to 58mm size, 6 × 25 Filler Wire Lang Lay with IWRC on 6 × 41 Seale Filler Wire Lang Lay with IWRC.

Dozers :

6 × 25 Filler Wire Regular Lay with IWRC (Blade hoist ropes)

Guy ropes :

Galvanised strand 1 × 7, 1 × 19, 1 × 37, etc., or 7 × 7 or 7 × 19 Galvanised Guy Rope.

Aerial ropeways :

(a) Bicable ropeway :

Track Cables : Lock Coil, Full Locked or Half Locked,

Traction Rope : 22mm and Larger, 6 × 19 Seale Lang Lay F.C, or 6 × 25 Filler wire Lang Lay with IWRC.

(b) Monocable :

6 × 7 Lang Lay, FC

6 × 21 Filler Wire Lang Lay FC.

Mobile Cranes :

(a) Main Hoist Rope :

6 × 25 Filler Wire Regular Lay with Fibre Core. (Use IWRC Ropes to take care of crushing of the rope on the drum).

(b) Boom Hoist Rope :

6 × 25 Filler Wire Regular Lay with IWRC.

Mass and strength of wire ropes :

The space factor of a rope is the ratio of aggregate cross sectional area of the wires to the area of a circle drawn around the rope. For a six standed rope the space factor is between 50 & 60% depending on the design but it is about 75% for a locked coil rope.

The mass of a rope depends upon the quantity of steel in it, *i.e.* the space, factor, and the design of the rope.

mass of rope = kd^2

where k is a constant depending on rope design, d is diam. of rope in cm (and mass is in kg/m length).

Strength (*i.e.* breaking strength) = s d^2

where s is a constant depending on rope design and quality of steel and d is the rope diameter in cm (and breaking strength is in kN).

If the steel has a breaking strength of 1570 MN/m² (nearly 160 kg/mm² which is the usual valve), the values of k and s for purposes of the above formulae are generally as follows, if d is in cm, m (mass) is in kg/m and the breaking strength is in kN.

Type of rope	k	s
Round strand with fibre core	0.36	52
Round strand with wire core	0.40	56
Flattened strand with fibre core	0.41	55
Flattened strand with wire core	0.45	58
Locked coil	0.56	85

The size of a wire rope is usually quoted in mm, but the centimeter (cm) leads to more convenient constants. If it is considered necessary to work with the rope diameter in mm all the time, the two formulae can be written as :

mass = k $(d/10)^2$ in kg/m, d being in mm

and breaking strength = s $(d/10)^2$ in kN, d being in mm.

Example :

A wire rope, round stranded with fibre core, has a diameter of 2.54 cm. If the steel has a tensile strength of 160 kg/mm², find out the mass of the rope and the breaking strength in SI units.

Ans. :

Using the formula

mass of rope in kg/m = kd^2 (d in cm)

and using the value of k as 0.36 from the tables, we get mass of rope (kg/m) = 0.36×2.54^2

= 2.32

For 100m long rope mass is 232 kg.

Breaking strength is given by the formula

B.S. (in kN) = sd^2. (d in cm)

From the tables, value of s is 52.

B.W. = 52×2.54^2

= 335 kN

Socketing or Capping a rope and :

The end of a rope where the load is to be attached should be a good portion of the rope, free from worn, rusted, bent or broken wires and free from effects of bending and corrosion. The simplest and easiest way to make the rope and suitable for attachment of load is to use a grooved thimble and bend back the rope end on it and part of the rope before finally tightening 4–6 rope clips of intervals on it as shown in Fig. 8.6. The method needs little skill. Such attachment is permissible for haulage ropes and skip ropes hoisting on inclined planes but not permitted for winding ropes. Rope length under clips is neary 30 times rope dia. there are different ways of attaching capels or sockets on winding ropes, haulage ropes, coal cutting machine ropes, crane ropes, etc.

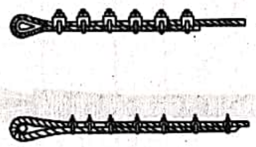

Fig. 8.6

Top : *Rope clips equally spaced for small diam rope (upto 25mm).*

Bootom : *Clips more closely spaced near the himble for large diam. rope (25mm–40mm)*

There are different ways of attaching capels or sockets on winding ropes, haulage ropes, coal cutting machine ropes, crane ropes, etc.

(a) Split capel with rivets : This is normally used on haulage ropes in mines but not permitted on winding ropes. Conical portion of capel fits the rope. (Fig. 8.7). Near the end of the rope mark two-points; one point a one cone length away and another point b, two cone length away from the end. Once rope between points *a* and *b*, wrap a number of turns of binding wire tightly to form a layer. Near *a* gives several wrappings of the wire to make it thick and slightly conical. Open out wires between rope end and point *a* and clean them with petrol, kerosene oil or diesel oil to remove grease, oil or rust.

After fanning out the wires cut $\frac{1}{3}$ rd of them to $\frac{1}{3}$ rd length and another $\frac{1}{3}$ rd to $\frac{2}{3}$ rd length. Turn back all the wires on the rope portion *a b* to give a cone and tie them on that rope portion with binding wire. Cut the exposed fibre core. Lay a thin layer of molten white metal on the cone with the help of a blow lamp (blow lamp is not allowed in an underground coal mine where the molten metal should be carried in a special container to prevent solidification before use). Hammer a thin wooden wedge into the cone at the end *a*. Push a split capel with its mouth slightly widened onto the cone and hammer the widened arms in position to grip the coned portion of the rope. Reivets are then hammered into the capel and through the rope at 3–4 points nearly 200m apart.

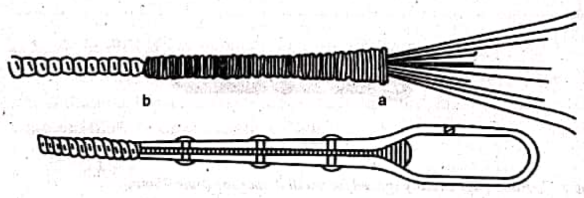

Fig. 8.7

Top : Cutting wires to make a cone
Bootom : Split capel withbent back wires.

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TABLE I – Mass and breaking strength of wire rope for Mining. Ropes manufactured Usha Martin Industries, Ltd. Only a few common sizes are included.

Rope construction	Rope nominal dia. (mm)	Approx mass kg/100m	Minimum breaking load for tensile designation 160 kgf/mm² kgf	160 kgf/mm² kN	180 kgf/mm² kgf	180 kgf/mm² kN
Round Strand 6 × 7 (6/1) F.C.	19 25 31	125 . 0 217 . 0 334 . 0	19 200 33 200 51 100	188 326 561	21 600 37 400 57 500	212 367 564
Round strand Equal Lay 6 X 19 (9/9/1) F.C.	19 25 32	131 . 0 226 . 0 370 . 0	19 200 33 100 54 200	187 325 532	21 500 37 200 61 000	211 265 598
Round Strand EqualLay 6 X 19 and 6 X 21 F. C.	32	378 . 0	55 300	542	62 200	610
Triangular Strand 6 X 8 (7/v) and 6 X 9 (8/v) F.C.	19 25 31	147 . 0 255 . 0 392 . 0	20 900 36 100 55 600	205 354 545	23 200 40 100 61 700	228 393 605
Triangular Strand 6 X 22. 6 X 23. 6 X 28 6 X 25. 6 X 31. F.C.	19 25 32	149 . 0 258 . 0 423 . 0	20 300 35 000 57 500	199 343 564	22 600 39 100 64 000	222 384 628
Round strand equal lay 6 X 26 to 6 X 41, IWRC	20 26	167 . 0 283 . 0	22 800 38 600	223 378	25 700 43 300	252 424
Locked coil winding rope	19 25 32	203 . 6 352 . 5 577 . 5	37 500 64 909 106 400			

(b) Coned-Socket type capel : The coned socket type caped is probably the most compact type of rope copping. This can be fitted on the rope used for practically every purpose, including winding. Near the rope and where the coned socket is to be used on the rope, wrap a few turns of binding wire tightly at a pont equal to $1\frac{1}{4}$ times the length of conical portion of the canel. Thread the rope and through the capel. Open out the end wires beyond the binding wire lashing, clean them with a suitable solvent like kerosene or diesel oil and cut the exposed fibre core. Reassemble the wires so that the rope end resembles a brush with the ends of the wires even. Pull the rope through the capel so that the branch remains inside its conical portion. Clamp the capel, complete with the rope in place, in a vertical position with the large end of capel pointing up, in readiness to receive molten white metal. (Fig. 8.9).

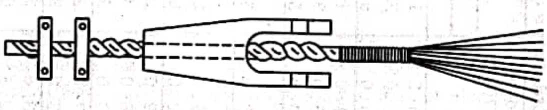


Fig. 8.8

Top : preparing "brush" for coned socket

Left : Capel fixed vertical for pouring white metal

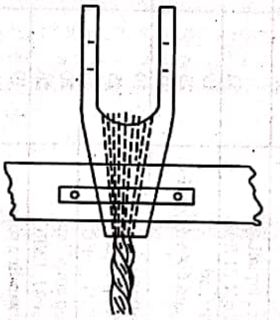


Fig. 8.9

Left : Open coned socket;

Right : Closed coned socket.

Seal the junction of rope and capel with absestos yarn and moist clay to prevent escape of molten metal. Heat the capel gradually and evenly all round the outside circumference by a blow lamp (not permitted in underground coal mine) avoiding heating of the wire rope and undue heating of the capel. Such heating is essential for free flow of molten metal. Immediately before

pouring molten metal, dust powdered resin over the rope wires inside the capel. Pour molten white metal (temperature not exceeding 365°C) to fill up the conical hole of the capel. Allow the metal to cool gradually till the capel cools to atmospheric temperature.

As an alternative to white metal the polyester resins offer considerable promise for rope socketing in a wide variety of circumstances. A particular advantage of such resins is that no heating equipment like a blow lamp or a heater is required in the capping process, and curing is by exothermic reaction. Socketing with synthetic resins can thus be undertaken in locations where the use of heat is prohibited (as in underground coal mines), or on difficult or exposed sites. A chemical accelerator, catalyst and aluminium powder or silica sand filler are mixed with the liquid resin and poured in to the rope brush within the capel. The resin can be made to cure within a short period, the actual time depending on the amount of catalyst and accelerator and the temperature.

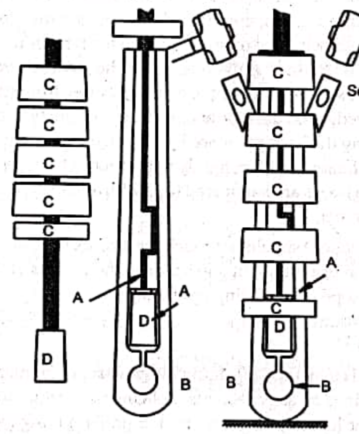


Fig. 8.10 Attaching reliance capel

(c) Interlocking wedge type capel (reliance capel) : In this capel there are 2 tapered iron wedges which grip the rope. The end of the rope is embedded in a block of white metal and the wedges are placed in a U-shaped steel strap on which 4–5 wrought iron hoops or clamps are fitted by hammering. The wedges have a machined groove curved to fit the rope surface and a taper of approximately 1 in 20 upon which the U-shaped strap is held. The jaws of the capel are about 24 times rope diameter in length.

The procedure for fitting such a capel on the rope is as follows :

Keep at hand 3–6 sets of rope clamps (each in two halves) to prevent slipping of the wires while mounting. Six clamps should be used for locked coil ropes. Near the end of the rope, at a distance equal to the tapered length of the capel, lash a layers of binding wire on the rope. Similarly, lash a layer of binding wire on the rope at about 200mm from the end. Thread the wrought iron hoops, C, on the rope in the correct order which they will occupy on the capel. Then thread the metallic cone Don the rope. Fix a set of clamps to the rope portion which will, at the end of the operation, remain inside the wedges. Open up the end wires for a length equal to the length of cone D, clean them, cut out the exposed fibre core, if any, and slide the metallic cone on the cleaned wires made as brush. Warm up the cone with a blow lamp and pour white metal into the core as described in (b) for coned socket type capel so that when the metal solidifies, the wires, the metal, and the cone become one solid mass. After removing the rope clamps and the remaining binding wires clean the rope for a length slightly in excess of the tapered – wedges length. Make sure that the wedge grooves are completely free from lubricant. Place the interlocking wedges on the rope and the whole assembly into the U-shaped strap after greasing the back of the wedges. Slide the wrought iron hoops on to the U-strap and hammer them lightly in position. The capel is placed on a hard floor and the hoops are hammered hard with a shaped set in conjunction with a sledge hammer.

The end cone D should not touch the interlocking wedges and when the capel will be in use the slight gap between the wedges and the cone will indicate that the rope is not sliding up the capel.

Rope Splicing :

Splicing is a method of joining two wire ropes permanently without using special fittings or attachements. Splicing of winding ropes, by which men are raised or lowered is not permitted under mining regulations, but haulage, power transmission, and aerial ropes can be used after splicing and the splice can be made nearly as strong as the original rope. The form of splice used on haulage ropes is that known as "long splice", "Short splice" is used on ropes employed on ships. Strength of the spliced rope depends on the length of the splice and on the friction between the interlocked strands. The length of splice will depend upon the diameter of the rope, the lay and the work it will have to do. It is nearly 6–9 m for 13mm diameter rope and 10–15m for 25mm diameter rope.

Method of splicing round ropes (Lang's or ordinary lay) :

1. Decide the length of splice

2. Bring the two ends of the rope to be spliced side by side for the length of splice. (In the Fig. it is 6m). On each rope, from the end, beyond the total length of the splice, tie twine on the rope.

3. Open out strands of the two ropes upto the twine binding and cut fibre core.

4. Cut out alternate strands of each rope about 30 cm from the twine binding.

5. Bring the two ropes face to face so that the cut-out cores meet. Temporarily lash the separated strands of left hand rope to the strands of right hand rope. The strands of RH rope are now ready for "running in" into LH rope.

6. Gradually unwind or unlay strand A of LH Rope which will be a short strand, and in its bed insert the meshing strand No. 1 from RH rope which will be a long strand. This must be laid firmly and lightly into the bed until all but 0.3m of strand 1 is laid in (as shown in Fig. 8.11, iii.)

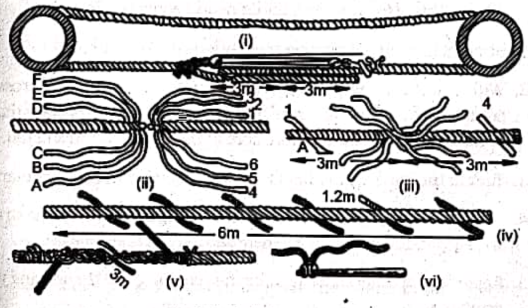


Fig. 8.11 Splicing a wire rope.

7. Cut off strand A to keep an equal length *i.e.* 0.3m and tie the strands temporarily in place.

8. In a similar manner lay strand 3 of RH rope into the groove formed by unlaying strand C of LH rope, but stoppig the pair about 1/5 of the length of splice short of the precediing pair.

9. Repeat the process for the pair, E strand of LH rope and corresponding meshing strand 5 of RH rope.

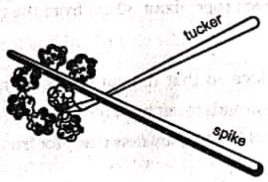


Fig. 8.12

All the long strands of RH rope are now laid into LH rope, leaving the shortened strands only of RH rope for treatment. The above operations of removing the short strands and replacing the short strands and replacing them by long strands are then repeated with the long strands of LH rope going into the beds of corresponding short strands of RH rope. The length of spliced rope will now have an appearance as in Fig. 8.11, iv, with 6 pairs of tails emerging at 6 crossing along the spliced portion.

10. Bend the splice back and forth until all strands rest firmly in their places. This also puts them under nearly equal tension.

11. Straighten each tail by removing any sprial formation.

12. With a vice and clamps untwist and open the rope at the end crossing, cut the fibre core at the centre, pull it out and tuck in its place the tail of the strand. Out off the fibre core at the end of the strand tail.

13. Tuck in the other strand tail of the same crossing in a similar manner.

14. Shift the vice and clamps to the next crossing and hammer the strands with a wooden mallet to fix them securely in their place.

15. Repeat the operations at the other five crossing and the splicing job is complete.

Long splicing of flattened strand ropes : Up to the stage of tucking, the procedure is the same as for round strand ropes. While the running in of the strands should present no difficulties in ensuring that the flats of the triangular strand from the circumference of the rope, it may be found that the corners tend to occupy this position. In such cases the strand should be clamped and then turned to the desired position.

At the tucking in stage there are slight modifications : As the core in a flattened strand rope is smaller than in a round strand rope of equal size, the tails should not be bound over their full 1.2m length which is to be run in. A short whipping is made on the end of the strands, merely to retain the wires in their stranded positions.

Care of wire ropes during storage and use :

The following points should be kept in mind during the storage and use of wire rope (A, B, C, of wire ropes) :

1. Avoid use of rope with fibre core, when the rope is subject to heat, fumes and extreme pressure.
2. Buy right construction of rope suitable for the job.
3. Corrosion can be delayed by using galvanised rope.
4. Don't load the rope beyond its safe working load.
5. Ensure that the rope is strongly seized before it is cut.
6. Flexibility of rope should be suitable to the size of drums and pulleys, and diameter of rope to grooves.
7. Grease the rope and cover properly before storing in a dry ventilated shed.
8. Handle the rope carefully while transporting and uncoiling to avoid kinks.
9. Inspect the rope periodically and lubricate with acid-free lubricant.
10. Judge the safe life of the rope for the conditions under which it has to work and replace it in proper time.

Comparative chart for breaking loads :

Size of Rope (diam)		6/37 constn. RHO, IWRC, 180 kgf/mm²		18/7 constn. RHO Std core 180 kgf/mm²		18/7 Hyflex	
13 mm	Wt./100m.	64	kgf.	68	kgf.	75	kgf.
	Min. Br. Load	9.7	Tons.	10.0	Tons	12.5	Tons
20 mm	Wt/100 m	138	kgf.	153	kgf.	176	kgf.
	Min. Br. Load	21.3	Tons	23.0	Tons	28.0	Tons
26 mm	Wt/100 m	245	kgf.	272	kgf.	296	kgf.
	Min.Br. load	36.0	Tons	40.0	Tons	48.0	Tons
32mm	Wt/100 m	371	kgf.	411	kgf.	452	kgf
	Min.Br.Load	54.5	Tons	59.0	Tons	76.0	Tons

QUESTIONS

1. Write notes on : Lang's lay, Ordinary lay, Equal lay, preformed ropes.
2. Describe the procedure of splicing a wire rope of an endless haulage.
3. Describe how reliance capel is to be fitted on a winding rope.
4. State the type of rope to be used for :
   (a) Coal cutting machine.
   (b) Winding rope.
   (c) Guide rope.
   (d) Crane hoist rope.
   (e) Dipper shovel boom hiist rope.
   (f) Guy rope.

O O O

CHAPTER – 9

PRINCIPLES OF AIR COMPRESSION

Compressed air is used as power mainly to operate drills and other machinery in mines, both on the surface and underground, and in the underground coal mines where electricity is widely used as the main power, compressed air is preferred for drifting in stone due to its relatively superior performance as power of drills. For operating pusher rams during decking operations at the pit-top and pit bottom, compressed air is required in mechanised mines. In deep underground coal mines with heavy methane emission compressed air is the safest power.

Compressed air provides power somewhat similar to steam but unlike steam, it can be used cold.

Air pressure :

A barometer records absolute pressure in mm of mercury columns. The value of atmsopheric pressure varies from place to place and depends further on climatic conditions. It is maximum at the sea level and gradually decreases with altitude abvoe sea level. The standard atmospheric pressure, also called the mean or normal atmospheric pressure, is defined as that pressure which supports a column of mercury 760 mm high at sea level when the temperature of mercury is 0°C. This is stated as absolute pressure of 760 mm of mercury. In the metric units the value of standard atmsopheric pressure is 1 kgf/cm². The internationally accepted values of standard atmospheric pressure in SI units is 101 325 N/m² *i.e.* approximately 100,000 N/m² or 10^5 N/m². The unit bar (b) used by the meteorologists to express atmospheric pressure is equivalent to 10^5 N/m², and therefore very nearly equal to the atmospheric pressure. A 760mm column of mercury is equivalent to $\frac{760 \times 13.6}{1000}$ metres of water column *i.e.* 10.34 m of water column. (sp. gr. of mercury is 13.6). If the barometric pressure at a place is h mm of mercury, its conversion into S.I. units is simple :

h mm height of Hg = (101 325/760) × h N/m²

= 133.3 h N/m²

Atmospheric pressure in mines below sea level is higher than the atmospheric pressure near sea level and the rate of pressure rise is the same as the rate for pressure decrease for places above the sea level.

Laws governing compression and expansion of gases :

There are certain laws governing prefect gases. Atmospheric air is also governed by these laws to a great extent, and it is desirable to understand them briefly as they enter into almost every compressed air problem.

1. Boyle's law : It states : The volume of a given mass of gas varies inversely as the absolute pressure when the temperature remains constant.

i.e. $V \propto \frac{1}{P}$ and PV = constant --- (1)

so that $P_1 V_1 = P_2 V_2$

where 1 and 2 denote initial and final conditions (of pressure and volume).

2. Charle's Law : It states : The volume of a given mass of gas varies directly as the absolute temperature when the pressure is constant.

i.e. $V \propto T$ and $\frac{V}{T}$ = a constant --- (2)

The absolute temperature is always measured from the absolute zero of temperature and this is stated as – 273.13°C. For calculation purposes this is taken as – 273°C.

Absolute temperature = 273°C + C.

3. The characteristic equation of a perfect gas is derived froma combination of Boyle's and Charles' Laws. It is :

$$\frac{PV}{T} = R = \text{Constant, or } PV = RT \quad \text{--- (3)}$$

Where, P, V, and T denote absolute pressure, volume and absolute temperature respectively.

This equation is applicable to a fixed mass of gas. Let this mass be m. Equation (3) can then be written as

$$\frac{PV}{T} = m \times \text{constant}$$

$$\frac{PV}{T} = mR \quad \text{--- (4)}$$

where R is known as the characteristic gas constant.

This equation (4) is known as the characteristic equation of a perfect gas. For a perticular gas the value of R is constant.

In S.I. units if P is in units of N/m², V in units of m³, T in units of K and m in units of kg, then the units of R are

$\frac{Nm^2}{m^2 Kkg} = or \frac{J}{kg\ K}$ (Joules/kilogramme Kelvin). For air the value of R is 288 J/kg. K.

The above laws are known as gas laws and a perfect gas is one which obeys the gas laws exactly. In practice no such gas exists but for the temperature and pressures of gases of air with which we are concerned in compressed air calculations the concept, of an ideal or perfect gas in quite helpful. The results arrived by such calculations for air will not show a significant deparature from the results obtained in practice. The gas laws do not apply to vapours and steam which get liquified at comparatively low pressure and temperature.

Example :

A volume of 0.045 m³ of air at a pressure of 300 kN/m² and a temperature of 15°C is expanded to a volume of 0.075m³. If the temperature falls to 3°C calculate the final pressure of the gas.

Ans. :

Applying the formula $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$

$P_1 = 300 \times 10^3$ N/m²	P_2 = final pressure
$V_1 = 0.045$m³	$V_2 = 0.075$m³
$T_1 = 15 + 273 = 288$ K	$T_2 = 3 + 273 = 276$ K

Then $P_2 = \frac{P_1V_1}{T_1} \times \frac{T_2}{V_2}$

$$= 300 \times 10^3 \times \frac{0.045}{288} \times \frac{276}{0.075}$$

$= 172.5 \times 10^3$ N/m²

$\cong 172.5$ kN/m²

The final pressure of the gas is 172.5 kN/m².

Example :

An air receiver having a volume of 0.11m³ contains air at a temperature of 18°C and under a pressure of 1720 kN/m². After working of some of the air engines to which the receiver is connected the pressure of air in the receiver falls to 1200 kN/m² and the temperature to 10°C. Determine the volume of the air at S.T.P. that has been used from the receiver.

Ans. :

Suppose the volume of the air used at 1720 kN/m² and 18°O is V m³

∴ Volume remaining inthe receiver = (0.11 – V) m³.

This volume then fills the receiver and occupies the volume of 0.11m³ and consequently the pressure falls to 1200 kN/m² and the temperature to 10°C.

$P_1 = 1720 \times 10^3$ N/m² $\quad P_2 = 1200 \times 10^3$ N/m²

$V_1 = (0.11 - V)$m³ $\quad V_2 = 0.11$ m³

$T_1 = 18 + 273 = 291$ K $\quad T_2 = 10 + 273 = 283$ K

Using the equation $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$

and substituting the values

$$\frac{1720 \times 10^3 \times (0.11 - V)}{291} = \frac{1200 \times 10^3 \times 0.11}{283}$$

$$0.11 - V = \frac{1200 \times 0.11 \times 291}{1720 \times 283}$$

$= 0.079$

∴ $V = 0.031$ m³

This volume has to be expressed under the S.T.P. conditions *i.e.* 101.3 kN/m² and 0°C.

Using the gas equation $= \frac{PV}{T} = \frac{P_sV_s}{T_s}$

Where $P_s = 101.3 \times 10^3$ N/m²; $T_s = 273$ K

V_s = Volume at S.T.P.

We get $= \frac{1720 \times 10^3 \times 0.031}{291} = \frac{101.3 \times 10^3 \times V_s}{273}$

$$V_s = \frac{1720 \times 0.031 \times 273}{291 \times 101.3}$$

$= 0.494$ m³

The volume of air consumed is 0.494 m³ at S.T.P.

Specific heat of gas :

The specific heat of a gas is defined as the amount of heat requried to raise the temperature of unit mass of the gas through one degree. If the gas be heated in a closed vessel, so that its volume remains constant, the whole of the heat applied is spent on increasing the temperature (and incidently the pressure) of the gas. If the gas be heated in a cylinder having a frictionless and weightless piston under a constant external pressure, its volume changes but its pressure remains constant. In this case, the temperature of the gas is raised and also work is done in pushing out the piston against the external pressure. It follows that the specific heat at constant pressure, C_p, must always be greater than that at constant volume, C_v. The units for C_p and C_v are J/kg.K. The specific heat capacities vary with temperature and usually an average value is used for the temperature range involved. For air,

$C_p = 1005$ J/kg. K and $C_v = 712$ J/kg. K so that $\frac{C_p}{C_v} = 1.4$

The ratio is denoted by the Greek letter γ (gamma)

Atmospheric air is compressed in a machine called compressor and such compressed air is stored in a cylindrical tank, known as receiver, for use in machines referred to as in motors or air engines. A compressor is said to be single-stage when it compresses the air to its final pressure in one compressor cylinder and it is said to be multi-stage when it compresses the air in two or more compressor cylinders.

Isothermal and adiabatic compression and expansion.

When air (or any other gas) is compressed from an initial pressure P_1 to a higher pressure P_2 without change of temperature, the process is called Isothermal compression. Boyle's Law strictly applies to such conditions and PV = constant. For a gas to be compressed isothermally (a condition unattainable in practice) it is necessary that the heat produced during compression, by the conversion of mechanical energy into heat, must be removed by some cooling arrangement as fast as it is produced.

A gas is said to be compressed adiabatically when there is no transference of heat to or from the gas during its compression. In other words, the mechanical energy converted into heat during the compression is allowed to remain in the gas and goes to increase its temperature. Adiabatic compression follows the equation.

$$PV^{\gamma} = C \text{ where } \gamma = \frac{C_p}{C_v} = 1.4$$

Pure adiabatic compression is never attained in practice because perfect lagging of the cylinder to prevent loss of heat cannot be secured, nor can friction and shock losses be avoided; but compression usually more nearly approaches the adiabatic than the isothermal unless effective steps are taken to counteract the temperature rise by some cooling arrangement. Except in small capacity compressors, which are provided with fins on the compressor-cylinder for cooling by atmospheric air, cooling is effected in most instances by surrounding the cylinder or cylinders with a water-jacket through which cold water is constantly circulated, and in two-stage compressors, an intercooler is also provided between the two stages. The intercooler is much more effective than water jecketing. (see Fig. 9.4 for stationary reciprocating compressor). An intercooler is a veseel with a network of tubes through which water as atmospheric temperature is circulated and the compressed air passes through the space outside the tubes before entering the high pressure cylinder for further compression. Its function is like that of a radiator on automobiles. Baffle plates are fitted in an intercooler so that the air, during its passage, makes contact with the water tubes several times. Other fittings on an intercooler are a spring loaded safety valve and a drain cock. By using intercooler the air temperature is much reduced and brought down within a few degrees more than the atmospheric temperature.

Because of the reduced temperature, the volume of air is reduced and the high pressure cylinder is of smaller size. Some compressors are provided with aftercoolers, described later. A two-stage air cooled compressor has an intercooler with fins, served by a fan mounted on the compressor which provides

air circulation around the intercooler and the compressor cylinders, air-cooled compressors are preferred where there is difficulty of getting cool, clean water in sufficient quantity *e.g.* in some underground mines and in quarries or sites located at hill tops.

Compression of air in a two-stage compressor employing inter cooler generally follows the equation $PV^{1.3}$ = constant.

The advantages of compressing in more than one stage are (i) The work of compression is reduced by about 10 to 15% for final gauge pressures of 4 to 7 kg cm² because compression more nearly approaches the isothermal, as will be clear in the following pages. (ii) The final air temperature is reduced owing to the effect of the intercooler. Difficulties of lubrication and excessive wear and tear are thus avoided. (iii) The stresses set up in the machine are greatly reduced and a better balanced compressor may be built.

The high temperature of compressed air, ranging between 150°C and 200°C has an adverse effect on the working parts, especially the valves, and it may also result in the ignition of carbonaceaous deposits if inferior lubricants are used. The temperature is brought down by the use of an aftercooler, placed between the last cylinder in which compression is carried out and the receiver. An after cooler is similar to an intercooler in construction, operation and appearance. An aftercooler does not improve the efficiency of compression but an intercooler does. An aftercooler and an inter cooler, by cooling the air, cause much of its moisture to settle down.

Work done in compression :

In Fig. 9.1 the area of ABYD is a product of the volume and the pressure in the case of isothermal compression and therefore it represents the work done during such compression. Area AHYD represents the work done in adiabatic compression and it will be observed that the work done during isothermal compression is less than in adiabatic compression by the amount of the shaded area. This shaded area represents the increase in internal energy of the air corresponding to its temperature rise but the air generally cools again during transmission and before its use in airmotors, so that this extra energy is wasted by dissipation of heat. It is therefore economical to compress isothermally rather than adiabatically and this is one reason for the water jacketing of air compressors. In practice the compression is neither isothermal nor adiabatic but polytropic and the index of compression is usually 1.3 so that the compression curve lies between the isothermal and the adiabatic curves.

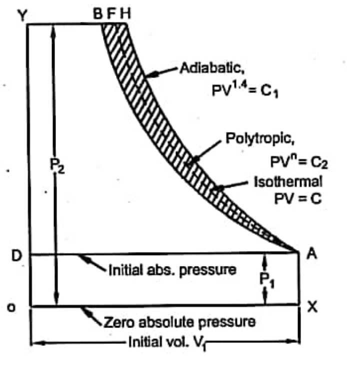


Fig. 9.1 Single stage compression

Isothermal compression :

To find the work done during compression let us consider the instant shown in Fig. 9.2 (a) when the absolute pressure of the air is P. If the strip height is P and the width dV, the area of the strip is PdV.

The total area ABNX represents the work done in part travel of one stroke of the piston in compressing a given mass of air from volume V_1 to volume V_2. The total area, ABNX (Fig. 9.2, a) for isothermal compression (which follows the law

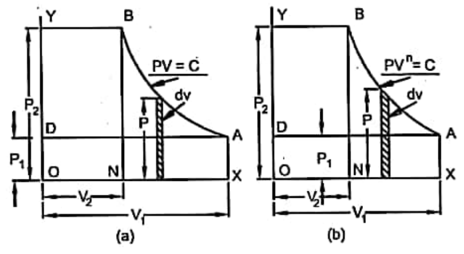


Fig. 9.2

$PV = P_1V_1 = P_2V_2 = C)$

$$= \int_{V_2}^{V_1} PdV = C \int_{V_2}^{V_1} \frac{dV}{V} = C \log_e \frac{V_1}{V_2}$$

The work done

$$W = P_1V_1 \log_e \frac{V_1}{V_2} = P_1V_1 \log_e \frac{P_2}{P_1}$$

If P is in N/m^2 and V in m^3

$$\text{then } W = P_1V_1 \log_e \frac{P_2}{P_1} \text{ Nm} \qquad \text{--- (5)}$$

The work done by the compressor piston per complete storke includes, besides this work of compression, the work done in expelling the air from the cylinder. The area ABNX is a measure of the theoretical work required to compress air from the initial pressure P_1 to the final pressure P_2. The area of the rectangle ONBY (P_2V_2) is a measure of the displacement work required to force the compressed air into a receiver at pressure P_2. The area of rectangle OXAD (P_1V_1) is a measure of the work done by the atmospheric pressure behind the advancing piston. It follows that the net work required of the primer mover to compress and deliver the air is equal to the area DABY. In other words.

Net work done = DABY =

work of compression atmosphere + displacement work – work done by atmosphere

= NXAB + ONBY – OXAD

= NXAB + $P_2V_2 - P_1V_1$

With the isothermal compression $P_1V_1 = P_2V_2$ and therefore the nett work required for isothermal compression and delivery of air (area = DABY) is the same as the theoretical work of compresin along (= area NXAB).

Adiabatic Compression :

In the case of adiabatic compression (Fig. 9.2) (b) work done at any instant, when the pressure of a given mass of air is P_1, is PdV for the strip dV. The total work done in compression from V_1 to V_2 is again the sum of the elementary strip PdV represented by the area ABNX.

Work done during adiabatic compression $\int_{V_2}^{V_1} PdV$

Since however $PV^n = C$ and $\therefore\ P = \frac{C}{V^n}$ we may substitute as follows :

$$\int_{V_2}^{V_1} PdV = C\int_{V_2}^{V_1} \frac{dV}{V^n} = C\int_{V_2}^{V_1} V^{-n}\,dV$$

$$= \frac{C}{1-n}\left(V_1^{(1-n)} - V_2^{(1-n)}\right)$$

$$= \frac{1}{1-n}\left(P_1V_1^n\ V_1^{(1-n)} - P_2V_2^n\ V_2^{(1-n)}\right)$$

$$= \frac{P_1V_1 - P_2V_2}{1-n}$$

If the pressure are given in N/m² and volume in m³, the work done is

$$W = \frac{1}{1-n}(P_1V_1 - P_2V_2)$$

$$= \frac{1}{n-1}(P_2V_2 - P_1V_1)\text{Nm} \qquad \text{--- (6)}$$

Note that this is the work done during compression only. It is the area ABNX in Fig. 9.2 (b). The work done per stroke, which has also to include the work done to push the compressed air into the receiver, is

= ABNX + ONBY – OXAD

$$= \left(\frac{P_2V_2 - P_1V_1}{N-1} + P_2V_2 - P_1V_1\right)$$

$$= \frac{n}{n-1}(P_2V_2 - P_1V_1) \qquad \text{--- (7)}$$

It can be shown by working out an example that adiabatic compression requires more work than isothermal compression. The extra energy received by the air during adiabatic compression is almost invariably lost in the form of heat which is dissipated to the atmosphere and by the time the compressed air is used in the air motors, the former is almost at the atmospheric temperature.

Example :

Find the work done in a compressor in compressing air (a) isothermally and (b) adiabatically with the following data.

Intial abs. pressure = 10×10^4 N/m² = 10^5 N/m²
Final abs. pressure = 50×10^4 N/m² = 5×10^5 N/m²
Initial volume = 6 m³
Index of adiabatic compression, n = 1.4

Ans. :

Isothermal compression

Work done = $P_1V_1 \log_e \frac{P_2}{P_1}$

$= 10^5 \times 6 \log_e 5$ Nm
$= 6 \times 10^5 \times 1.6$ Nm
$= 9.6462 \times 10^5$ Nm
$= 964620$ Nm

Adiabatic Compression

$$P_1V_1^n = P_2V_2^n$$

or $\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n$

or $\text{Log}_{10} 5 = 1.4 \log_{10} \frac{V_1}{V_2}$

or $\text{Log}_{10} \frac{V_1}{V_2} = 0.4993 \quad \therefore \quad \frac{V_1}{V_2} = 3.157$

$\therefore\ V_2 = \frac{6}{3.157} = 1.9\text{m}^3$

$$\text{Work done} = \frac{n}{n-1}(P_2V_2 - P_1V_1)$$

$$= \frac{1.4}{0.4}(5 \times 1.9 - 1 \times 6) \times 10^5 \text{ Nm}$$

$$= 12.25 \times 10^5 \text{ Nm}$$

$$= 1225 \times 10^3 \text{ Nm.}$$

$$\frac{\text{Work done (adiabatic)}}{\text{Work done (isothermal)}} = \frac{1225000}{964620} = 1.27 \text{ or } 27\% \text{ more}$$

If a gas is to be compressed from a lower pressure to a higher pressure then work will be required to carry out the compression. When the gas is at the new high pressure, it possesses the potential to expand and do work in expanding. The equations (5) and (6) will apply equally well to compression or expansion. During compression the work is done on the gas; during expansion work is done by the gas. If work done during compression is considered positive, the work done by the gas during expansion is negative.

Example :

A gas at a pressure of 2070 kN/m² has a volume of 0.014 m³. It expands to a low pressure of 207 kN/m² according to the law $PV^{1.35} = C$. Find out the work done by the gas during expansion.

Ans. :

$P_1 = 2070$ kN/m² $\quad P_2 = 207$ kN/m²

$V_1 = 0.014$ m³ $\quad V_2 = ?$

$n = 1.35$

Now $P_1V_1^n = P_2V_2^n$

$$\therefore \quad V_2 = \left(\frac{P_1}{P_2}\right)^{\frac{1}{1.35}} \times V_1$$

$$V_2 = 0.014\left(\frac{2070}{207}\right)^{\frac{1}{1.35}}$$

$$= 0.014 \times \sqrt[1.35]{10}$$

$$= 0.014 \times 5.5 = 0.077 \text{ m}^3.$$

Work done during compression only is given by equation. (6). If it is considered as positive work, the work done by the gas during expansion is negative and the equation. (6) therefore, the expansion becomes

$$W = \frac{1}{n-1}(P_1V_1 - P_2V_2)$$

$$\frac{(2070 \times 10^3 \times 0.014 - 207 \times 10^3 \times 0.077)}{1.35 - 1}$$

$$= \frac{10^3}{0.35}(29 - 15.95) = \frac{10^3}{0.35}(13.05)$$

$$= 37.3 \times 10^3 \text{ Nm}$$

$$= 37.3 \text{ kJ.}$$

Two Stage Compression :

The saving effected by two-stage compression is shown in Fig. 9.3 where the load dotted curve AB represents an isothermal curve and AB, and adiabatic curve, corresponding to an initial pressure P_1 and a final pressure P_2 assuming single stage compression. The curve AL is a polytropic curve which is assumed to obey the law $PV^{1.3}$ = a constant. Then, for single stage compression, the work required for compression and delivery is represented by the area ALYD.

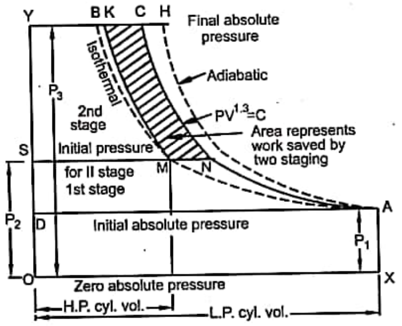


Fig. 9.3 Two-Stage Compression.

Assume now that two stages are used and that, in the first stage the pressure is raised from P_1 to P_2 polytropically according to the curve AN, the volume of the same time decreasing from OX to SN. The compressed air is then passed through a intercooler where its temperature is reduced to the initial free air temperature, the pressure remaining approximately constant. The volume is thereby reduced to SM where M is a point on the isothermal curve. In the second stage, compression starts at M on this curve and the pressure is raised from P_2 to P_3 according to polytropic curve MK. It is evident that the work required for two-stage compression is represented by the area ANMKYD and that the amount of work saved is that equivalent to the shaded area MNLK. To obtain the highest effciency, the cylinders should be of such a size that equal work is done in each and the absolute pressure P_2 at the end of the first stage should be such that $P_2 = \sqrt{P_1 P_3}$ where P_1 = intake absolute pressure and P_3 = delivery absolute pressure. Two-stage compression is used for gauge pressures between 4.5 kgf/cm² and 7 kgf/cm², this being the range of pressure commonly adopted in mining. The percentage saving of power over single-stage compression (within the range of pressure given) is of the order of 10 to 15%. Three stage compression is used for higher pressures upto about 50 kgf/cm² and four or even five-stage compression for still higher pressures.

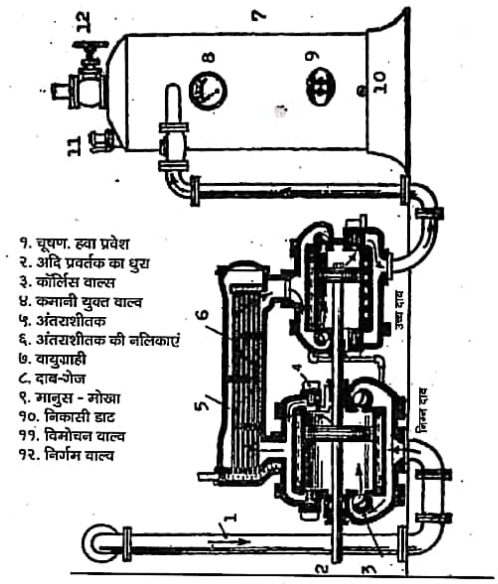


Fig. 9.4

Horizontal 2-stage stationary compressor in section. 1-Atmospheric air suction. 2. Shaft of prime mover. 3. Corliss valve. 4. Valve, spring loaded. 5. Intercooler. 6. Intercooler water tubes. 7. Air receiver. 8. Pressure gauge. 9. Manhole. 10. Drain cock, 11. Relief valve, 12. Main of delivery valve.

Capacity of a compressor :

The capacity of a compressor is always expressed in cubic feet or cubic metres of free air per minute and this refers to the actual volume of air compressed and delivered expressed, in terms of free air at the inlet temperature and pressure.

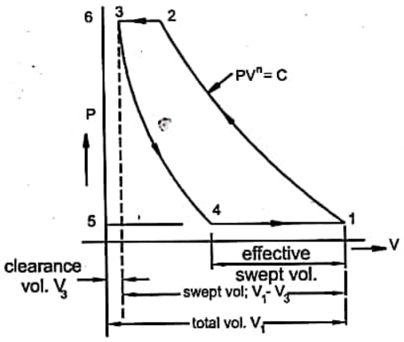


Fig. 9.5

Effect of clearance volume :

The clearance volume in a reciprocating compressor is the volume between the cylinder cover and piston, and the valve passages when the piston has reached the end of its compression stroke. In Fig. 9.5 the diagram enclosed by 1,2,3,4 is a theoretical diagram of a single stage compressor having polytropic compression according to the law $PV^n = C$. At 1 total volume (or initial volume) in the cylinder is V_1 before the piston starts on its compression stroke. Points 2 represents the pressure to which the air is compressed. At 2, delivery valve theoretically opens and for the piston-travel from point 2 to point 3, the compressed air is delivered from the cylinder. Point 3 represents the end of piston-travel and delivery of compressed air cases at 3. The volume between the cylinder over and piston and also in the passages to the valve (clearance volume, V_3) is filled with compressed air. As the piston starts on its return stroke the residual compressed air (V_3) expands and when its pressure drops to the intake pressure at 4, inlet valve begins to open and thereby permits the intake of fresh atmospheric air. During the travel of piston from point 4 to point 1, the atmospheric air is admitted into the cylinder and the volume of atmospheric air so admitted (from point 4 to point) is called the effective swept volume.

Volumetric efficiency of a compressor is defined as

$$\text{Vol. Efficiency} = \frac{\text{Quantity of air delivered in a given time}}{\text{piston displacement in the same period}}$$

A little consideration will show that

$$\text{Volumetric Efficiency} = \frac{\text{Effective swept volume in a given time}}{\text{Swept volume in the same period}}$$

$$= \frac{V_1 - V_4}{V_1 - V_3}$$

The figure is usually in the range of 60% to 85%.

QUESTIONS

1. Explain the difference between isothermal compression and adiabatic compression. Show by calculations how, and by what extent, adiabatic compression requires more power than isothermal compression.
2. Write short notes on : gauge pressure, intercooler, aftercooler.
3. Calculate the mass in kg of 1500 litres of air at a gauge pressure of 1.9 MN/m² and a temperature of 27°C. Take the characteristics gas constant for air as 288 J/kg K.

   (Ans. 34.7 kg).

O O O

CHAPTER – 10

GENERATION, DISTRIBUTION & USE OF COMPRESSED AIR

In the earlier chapter dealing with principles of air compression theoretical aspects of air compression have been considered. Fig. 10.1 shows a typical indicator diagram of a reciprocating compressor, available in practice, and its comparison with the theoretical diagram should be noted. The pressure of air compressed in the cylinder has to be slightly more than the receiver pressure for the delivery valve to open as at B. At D the pressure of air in the cylinder must be lower than the atmospheric air so that pressure difference across the inlet valve forces it off its seat. The inertia of the valve also requires a slightly higher pressure in the cylinder at B and slightly reduced pressure in the cylinder at D.

A – Piston travel starts. Compression begins.
B – Delivery valve opens.
C – Delivery of compressed air ends.
D – Suction valve opens; atmos pheric air suction starts.

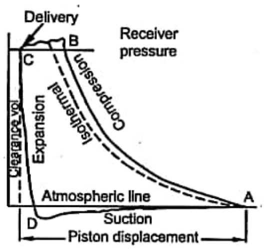


Fig. 10.1 Typical indicator diagram of a reciprocating compressor.

Types of Compressors :

Based on that types of construction, compressors are classified as follows.

i. Reciprocating compressors
ii. Rotary compressors :
   (a) Rotary vane type
   (b) Rotary screw type
   (c) Centrifugal or turbo compressor
   (d) Rotary twin hole compressor.

Reciprocating Compressor :

The reciprocating compressor is by far the most commonly used compressor for mining operations. A reciprocating compressor consists of a piston working to and fro in a cylinder which is neither isothermal nor adiabatic, but somewhere between the two, and is then discharged at the required pressure into a receiver. During the return stroke, air at atmospheric pressure is drawn into the cylinder through the inlet valves in readiness for compression. In a single-cutting compressor, delivery takes place in one cycle during one stroke only; in a double-acting compressor, it takes place during both strokes of the piston.

The valves are usually opened automatically by or pressure-difference and returned to their seats by a spring. The most usual from is the plate or disc valve, in which the valve itself comprises a steel plate, and provided with segmental openings to permit air to pass. Two other somewhat similar plates are included in the valve assembly, one being a cushion plate and the other a retaining plate, the details of construction varying with different makers. Poppet valves are out of date. Suction valves are usually fitted with an unloading mechanism, described later.

Force feed lubrication is adopted or reciprocating compressors. All oils give off combustible gases when heated. The lowest temperature at which this begins is the flash point and the tmeperature at which the combustible gases ignite is the ignition temperature. Ordinary lubricating oils flash at about 120°C, a temperature below the usual working temperature of compressors and therefore special cylinder oils with high flash point should be used.

Starting and operation of a reciprocating compressor :

Before starting a reciprocating compressor, routine check sould be made of oil level in engine and compressor, fuel, water in the radiator and a check for loose nuts, bolts, etc. The receiver drain cock should be opened and the clutch disengaged.

After starting, the engine is run for several minutes to warm up, and then clutch is engaged. Soon after it turns the compressor without choking, the drain cock, should be closed. If the engine is not thoroughly warm or is in poor condition, it may not be able to build up full receiver pressure within a few minutes. If the threatens to stall, the clutch should be disengaged or the compressor unloaded by a hand control and further time allowed to warm up.

Carbon and explosions :

The high temperature of compressor during working causes some vaporisation of the lubricating oil. The non-volatile residue, in combination with any dirt in the air, is liabe to build up hard or gummy deposits that will interface with valve operation and sometimes will choke air passages.

If exhaust valve leaks some of the hot compressed air which has just been forced out of the cylinder will come back in the cylinder and be re-compressed resulting in very high pressure locally and excessive temperature which will boil off and vaporise the lubricant. Under these conditions enough oil may be present in the air to cause explosion, most often in the cylinder or the passages, but occasionally in the reciver.

Rotary compressors :

Rotary Vane Type Compressors :

In the rotary vane type compressor a slotted rotor or its equivalent is mounted eccentrically in a stationary casing. (Fig. 10.2) Sliding vanes are fitted into radial lengthwise slots in the rotor and centrifugal force keeps them in contact with the casing will whenever the rotor is turning rapidly.

Fig. 10.2 Principle of working of a rotary vane type compressor.

The vanes divide the space between the rotor and the casing into a series of compartments. The cylinder is of considerably larger diameter than the rotor

which is placed off centre, so that the air pockets formed by the vanes between the cylinder and the rotor gradually increase and then decrease in size during each revolution. Air enters the compartments when the space between the rotor and stator is increasing. As the rear of compartment gets smaller during rotation, the air is compressed and eventually forced out of a delivery passage just before the point of closest contact between rotor and casing. The air passes to the receiver through a plate type non-return valve. cooled oil is injected directly into the air during compression. This atomised oil mixes with the air and performs three functions.

1. It cools the air by absorbing the heat of compression.
2. It forms an oil film between the vanes and the casing and prevents slippage of air from one pocket to the next across the vane.
3. It lubricates the sliding vanes, bearings, gears and other parts. The oil which mixes with the air during compression, passes out through the discharge into a receiver cum oil separator. The bulk of it drops out when the air stream looses velocity. The balance oil then passes on to a secondary separator, where, by means of baffles and a special filter, it is removed from the air before the latter goes into the air hose.

The oil is cooled by passing it through a radiator type oil cooler located in front of the engine radiator. On the radiator type oil cooler air is passed by a fan. Oil is then pumped back to bearing and compressor (Fig. 10.3.). On the same shaft may be mounted a second set of vanes at one end along its length and the air compressed by the first set may be introduced into the second set for further compression. This is the usual arrangement on Gyro-Flo rotary compressors manufactured by Ingersoll Rand. In the Kirloskar rotary compressors, W.R. type, the stator body is double bored, the L.P. stage and the high pressure stage being arranged side by side in a common stator casing. The first stage compresses to approximately 2.5 kgf/cm^2 and the final stage to 7 kgf/cm^2.

The main features of such rotary vane type compressor are :

1. Lubricating oil absorbs the heat of compression, No intercooler or other air cooling device is required.
2. Flow of air is steady dispensing with air receiver and the air can be taken directly from the separator into the transmission hose pipes. Automatic controls can regulate the air output.

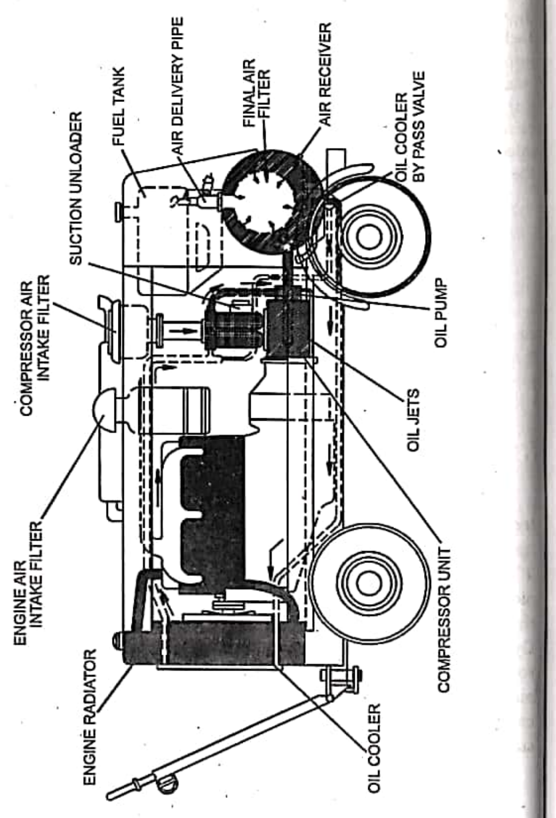


Fig. 10.3 Kirloskar compressor, rotary vane type (WR type), This type, portable with diesel engine, is available upto 17.5 m^3 per min. capacity.

(Courtesy of Kirloskar Penumatic Co. Ltd)

3. Before starting the engin the vanes are deep in the rotor slots. This provides automatic unloading at start. It is therefore not necessary to disconnect the compressor to start the engine.
4. During idling of the engine, the insufficient centrifugal force cannot keep the vanes in contact with the casing. This results in leakage around the vanes from one compartment to another causing automatic unloading and economy in fuel consumption.
5. The discharge temperature of a rotary compressor is about 40°C above ambient and therefore about 40°C cooler than a 2-stage reciprocating compressor. This results in efficient operation and prolonged life to the drills and other pneumatic tools.
6. As there is no reciprocating motion involved, the rotary compressor is practically vibrationless and the air is delivered smoothly and not pulsating unlike in a reciprocating compressor.
7. In this type of compressor, the condensation water should be drained out of the oil filter daily, the oil needs frequent replacement, and the air filter as well as the oil filter have to be changed often. The set of vanes on he rotor lasts about 2,000 hours. These vanes are non-metallic and sometimes of plastic or nylon.
8. A spring-loaded by-pass valve is provided in the oil pump assembly to protect the machine against exessive oil pressure when starting up from cold. To assist the oil circulation under these conditions, a pipe by-passes the cooler and the oil pump is supplied through a spring-loaded non-return valve direct from the base of the air receiver.
9. A non-return valve is fitted on the outlet branch of the H.P. stator to prevent oil from flowing back into the commpressor from the oil reservoir when the unit is being towed or left standing on a gradient.

Rotary Screw Type Compressor :

These compressors are manufactured by Kirloskar Pneumatic Ltd. K.G. Khosla Compressors Ltd., Elgi and others.

Kirloskar Pneumatic Co. Ltd. manufactures a versatile range on rotary twin-screw air compressors in the capacity range of 7m³/min. to 21.2 m³/min and working pressure upto 10.5 kgf/cm². These are available in stationary as well as portable models.

Elgi Compressors Ltd. manufacture air screw compressors with a maximum free air delivery of 42.5m³/min. The pressure may be upto 17.5 kgf/cm² gauge.

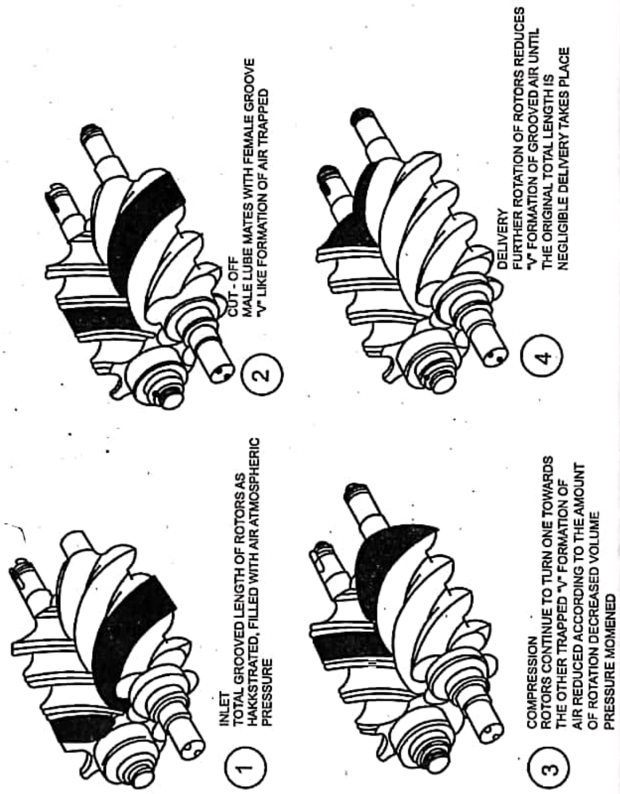


Fig. 10.4 Principle of air screw compressor.

Compressor consists basically of two helically cut intermeshing rotors, one male and the other, female. The male has four lobes and the female, six flutes. The rotors turn in opposite directions in a common casing. Due to a positive clearance between the rotors and rotor casing, wear and frictional resistancse are reduced to a minimum (Fig. 10.4).

The compressor is single-stage. The drive is direct and it is transmitted through a pair of step up gear from the engine crank-shaft to the male rotor; step up gears have 2.17:1 ratio between engine and compressor; no clutch is fitted. The helical engagement of the two rotors transmits the drive to the female. Air flow is in the axial direction and the compressor has a built-in compression ratio. Oil which is used as cooling and sealing medium, also lubricates the bearings. The lobes on the male rotor and the flutes on the female rotor are filled with air atmospheric pressure. This air is carried within the male and female rotors around the the casing. As the male lope intermeshes with its corresponding female flute, air is trapped, compression commences and oil, which is being continuously injected, mixes with the air. Further rotation of the rotors reduces the effective length of the meshing lobes and flutes until the trapped air/oil mixture is compressed to a predetermined outlet pressure. It is then discharged through the delivery port to the air/oil tank via a non-return valve. The air/oil mixture enters the tank and the now is so directed that the oil spreads along the internal surface of the shell. In this way the majority of the oil is separated from the air and drains to the bottom of the tank.

When the air delivery cock is opened, air in the reclaimer delivery pipe is released to work. At the same time air passes rapidly from the air/oil tank into the oil reclaimer via the inlet deflector where any remaining oil is effectively rmoved by passage through mechanical and wool-packed filters. Oil from the inside of the filter packs is then fed back to the inlet casing via connecting pipes and an internal port, while oil from the main section of the reclaimer unit drains into the air/oil tank and is recirculated via the oil cooler by the oil pump. The arrangement is similar to that in the rotary vane type compressor.

Turbo or Centrifugal Compressor :

The principle of operation of a centrifugal compressor is the same as that of a centrifugal pump. The centrifugal compressor is sometimes referred to as "blower". A turbo compressor works in the same manner as a turbine pump and the principle of centrifugal force is utilised to attain the desired pressure in the centrifugal and the turbo compressors.

As in a turbine pump, a number of steel impellers are mounted on a shaft. The impellers have backward curved blades (also called vanes). The shaft, and alongwith it the impellers keyed to it, rotate at a high speed within a fixed casing known as stator. The stator is provided with diverging channels DD known as diffusers which open out into the return channels RR which have progressively increasing cross-section (Fig. 10.5). As the rotor revolves, air enters the first impeller which, by virtue of centrifugal force, throws it to its periphery with a higher velocity and pressure. The air then enters the diffuser where its kinetic energy is converted into pressure energy, and with such higher pressure energy and a little kinetic energy the air enters the second impeller where the same process is repeated. Such repetition in a number of impellers raises the pressure of the air which leaves the compressor outlet at the desired pressure. Each set of an impeller and diffuser is known as a stage and for a gauge pressure of 5 to 8kgf/cm^2 about 10 to 15 stages are essential because the ratio of compression in each stage does not exceed 1.2. Such large number of stages necessitates a long shaft. As the air is compressed during its passage through successive stages, the radius of the impellers is gradually reduced towards the outlet end of the compressor. Water jacketing of the outlet casing and cold water circulation is the general method adopted for cooling the compressor, there being no intercoolers.

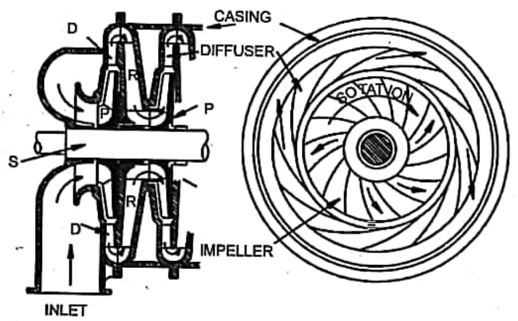


Fig. 10.5 Turbo or Centrifugal compressor in section.

In these type of compressors, since the compression is so rapid, there is little time for heat exchange between the gas and the surroundings and therefore the compression is very nearly adiabatic. The velocity of air through the compressor is high so that the air encounters considerable friction both internally and with the compressor walls. The changes in the direction of air flow results in turbulence and shock. All this generates internal energy within the air and produceds a temperature higher than the theoretical adiabatic temperature. The compression is adiabatic and intercooler cannot be used a sit becomes inferective due to the high velocity of air within the compressor.

Turbo compressors are not used as portable equipment in mining practice. They are suitable for large quantities of air but small pressure; minimum quantity is nearly 170m^3 of free air/min. and the maximum pressure attainble is about 11 kgf/cm^2. Reciprocating compressors are suitable even for small quantities and for high pressures. Turbo compessors are high speed machines operating at 10 000 to 30 000 r.p.m.

Rotary twin-lobe compressor :

The twin-lobe compressor or rotary two-impeller blower for sometimes called Root's blower is a valveless displacement machine consisting essentially of two symmetrical lobes mounted on parallel shafts inside a cylinderical casing. When these lobes rotate in opposite directions within the housing of the compressor, each traps between it and the cylinder a known volume of air carrying it to, and forcing it out positively through, the discharge opening. The action occurs twice for each revolution of one pair of lobes. The rotors intermesh but their internal clearance is maintained by a pair of timing gears. The compression space is not lubricated and the compressor are aircooled. Since these compressors displace a definite amount of air with each revolution, the volume moved at any r.p.m. and pressure can be calculated or speed can be selected for desired capacity. The P-V diagram for such compressor is rectangular which means a low efficiency. This restricts the use of this compressor to rather low pressure ratios. (Fig. 10.6).

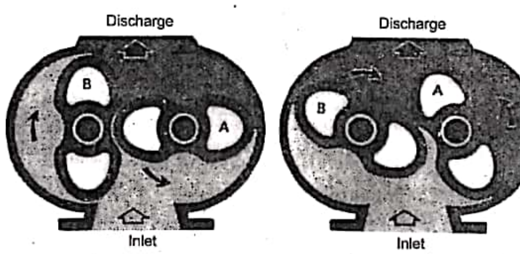


Fig. 10.6 Root's blower or rotary twin lobe compressor.

Kay International (P) Ltd. manufactures Kay twin lobe compressors upto 50,000 m^3/hr capacity. The pressure are upto 1 kgf/cm^2 (single stage and upto 2 kgf/cm^2 in two-stage).

Unloading arrangements :

The demand of compressed air from the receiver is very often fluctuating. When it is low, working of the compressor for its normal output is simply wasteful. There are three methods of varying the output of compressed air to keep pace with the demand; *(a) by varying the speed, (b) by varying quantity of sucked air at constant speed, (c) by automatically starting and stopping the compressor at pre-determined pressure limits,* or by varying the speed by predetermined pressure limits. All governors, regulators and unloading devices are controlled by variations of pressure of air in the receiver. An automatic device that varies the amount of air sucked by a compressor, is called an *unloader.* With a constant speed compressor, for example, the one havng an A.C. synchronous motors as prime mover, the unloader permits the driving unit to run at full speed without delivery of more air than is required. An unloader also facilitates starting a compressor, specially one without clutch provision, permitting a short-period operation at start without doing full amount of work required for normal compression.

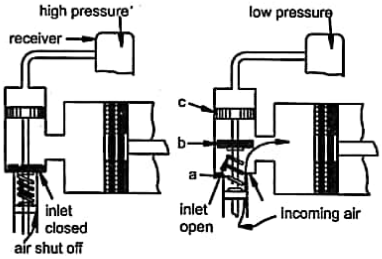


Fig. 10.7 Unloader valve arrangement in a reciprocating compressor

Fig. 10.7 illustrates the basic operating principle of an unloader on air inlet line. On a spindle air mounted a valve *b* and a piston *c* which is acted upon on one side by the air pressure from the receiver and resisted by a spring a from the other side.

When the pressure is low in the receiver, the spring pushes the piston *c* and valve *b* and opens the air inlet. The free air is then drawn into the cylinder during the suction stroke and the compressor pumps up the pressure in the receiver. When the pressure in the receiver builds up to a predetermined value, the piston *c* overcomes the spring tension and forces valve *b* on its seat,

thereby closing the air inlet of the compressor. When this happens the compressor, through working, cannot pump air into the receiver. As the air from that receiver is consumed in the air motor and its pressure drops below a predetermined working pressure the unloader valve functions again to admit air into the cylinder.

Another method of unloading, widely adopted, is to provide an automatic arrangement, operated by receiver pressure, which holds open the inlet valve when the pressure reaches 7kgf/cm² or a predetermined higher limit. The air which enters the cylinder during suction stroke is thrown out of the same inlet valve during compression stroke and only negligible work is done by the compressor. A governor on the prime mover reduces its speed at the same time when the unloading takes place. This continues till the receiver pressure comes down to 5.5 kgf/cm² or a predetermined lower limit when the inlet valve again functions in its normal way and air is compressed.

In both the arrangements described here, during the unloading operation, the engine and compressor continue to run, though at a lower speed.

On the Holumn Rotair 600 and Kirolskar rotary compressors WR type, the speed control and progressive offloading system controls the operation of the compressor unit entirely by the demand for air. The output controller operated by the air receiver pressure regulates the engine speed from full speed to a half and thus regulates the air demand down to half capacity. Should the air demand still fall, the inelt is progressively closed to supply demands of less than half the machine capacity down to zero.

A compressor should suck cool and clean air free from direct and particularly abrasive rock dust. Air filters on the intake pipes are highly advisable. Stationary compressors suck the cool air from outside the compressor room which is usually hot; portable compressors in mines and quarries should be so positioned that the inlet of suction air is away from holes being drilled if rock dust is not effectively suppressed and wind direction should also be considered.

The compressed air delivered by a compressor is discharged into a large cylindrical vessel with convex ends called receiver where the pressure fluctuations even out. The receiver thus acts as a reservoir of compressed air with nearly uniform pressure. The lubricating oil carried with the compressed air and the moisture contained in it are also deposited in the receiver and they can be drained away when desired. The fittings on a receiver are : (i) a drain cock at the bottom to drain out water, oil and outer sediments, (ii) inlet pipe for compressed air, (iii) main or delivery valve, (iv) pressure gauge (v) safety valve of spring type having hand trip mechanism by which it can be opened and any sediment blown out daily (vi) manhole on large receivers of stationary compressors (vii) in some cases only a fusible plug which will melt if the air temperature gets high enough to be near the flash point of lubricating oil vapour.

In two-stage or multi-stage compressors a relief valve is fitted on the intercooler also. The safety valves are set to operate when the pressure exceeds 0.35 kgf/cm² more than the working pressure of the system controlled by it. A smaller air pipe from the receiver goes to a pressure gauge and to the automatic controls which are operated by air pressure.

An air receiver do different design is incorporated in a compressor unit, model VT5PD (1) manufactured by Atlas Copco (India) Ltd. This reciprocating compressor with a free air delivery of 7m³.min. has a tubular air receiver incorporated in the chassis frame, providing a low centre of gravity and good road holing.

During normal operation the drain cock, at the bottom of the receiver should be opened at intervals of 3-4 hours to remove the water that may be formed due to condensation. The safety valves on the receiver and the intercooler should be tripped by hand at least once a shift to blow out any carbon or sludge deposists which might prevent them from working in an emergency.

The filters on the intake side of both engine and the compressor should be cleaned as often as may be considered necessary after stopping the compressor and the engine.

Moisture in compressed air system :

Atmospheric air always contains some quantity of water vapour. The maximum amout of water vapour that a gives space can hold is the same whether that space is a vacuum or filled with air or other gas. It should also be noted that the capacity of the space to contain water vapour depends only on the temperature and is independent of the quantity of air present or its *pressure*. If a given volume of air holds less than the maximum weight of water vapour it is capable of carrying, it is said to be unsaturated, but if it holdes the maximum weight or more, it is said to be saturated and its relative humidity is 100%.

$$\text{Relative humidity} = \frac{\textit{mass of water vapour per } m^3 \textit{ of air}}{\textit{mass of water vapour requried to saturate one } m^3 \textit{ of air}}$$

The relative humidity of saturated air is 100%. The percentage of saturation, except in arid regions, is rarely less than 35% and it higher near the sea shore.

In a compressor when a given volume of air is compressed, temperature remaining unchanged, its content of water vapour will saturat the space, reduced due to compression, and the excess water vapour is then deposited as water. But the temperature of air rises during compression, and the compressed air is able to retain the moisture in the form of water vapour at the high temperature inspite of the reduced volume. On cooling, the water vapour reaches a saturation point and the excess water vapour is deposited as water in the compressed air system including intercooler, aftercooler, air receiver and the transmission pipe line. The condensed water, if not removed before using the compressed air in air motors or other equipment, causes water hammer and leaky pipe joints. It also accumulates a low points in the pipe line resulting in restricted air passage, more friction to air flow and thus loss of power. It is therefore essential to drain water from the compressor cylinders, intercoolers, aftercoolers, air-receivers and air transmission pipes before start of work and at intervals during working.

Some receivers are fitted with automatic condensate draining devices:

Water trap. Most compressor installation are provided with after coolers and water traps to separate condensed water from the compressed air. With a good water trap, a separating efficiency of 80–90% is generally obtainable. The rest of the condensate accompanies the compressed air as a mist into the air receiver. The velocity of the air is decreased considerably in the receiver and most of the residual condensate will settle on the receiver walls and run down to the bottom drain of the receiver.

Freezing at exhaust ports of air engines :

The compressed air in the air receivers and transmission pipe lines that reaches the air motors is often in a saturated state after it has cooled down to near atmospheric temperature. If the compressed air is used expansively in the air motors so that the exhaust air is at approximately atmospheric pressure, its final temperature may be of the order of –20°C to –40°C and the water in the air will freeze into ice which will choke the exhaust passages and hamper the working of the air motors. The ways to prevent such ice formation are :

i. use of auotmatic water traps at low points in air transmission pipes.
ii. to drain the receiver, intercooler, aftercooler and pipe lines before admitting compressed air into them.
iii. installation of additional receivers near the air engines using them. This is rarely practised.
iv. suitable design of exhaust passages in the air engines.
v. to use compressed air practically non-expansively.

If compressed air is used with a late cut-off the air will be used practically no-expensively but the exhaust air will still have a temperature much lower than the working place temperature and in deep mines where the strata temperature is higher at the working places, such low-temperature exhaust has a cooling effect. Electrically powered machines, in contrast, generate heat during working and their use at the working places in deep mines, apart from other hazards, is a disadvantage despite the high efficiency of electric motor compared to the low efficiency of air engines.

Compressor efficiency :

The efficiency of a compressor may be expressed in any one of the following ways.

1. *Volumetric efficiency.*
2. *Isothermal efficiency.*
3. *Adiabatic efficiency.*
4. *Mechanical efficiency.*
5. *Overall efficiency.*

$$\text{Volumetric efficiency} = \frac{\text{Quantity of air delivered in a given time}}{\text{Piston displacement in the same period}}$$

The air delivered refers to the capacity of the compressor and its the actual volume delivered in m^3/min. expressed in terms of free air at the inlet temperature and pressure.

$$\text{Mechanical efficiency} = \frac{\text{Actual air indicated power}}{\text{Brake power of prime mover}}$$

$$\text{Overall efficiency} = \frac{\text{Calculated isothermal power to compressor and deliver the air.}}{\text{Measured electrical power input}}$$

The overall efficiency thus induces all electrical, frictional and thermodynamic losses. The overall efficiency of compressors and their prime movers is usually of the order of 60% to 70%. The basis of comparison may be either the isothermal or adiabatic theoretical power, according to which one is specified.

Transmission of compressed air :

On the receiver the mounting after the main valve or stop valve is a manifold for two or more tools and branch pipes, and each feed pipe is fitted with a stop valve. Rubber hose pipes, oil resistant, carry compressed air to portbale tools. Hose pipes suitable for water under pressure cannot be used as a substitute since compressed air invariably carries oil particles in suspension in the air stream. The 25mm bore hose, generally used for wagon drills, lasts about 100 shifts (8-hr shifts) and the 19mm bore hose, mainly used for jack hammers lasts for about 200 shifts. The pressure drop in transmission line depends on size and length of pipe, pipe bends and types of fitting used, volume of air flowing and the pressure of air. The pressure drop in the hose pipes is heavier than in equivalent size of steel pipes and it is an advantage to connect a large diameter long steel pipe to the receiver and fix up a manifold at the other end for hose pipe connections. This prolongs the life of hose pipes and enables operation of the compressor from a place comparatively more distant from the dust-raising drills resulting in a beneficial effect on the filters and the cylinders of the engine and compressor. A hose pipe should not be used after the inner tubing starts to deteriorate as broken pieces will clog filters or valves in the tools. Such pieces are often so small as to be mistaken for carbon.

Table 1 shows the pressure drop of compressed air at various points and fittings in a transmission line. Fig. 10.8 shows a nomogram of air pressure drop.

Compressed air driven tools and machines are generally designed for an operating pressure of 6–7kgf/cm^2 upon which the air consumption figures issued by the makers are based. 25-mm diam. branch lines to the points of consumption, provided with to 19-mm diam. Outlets will cover practically all comprssed air requirements in workshops and in mines. The permanent transmission pipes should be so dimensioned that the total pressure drop does.

TABLE 1

Item		Equivalent pipe length in m — Inner pipe dia. in mm						
		25	40	50	80	100	125	150
gate valve; full open		0.3	0.5	0.6	1.0	1.3	1.6	1.9
diaphragm valve; full open		1.5	2.5	3.0	4.5	6	8	10
globe valve; full open		7.5	12	15	24	30	38	45
bend R = 2d		0.3	0.5	0.6	1.0	1.2	1.5	1.8
bend R = d		0.4	0.6	0.8	1.3	1.6	2.0	2.4
Tee		0.5	0.8	1.0	1.6	2.0	2.5	3
Side outlet tree		1.5	2.4	3.0	4.8	6.0	7.5	90
reducer		0.5	0.7	1.0	2.0	2.5	3.1	3.6

Example : What is the pressure drop when 180 1/s at 8 bar absolute pressure flows through a 300 m pipe of 90 mm dia. ?

***Ans. :* 0. 10 bar.**

not exceed 0.3 kgf/cm² including drops in connections and fittings. Depending on the length of the pipe line the air flow speed should be between 6 and 10 m/s to give moderate pressure drops. The following table indicates approximate value of pressure drop for usual pipe sizes and flow rates, valid for a pipe length of 100 m and pressure of 7 kgf/cm².

Pipe size; bore in mm	25	38	50	63	76	88	100
Air flow, m³/min.	1.5	3	6	10	15	25	35
Pressure drop, kgf/cm²	0.2	0.1	0.09	0.08	0.08	0.08	0.08

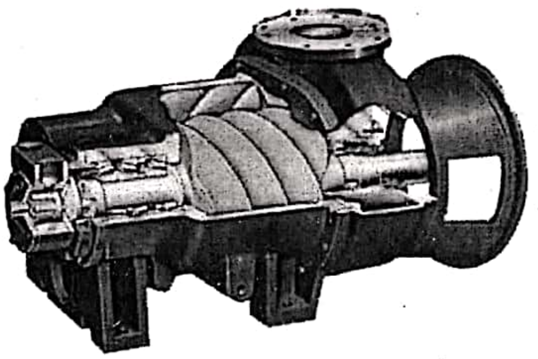


Fig. 10.8 Cross-sectional view of an air screw compressor
courtesy of K.G. Khosla Compressor Ltd.

Leakage of compressor air :

In most permanet installations with penumatic tools and machines the leakage should not exceed 5% of the total compressor capacity. In mines and quarries the pipe system is often long with many branch lines which are frequently temporary in nature. A leakage loss of upto 10% can be tolerated in such cases. Damaged packings and other sealing elements, O-rings, and loose hose couplings are main causes of leakage.

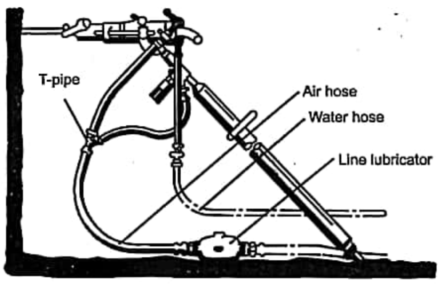


Fig. 10.9. An oil bottle (line lubricator) for a jack hammer.

Utilisation of compressed air :

Pressure energy of compressed air is transferred into mechanical energy in a compressed air engine or air motor. The air under pressure is put to use machines like reciprocating air engines, compressed air motors of vane type or piston type and in air turbines.

A reciprocating compressed air engine words in the same manner as a conventional steam engine with slide valve. An early cut-off is not used in such reciprocating engines to avoid freezing at the exhaust ports (Fig. 10.10) Speed is controlled by a throttle valve.

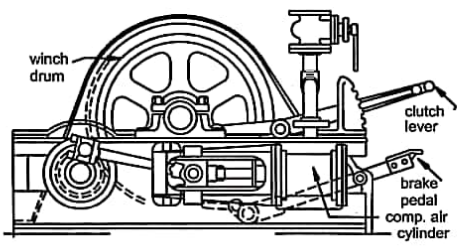


Fig. 10.10 A reciprocating compressed air engine.

In the piston-type air motor the air expands in the cylinders and in the vane type motor between the vanes. Piston motors have from 4 to 6 cylinders in star formation. They work with low rotary speeds, most frequently under 5000 rpm. In the vane motor the cylinder volume is limited by the tube shaped cylinder, the rotor the vanes and axially, by the cylinder end plates with ball bearings. The rotor is positioned off-centre in relation to the cylinder. Grooves in the cylinder walls admit and exhaust the air. When the motor is started the vanes are forced against the cylinder wall by compressed air which is conveyed to the vane grooves through grooves in the cylinder end plates. When the rotor has gained speed the vanes are kept in contact with the cylinder wall principaly by centrifugal force. Idling speed is controlled by throttling, or by a centrifugal speed governor. The principal feature of the vane-type motor is the high powe in relation to mass which makes it well suited to use in portable tools. At constant throttle the speed adjust itself to the load. Vane motors are widely used in handheld drilling and grinding machines, and they have almost completely replaced the piston-type air motors in hand-held machines. However, in hoists and winches piston motors with their better starting and low speed properties are often used.

The familiar jack hammer drill widely used in mines is of the piston type and its method of working is much different from that of a compressed air engine.

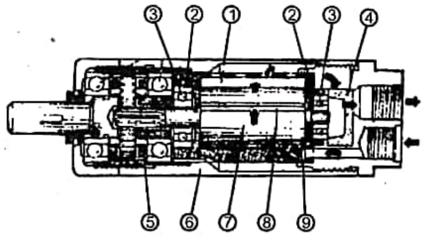


Fig. 10.11 A vane type air motor in section.

1. Cylinder 2. end plate 3. ball bearing 4. silencer 5. planetary gear 6. motor housing 7. rotor 8. vane 9. air inlet to cylinder.

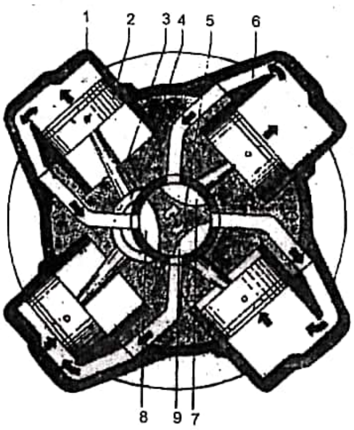


Fig. 10.12 Piston type air motor in a tool.

1. Cylinder 2. Piston 3. Connecting rod 4. Crankshaft 5. Rotary valve 6. Air channel to cylinder 7. Main air outlet through rotary valve. 8. Residual air outlet through rotary valve 9 air inlet through rotary valve.

Jackhammer :

Fig. 10.13 shows a jackhammer in section. Compressed air enters the drill through a hose pipe. (19mm bore) and a curved metal tube with swivel connection. The speed of the drill is manually controlled by the throttle valve which regulates the quantity of air.

The air passage is past the pawls on the rifle bar into an automatic valve which directs it alternately to the top and bottom of the cylinder. As the piston is forced down by the air pressure, its lower portion, called the stem, strikes the upper end of the drill steel through the medium of a chuck. In the top of the piston is a riffle nut having splines which matches with those of the rifle bar or twist bar. These splines are not straight but slightly twisted.

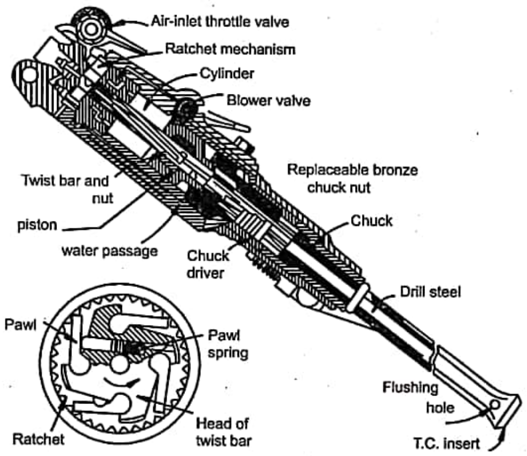


Fig. 10.13 A jack hammer drill in section.

The upward movement of the piston is also by compressed air entering the bottom of the cylinder through the automatic valve located around the upper part of the rifle bar and as the piston moves up, the rifle nut splies exert a twisting force on both the rifle bar and the piston. The rifle bar is held by the ratchet, and thus compels the piston to turn. On the downward stroke of the piston, the rifle bar turns. The piston, therefore, experiences slight rotation, nearly $\frac{1}{12}$th of one revolution, during the complete cycle of one up and one down stroke. The extent of rotation is dependent on the twist in the rifle bar and the spacing of the ratchet teeth.

The steam of the piston has straight splines corresponding with those of the chuck driver. This chuck driver is threaded into the chuck which has, at its lower end, a socket to receive the shank of the drill steel. The twist which the piston undergoes in each cycle of up and down movements is transmitted through steam to the chuck driver; through it to be chuck, and through the chuck socket to the drill steel. The drill steel, therefore, receives hammering during each down stroke of the piston and between successive storkes, it receives a slight twist as well so that the drill bit strikes new surface of the rock during each stroke. The number of strokes per minute are about 22000. When jackhammer is working, air from the top of the cylinder exhauss into the atmosphere. Exhaust from below the piston goes through a drilled passage into the piston steam bearing and further along the chuck driver splines into the space below the piston stem. There it enters the central longitudinal hole of the drill rod and reaches the drill bit to clear away the drill cuttings. The air also keeps the drill bit cool. The air consumption is nearly 3–3.5 m^3 per min. of free air.

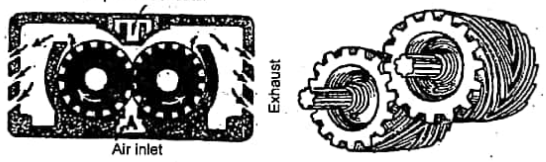
It is important that the shank be of the correct length, as the drill willnot operate properly on a longer a shorter stroke than that for which it is designed.

Generally hard rocks require a slower and heavier blow than do the softer ones, while such rocks as freely cutting non-abrasive limestone respond to quicker rotations.

Air turbines :

In the impulse-type turbine motor the air pressure is transformed to kinetic energy in air jet, which powers a turbine wheel. This motor type has a better power/mass ratio than the vane motor, but as the peripheral speed of the rotor is very high it is difficult to gear it down to practical speeds. This is the reason why it is used mainly in tools such as high speed die grinders. In the coal mines compressed air driven coal cutting machiens and some conveyors employ air turbines. Since the air turbines run at high speeds, the load connected to it should not be suddenly removed as there is danger of the turbine racing up and even the rotor may burst. Such situation may arise, for example, when a conveyor belt breaks. To prevent this, a centrifugal governor is fitted.

An air turbine consists of two cast iron rotors having double helical teeth meshing together so as to form an air sea all along the line of contact. The rotors move on ball bearings in a close-fitting casing and air is admitted below and between them (Fig. 10.14). The radial clearance is very small varying from 0.05mm to 0.07mm and the only way air can expand is in the V-shaped spaces between the teeth. As it expands, it exerts a turning moment on the rotors causing them to revolve in opposite directions as indicated by the arrows. The air, after expansion, then discharges to exhaust at either side. An air turbine developes an even torque and it can start from any position equally well as there is no dead centre. The rotors run in opposite directions but the rotation cannot be reversed.



Left – Compressed air turbine. *Right – Air turbine rotors.*

Fig. 10.14

For effecting reversal of the drive, in some designs a pinion is fitted on each rotor and a sliding pinion is fitted on the drive shaft and the drive shaft pinion is capable of engaging either rotor pinion.

Fig. 10.14 shows two rotors in mesh and they are lubricated. In oil siphoned from the reservoir and dripping on the rotors by an air turbine the power developed is roughly proportional to the air pressure. Clean dry air is essential for their satisfactory operation and an air strainer is generally an integral part of the machine.

Compressed air Vs electricity as power :

Where large power requirement is involved electricity is definitely a superior power for generation, distribution, transmission and utilisation. For certain special applications, compressed air, however, is best suited; for example, percussive rock drilling in hard rocks, mechanical picks and pusher rams. For underground coal mines electric sparks are always dangerous and compressed air is a very safe power, particularly at the coal faces with heavy gas emission.

Efficiency, *i.e.*, $\dfrac{\text{power output of generator}}{\text{power input to generate shaft}}$ is nearly 80 to 90% for electric generators but $\dfrac{\text{actual air indicator power}}{\text{brake power of prime mover}}$ is the order of 70% for air brake power of prime mover compressors. During transmission pressure drop due to friction, and the leakage, accounts for only 60% to 70% efficiency with compressed air in reasonably well maintained installations and it may drop down to 50% and even low, if maintenance is neglected. With electricity the efficiency in transmission is 85 to 90%. Utilisation of compressed air, which cannot be used with any large degree of expansion in air motors due to danger of freezing at exhaust passages, is notoriously inefficient, with only 40 to 50% efficiency. Electric motors are nearly 85 to 90% efficient in operation. The exhaust of compressed air motors has a cooling effect on the surroundings, a point in its favour in deep hot mines. Electric motors, on the other hand, generate heat during operation.

QUESTIONS

1. What are the different types of compressors used for mining ?

   Describe a 2-stage reciprocating compressor.

2. Write shorts notes on : unloading of compressor, moisture in compressed air and its effect, compressor efficiency.

4. Give approximately values for pressure drops of compressed air in a transmission line including the drop at various fittings on the line. Assume air pressure to be 7kgf cm^2 (gauge). What steps should be taken to ensure minimum pressure drop.

5. Compare the advantages and disadvantages of a reciprocating compressor with those of a centrifugal compressor, and state where each one can be more conveniently used.

O O O

CHAPTER – 11

WINDING : GASES & SHAFT FITTINGS

For hoisting of mineral from an underground mine and for lowering and raising of men a lift-like conveyance, called cage of skip, is used. Other equipment for raising of mineral and men include :

i. a tower-like structure of steel or concrete called headgear

ii. winding rope.

iii. attachment gear between the cage and the rope, such as cage chains, rope capel and distributing plate.

iv. headgear pulleys.

v. winding drum

vi. steam engine / electric motor and its controlling apparatus .

vii. arrangement for guiding the cage during its ascent and descent.

Headgear :

The headgear is a steel or concrete frame work on the shaft mouth. Its purpose is (i) to support the headgear pulleys, the weight of the hoisting ropes, cages and the rope guides, and (ii) to guide the cage to the banking level. It should withstand dead and live loads and wind pressure. The dead loads on the headgear are reasonably constant and calculable but the live load due to winding is a variable one depending on the length of ropes in the shaft, the contents of the cages and the rate of acceleration or deceleration. Headgears used for tower mounted Koepe winders are designed to carry in addition the load of motors, winding pulley and other equipment for winding.

The headgear consists of nearly vertical columns or girders braced with horizontal girders. The members narrow at the top and battered at 1 in 8 to 1 in 10 for a larger width at the foundations. Of the four legs the two nearly vertical main legs are connected to two inclined back legs (towards the winding engine room). The top of the headgear has a steel platform or plate and the bush bearings of the winding pulleys rest on the vertical members of the headgear frame. It is usual to design the upright members of the headgear frame to carry the dead weights and wind pressure, leaving the back legs to take care of the resultant of the live loads due to the ropes and cages.

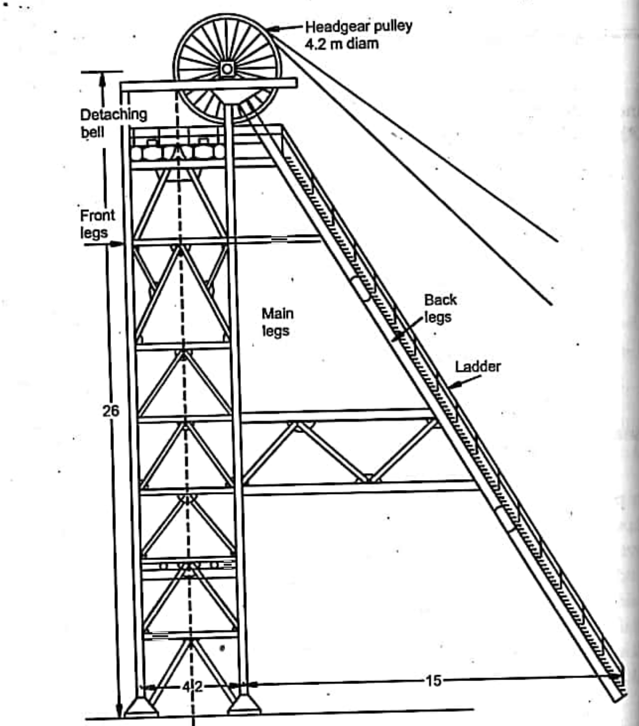


Fig. 11.1 Headgear ; measurements in metres.

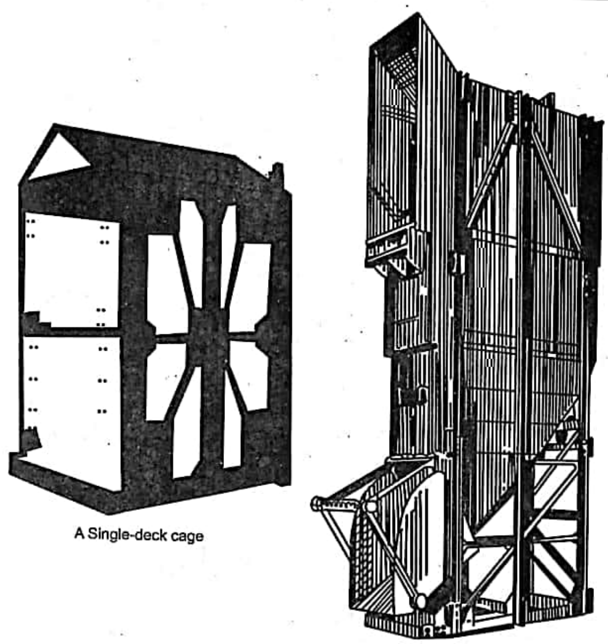


Fig. 11.2 A single-deck cage

The height of the headgear is decided by considerations of number of decks on a cage, banking level or skip discharging point, pit-top layout, and depth of the shaft.

The level of joists carrying the detaching plate or bell above the decking level is equal to the overall height of cage/skip, plus length of cage chains and suspension gear plus a margin of 3-7 m. This margin of 3-7 m allows a cage to be lifted for materials to be slung beneath it.

The headgear pulleys should be at height above the detaching plate the the rope capel is released before it comes into contact with headgear pulley. The distance is about 3 m.

A derrick is fitted on some headgear to facilitate lifting of the headgear pulleys at the time of replacement or repairs.

Headgear of wood are prohibited by Law.

Headgear Pulleys :

The headgear pulley should have as large a diameter as possible to minimise bending streesses in the winding rope. Its diameter should be at least 100 times the rope diameter. Pulleys of over 2.5 m diameter are generally constructed in two halves and bolted together. Normally the diameter of the groove of the headgear pulley should be 110% of the rope diameter for stranded ropes and 105 % for locked coil ropes. This ensures that 1/3rd of the circumference of the rope is in contact with the groove . A lesser angle of contact causes excessive strain on the rope and wear on the pulley. The headgear pulley is keyed to a mild steel forged shaft which rests in plain bushed journal bearings.

The angle of fleet which is the angle between the vertical plane of the pulley and the rope when the cage is at the pit-top or pit bottom, should not exceed 1.5°. More fleet angle results in wear of the rope and wear of the pulley also. (Fig. 11.3)

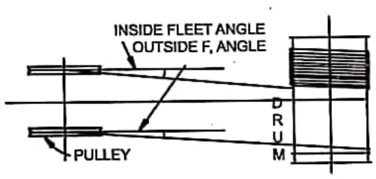


Fig. 11.3 Fleet angle

The shafts of the two head gear pulleys which are placed side by side are in a horizontal line and their planes of rotation are vertical and parallel. In the case of Koepe winders, ground mounted, the planes of rotation of the two headgear pulleys are one below another (though not vertically one below another).There is therefore no fleet angle in the case of a Koepe winder pulley, (Fig 11.4). If a drum winder is used for a deep shaft, it may be necessary to consider double layer coiling of rope in order to accommodate all the rope on the drum and keep the fleet angle limited to 1.5°. *(See also pg. 264.)*

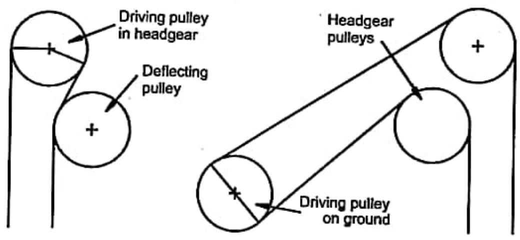


***Fig. 11.4** Arrangment of driving sheave &pulleys in koepe winding*
Left-tower mounted ; right-ground mounted..

Cage and shaft fittings :

The cage is a lift suspended from the winding rope, open at both ends where gates can be positioned during man riding and it has rails fitted to the floor for mine cars or tubs.To prevent the mine car/tubs from falling outside the cages, catches are provided on the floor which act against the axles of the mine car/tub; in addition, a long bar, turned at both ends and hinged at one side of the cage; prevents movement of the tubs during travel up or down the shaft. Cages used for man riding have a hand-bar near the roof for the men to hold and at both ends collapsible gates are provided which can be closed or opened manually or by compressed air. The roof has a hinged or removable door for accommodating long timber or rails whenever necessary. A cage which can accommodate only a single tub is called a single cage; a cage which can accomodate two tubs is called a tandem cage. In our country most of the cages whether single or tandem,are with only a single deck, but cages with two or more decks are used in mechanised mines dealing with large outputs e. g. Sudamdih, Mosbani, etc. The cage travels in a vertical plane only unlike the skip which travels in the vertical plane or along an Inclined plane, depending upon the type used. **See 2-deck cage on**

Skip :

The term skip is sometimes used for a cage of larger size which accommodates mine cars but very often is restricted to a lift which does not accomodate mine cars but can be filled with mineral through its top opening. Skips travelling in a vertical plane have a discharge opening at the bottom for unloading the mineral content but skips travelling on rails along an inclined haulage plane are so tilted, during travel, near the unloading end that their contents are discharged from the top end. Skips moving in a vertical plane are sometimes partitioned for accommodating men at the upper half and material/ mineral at the lower half.

Cages and skips are provided with cast steel guide shoes having malleable cast iron bushes. usually four shoes per cage or skip. The bushes are renewable.

A skip carries a large payload, usually 8 te or more, compared to the cage and the ratio $\frac{\text{payload}}{\text{gross weight of skip (loaded)}}$ is high for a skip

Skip winding Vs. cage winding :

Skip winding is best suited for deep shafts where high output is desirable in view of the large investment on deep sinking and the need for early return on such large outlay. The ratio $\frac{\text{payload}}{\text{gross load of loaded skip}}$ is heigh, nearly 0.6, in the case of skip winding but with cage winding the ratio is only about 0.35. With a skip installation, the number of mine cars used underground is less as they are not be raised to the surface; moreover such mine cars are independent of the size of the shaft or skip. Skip lends itself to automatic loading, unloading and decking operations, Thereby providing a quicker cycle of operations of winding of mineral. This also means less manpower required for skip installation. Trackless mining is possible from the working face right upto the surface. Pit-top and pit-bottom layouts for cage and skip are given in another chapter.

The provision of fully automatic operation for an A. C. cage winder is not simple, involving as it does, accurate landing of the cages at the decking level. On the other hand there are many installations of fully automatic Ward Leonard skip winders operating successfully in South African gold mines and in the coal mines of Great Britan, Germany and other European countries.

Skip winding has, however, the disadvantages that separate arrangements have to be made for winding of men and material, though some recent installations have modified the skips for manwinding. With skip winding it is difficult to import dirt, washery refuse or mill tailings for underground packing of goaf or stope. Degradation of mineral, particularly soft mineral

like coal, takes place during loading and unloading of skip and to prevent coal dust from entering the mine it is essential to install the skip in U. C. shaft. Winding of coal of mineral from different level is not as convenient as in cage, winding requires large excavations at the pit bottom to accomodate measuring pockets, tippler and small bunker to store the mineral. A higher headgear is essential and the shaft has also to be sunk deeper than the level of the mineral bed, as compared to cage winding.

Cage attachment to winding rope :

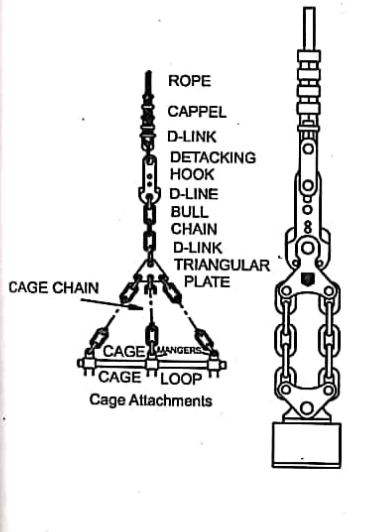


Fig. 11.5

Left (a)-typical rope attachment to cage
Right (b)- rope attachment at Mosabani copper mine.

A typical arrangement of attaching cage to the winding rope is shown in Fig. 11.5. Four cage chains in the case of the single cage (and 6 chains in the case of a tendem cage) attach the cage to a triangular distribution plate which is connected to a safety detaching hook through D-links or bull chains. The detaching hook is attached to the rope capel which may be a cone type capel or reliance capel.

The triangular distribution plate is wrought iron or mild steel. The cage chains and all D-links or shackles and bolts are of wrought iron or mild steel. Under the mining regulations all chains must be annealed at least once every six months. As an alternative to wrought iron or mild steel, the various chains, links and shackles, the distribution plate of 1.5% manganese steel which is exempted from periodical heat treatment.

Detaching hook :

Detaching hook which is always placed just below the rope capel, is a safety device which acts when an overwind takes place. Its purpose is to suspend the cage/skip in the headgear if an overwind occurs, and at the same time to release the rope (along with the capel) to go over the headgear pulley. Detaching hooks are used only for vertical shafts served by drum winders but they are not used on koepe winders. Our Regulations demand the use of detaching safety hook but in many countries such as U.S.A., Canada, U. S. S. R;, Germany, etc. the mining regulations do not insist on the provision of a detaching safety hook.

Two types of detaching hooks are in common use :

1 Ormerod detaching safety hook, and

2. King detaching safety hook.

Ormerod detaching safety hook :

The Oremerod detaching hook consists of three mild steel plates *i. e.* one centre plate and two outer plates. The plates are assembled on a common centre pivot and a copper rivet, 16 mm diam. passes through a small hole of all the plates when assembled. The upper rivet holds the three plates in their relative position after assembly. The top portion of plates is attached to a D-link hung from the rope capel by means of slotted holes (Fig. 11.6) and the bottom portions open out to give the appearance of a wedge. The bottom slotted holes in each plate carry the pin of D-link which supports the triangular distribution plate and the cage/skip. The outer plates actually transmit the load of the cage to the rope. In effect the centre plate is not taking any the load but is preventing the pins from moving laterally along the slots. A plate or a bell-set securely fixed on the horizontal member of the headgear frame, a few metres below the headgear pulleys, is required for operation of the safety hook in case of an overwind. The hook in its assembled form cannot pass clear through the bell-set.

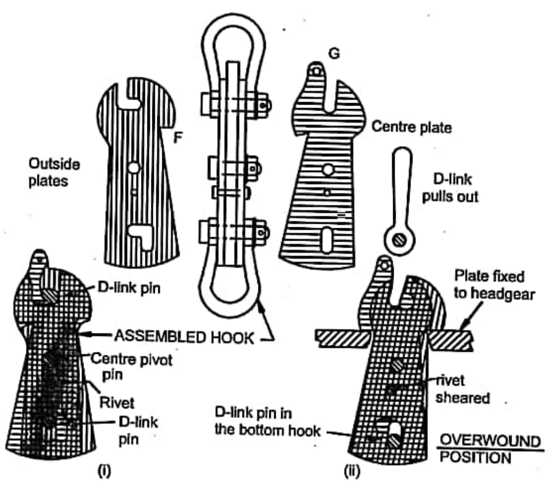


Fig. 11.6 Ormerod detaching safety hook

When an overwind takes place and the ascending cage rises high enough for the hook to pass through the bell, the wedgeshaped portions of the outer plates are pressed inwards. This causes the copper rivet to shear. The plates open out at the top, releasing the D-link pin of the rope capel which then passes over the headgear pulley. At the same time, the projections F of the outer plates open out and hook, released from the rope, rests on the bell-set by virtue of the projections F. The D-link pin in the bottom slotted holes assumes the position, after overwind, as shown in the figure and prevents the plates from returning to their original relative positions.

After an overwind, to put the cage back in operation, cover the pit top with a few rails so that the cage/skip, if accidentally released during restoration operation, does not fall through the pit. To remove the safety hook from bell-set, attach the rope to the top hole G of the centre plate by a D-link, lift the cage slightly and the outer plates will move downwards slightly so that the top part of the hook becomes narrower. This allows it to pass through the bell and the cage can be sent down to rest on the detaching hook and suspension gear; so it will be necessary in the interests of safety, to replace both hook and cage chains with a set which have been annealed.

King detaching safety hook :

It consists of 4 wrought iron plates, *i.e.* two being movable inner plates and two fixed outer plates. The two inner plates are placed together in opposite ways so that the hook m of one plate and that of other jointly form a secure hole for the reception of the rope capel bolt. A main bolt or centre-pin passes through the holes f in all four plates and serves (i) to bind the plates together; (ii) to transmit the tension of winding rope from the hooks of the inner plates to the shackle-bolt of the main D-link (placed through the hole O in the outer plates); and (iii) to provide a pivot on which the two inner plates can move. The hooks m are so curved that pull of the winding rope has no tendency to open out the inner plates. A copper pin is placed through the holes C in all four plates and riverted over to prevent inadvertent movement of the inner plates when they are not under tension. (Fig. 11.7 & 11.8)

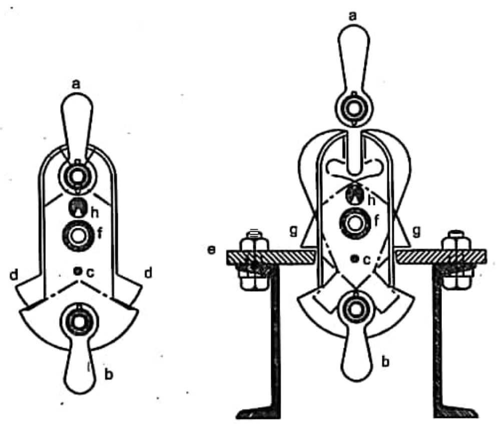


Fig. 11.7 King detaching safety hook

Left - hook assembled and in working order.
Righ t - hook detached and cage suspended during overwind

• During an overwind as the ascending cage goes up the hook is partially drawn through the circular hole in a catch plate, e, securely attached to a horizontal member of the headgear and the lower wing d of each inner plate is forced inwards. The copper pin is thus sheared and the hooks m are forcibly separated, so releasing the D-link of winding rope capel. Simultaneously, the catches g g´on the inner plates are forced outwards so that they rest on the upper side of the catch plate and the cage is therby safely held. When the weight of the cage is taken by the catches g g, the inward pressure of the wing d is borne by the stoping sides of a wedge shaped block (shown dotted in ii) which is placed between the lower ends of the two outer plates and is securely bolted to them.

For lowering the cage after an overwind, a vertical slot h is provided in each outer plate and an inclined slot in each inner plate. The cage being suspended, the slots in the outer plates remain vertical but those in the inner plates take different positions so that a clear, almost circular hole is still maintained through all four plates. To restore the cage, place a few rails across the shaft top. Bring the winding rope capel back over the pulley and attach it to the plates by a special D-link whose pin should pass clear through the hole at h. Raise the cage slightly and the pull of the rope on new D-link pin causes the latter to rise along the inclined faces of the inner slots. This forces the hooks m and catches g inwards to their normal positions. Now lower the cage to the banking level. Replace the hook and fit it with a new shearing pin. The catch plate e also should be changed.

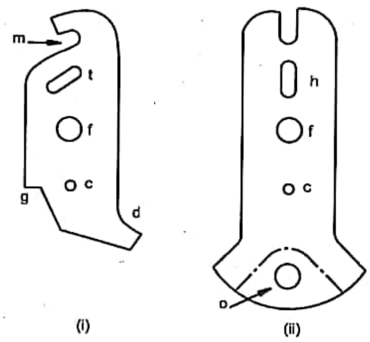


Fig. 11.8 King detaching safety hook.
Left - inner plate; Right - outer plate

The detaching hooks are made of wrought iron, or of 1.5% maganese steel or of good quality mild steel. The shearing pin is of very ductile copper. To maintain the detaching hook in efficient working condition it must be dismantled, examined, and refitted at least once in every six months.

The opening of the detaching plate or "bell" is tested monthly by calipers or gauges for wear occuring from lashing by the rope. All the suspension gear should be renewed after every 10 years.

Safety catches :

As a safeguard against the failure of the detaching plate to hold the cage, safety catches may be fitted in the headear. These safety catches consists basically of short levers mounted in the headgear at intervals that very from 0.3 to 1 m. These are located above the normal running position of the cage.

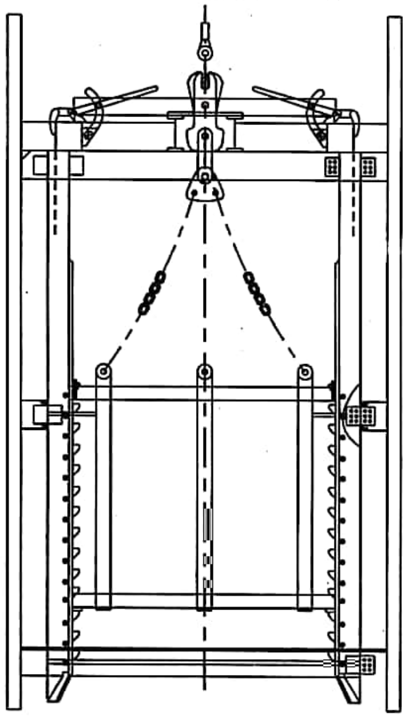


Fig. 11.9 Safety catches

The catches are free to turn on a pivot. In the event of an overwind the catches are lifted the cage to pass up into the head gear. they then fall back to the normal position and so prevent the cage falling back down the shaft. A mechanical linkage is provided so that all the catches may be withdrawn simultaneously in order to lower the cage after an overwind, or when the apparatus is to be checked/tested. This operation is performed by a single hand lever for each set of catches. The safety catches should be inspected regularly to prevent accumulation of dirt or coal dust and to ensure their free movements (Fig. 11.9).

The detaching safety hook provides safety for the ascending cage and arrests its ascent; the safety catches also provide for safety of ascending cage but no safety device is employed for the descending cage which in the event of overwind strikes the pit-bottom joists with full speed and the consequent damage to the installations and injuries to the persons travelling in the cage. If jists are avoided as supports for the pit-bottom cage, two safety arrangements are possible :

i. Where rigid guides are provided in the shaft, they may be arranged to gradually narrow down below the pit-bottom decking level.

ii. If rope guides are provided in the shaft, receivers, gradually narrowing down below the pit-bottom decking level, may be fitted.

These two arrangements are, however, possible with Koepe system of winding in which supporting joists are not to be provided at pit-bottom.

Keps :

Keps are retractable supports for cages and have to be used at the pit top under our mining regulations. Their use is not necessary at the pit bottom as the cages rest on the rigid platform of steel girders and wooden planks.

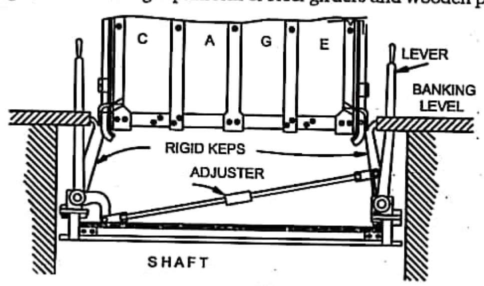


Fig. 11.10 Rigid Keps

Keps are not required at the mid-set landing and in a shaft served by koepe winding system. Keps ensure not only support to the cage but their use results in proper alignment of the cage-floor and decking level so that the stretch of the winding rope creates no difficulties arise and are overcome by the use of titled or higned platforms. Keps are manually operated by the banksman at the pit top. The ascending cage pushes the keps back and as it is raised slightly higher than the decking level, the keps fall back in position by gravity as the banksman releases the operating lever. The cage, after it has come to a halt, is lowered by the winding engineman to rest on the keps. When the top cage is to start on its downward journey, the winding engineman raises the cage only slightly to make it clear of the keps, the banksman withdraws the latter by manual operation of a leaver which is held by him till the cage is lowered past the keps.

The disadvatages of this type of rigid keps are :

1. Accumulation of slack rope on the pit bottom cage when the top cage is raised a litle for withdrawal of keps. Ascent of the pit bottom cage is generally associated with shock load on the winding rope and the stress may amount to nearly 200 % of the static load.
2. Loss of time and power of lifting the top cage before its downward travel.

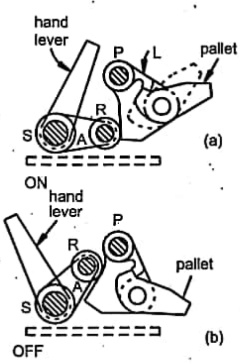


Fig. 11.11 Davies improved Kep-gear

Because of these defects keps have been devised which can be withdrawn from under the cage without lifting it. Notable of such keps are the Stauss keps and the Davies' improved keps gear. Fig 11.11 shows Davies' improved kep gear omitting all supporting plates and bearings to avoid confusion. The gear consists essentially of the shafts S to which is keyed the hand lever and a pair of arms A with a steel roller R mounted on a pin between the arms. The roller presses against a renewable roller path on a swing-lever L which is pivoted at Pand carries a "pallet" mounted on a steel pin at its other end. The pallet is free to move upward and around this pin, as shown dotted in Fig. 11.11 (a) thus permitting the upward passage of the cage, but it is prevented from

moving downwards by a projection on the lever L. The cage is thus securely supported on the upper surface of the pallet. The gear may be withdrawn, however, without first raising the cage, in the manner shown in Fig 11.11 (b) It will be seen that when the handlever is moved to the left, the roller R moves upwards along the roller path on lever L, thus allowing the lever to rotate downwards by gravity around the pin P until the pallet is clear of the cage.

Keps may be operated by hydraulic or pneumatic power. Where the keps are pneumatically operated they are interlocked with other decking equipment so that they can be withdrawn or brought into use at the correct time in the cycle of operations of the associated equipment at the pit top.

Tilted or hinged platform :

In deep mines the winding ropes stretch in course of use. If keps are used at the pit top, the loading or unloading of mine cars/tubs into the cage or out of it can be done without difficulty as the cage floor is in level with the decking level when the cage rests on the keps. In the case of koepe winders,since keps are not used, the rope stretch creats problems of alignment of cage floor with decking level and platforms have to be used.

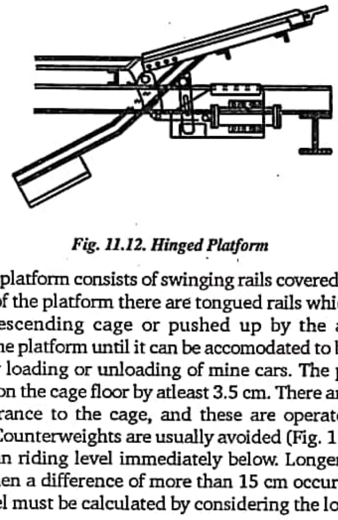


Fig. 11.12. Hinged Platform

A tilting platform consists of swinging rails covered with steel plates. At the frontend of the platform there are tongued rails which can be pushed down by the descending cage or pushed up by the ascending cage, Thereby tilting the platform until it can be accomodated to be in line with the decking level for loading or unloading of mine cars. The platform tongues overlap and rest on the cage floor by atleast 3.5 cm. There are two platforms, one at each entrance to the cage, and these are operated separately by compressed air. Counterweights are usually avoided (Fig. 11.12) as their use obstructs the man riding level immediately below. Longer tilting platform must be used when a difference of more than 15 cm occurs. The maximum difference in level must be calculated by considering the lowest level on the loading side and the highest level on the unloading side. The stretch of the winding rope is not so great at the pit top as at the pit bottom and so a shorter type of platform may be used at the surface.

Guides used in mine shafts :

The guides used in mine shafts are :

i. *rigid guides*

ii. *flexible guides or rope guides*

Rigid guides are of hard wood or steel (rail section)Example of wooden rigid guides are rare in this country. They are of rectangular cross-section, usually 10 cm ×20 cm, and are fixed by countersunk bolts to the buntons placed across the shaft at intervals of 1.8 m -3 m. They suffer from the risk of fire.

Steel rigid guides are installed in some deep shafts in this country. They are made of flat-bottomed or T-section rails weighing from 20-55 kgf/m lengths of upto 13 m. Owing to their shape and the manner in which the cage shoes embrace them, they need only be placed at one side of each cage. Only one line of buntons, in the middle of the shaft, is required for fixation of guides if the guide shoes are on inner sides of the cages but on either side of the bunton.

Flexible guides consists of wire ropes which may be of locked coil constructions or of 1×6 construction with thick wires. They are suspended in a vertical shaft from a secure attachment placed on the top cross member of the headgear while at the shaft bottom they are given the requisite amount of tensioning by placing cheese weights on them. These weights ensure correct vertically and also eliminate to a great extent oscillations of the guide ropes during a wind. In shafts which are not deep, 2 or 3 guides per cage suffice but for deep shafts 4 guides per cage is the standard practice and the guides are arranged near the corners of the cages. If the clearance between the cages and shaft sides is limited "buffer" ropes are arranged between the cages and outside of the cages-sides. These buffer ropes are not attached to the cages through shoes but are hung freely with proper tension. The minimum space between the 2 cages as the guide ropes oscillate 40 cms to prevent collision of the cages as the guide ropes oscillate during the wind, the maximum oscillation being at the mid-run of the cages. The tensioning weights are about 10 kN per 100m depth in deep shafts.

The capelled suspenders placed on the cross members of headgear may be in the form of a reliance capel (fig. 11.13 i) or in the form white-metal-filled conical capel, (fig. 11.13, iii) formed in the same manner as for wire ropes.

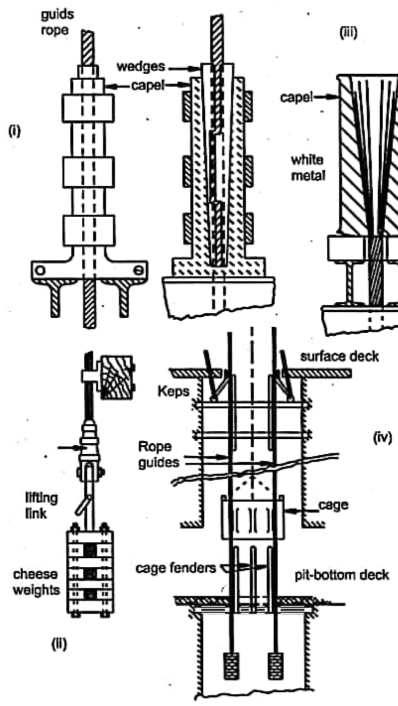


Fig. 11.13 Rope guides

Rigid guides require a shaft truly vertical and perfect alignment of the guides is essential. As oscillations of cages are non-existent, there is no risk of collision of cages, shafts need not be roomy and guides can be fitted in shafts of limited size, permitting practically entire cross-section, nearly 80 %, to be used by the cage/skip. The rope guides are cheap, easy to install and do not require frequent attention to maintenance. Buntons throuhout the shaft at intervals are not required. Flexible guides provide a smooth run for the cages compared to the rigid guides. The ventilating air current encounters practically no resistance in the shaft fitted with flexible guides unlike with rigid ones.

Flexible guides impose a considerable extra load, in some cases even upto 100 te, on the headgear. Mishap to the guide rope at any point means its complete replacement. Receivers are necessary in a shaft equipped with flexible guides near the approach to the decking level and also at the pit bottom.

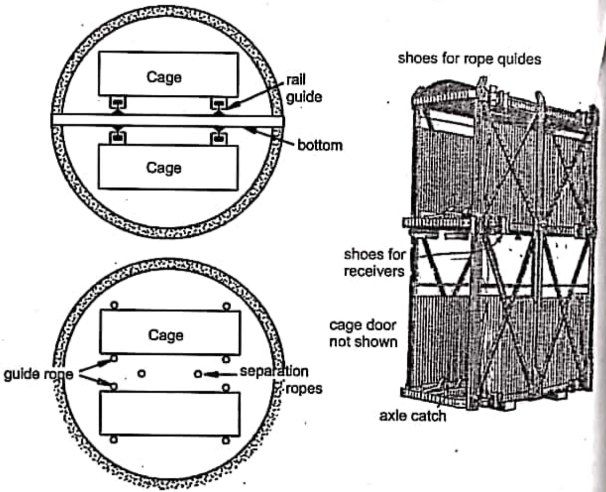


Fig. 11.14 A two-deck cage.

The recommended life of rope guides is 10 years. If the wires of the guide ropes are thick enough, they can be used for making dog nails after the guide rope is discarded.

Winding ropes and factor of safety :

The types of winding rope in use have described in the chapter on wire ropes. A hoisting rope under load is subjected mainly to the following stresses :

i. Stresses due to load being raised.

ii. Stresses due to bending of rope round the sheves and winding drum.

iii. Stresses due to sudden starting and stopping.

iv. Stresses due to intial slakness in rope.

The stresses due to load being raised are because of the static load which consists of :

1. Cage or skip and its contents.
2. Rope attachements like suspension chains, distribution plate, safety hook, and rope capel.
3. Weight of rope suspended in the shaft when the cage is at its lowest point but not resting on the pit botom.
4. Weight of balance rope, if any.

The *static* factor of safety of a wire rope

$$= \frac{\text{ultimate nominal breaking strength of rope}}{\text{maximum static load}}$$

The following factors influence the factor of safety of a winding rope :

1. Depth of wind : A high factor of safety, nearly 10, is difficult to adopt in a deep shaft as this would require a rope of large diam. increasing its weight . Fortunately, the capacity of a rope to resist kinetic shock is roughly proportional to its length and therefore a factor of safety, somewhat less than 10, is pemissible in a deep shaft.

2. Accelerating force : This force is dependent upon the mass that has to be accelerated and the rate of acceleration. As the acceleration is Increased the tension in the rope is also increased, thereby reducing the factor of safety. A high rate of acceleration is undesirable from this consideration. Magnitude of the additiomal accelerating force on the rope can be determined from the relationship, Force = mass × acceleration.

Sudden stopping of the hoist drum when the load is being lowered Induces a stress $F = 2f_{st}$; (f_{st} is the static stress equal to that when starting without a shock).

3. Type of construction of rope : For a given diameter of rope a locked coil rope is stronger than a stranded rope with fibre core. It may, therefore, be sometimes necessary to use a locked coil rope to avoid a large diameter stranded rope if existing headgear pulleys, winding drums, etc. have to be used for a deepended shaft.

4. Conditions under which rope : is used and the period of use : A winding rope which has been used for some time has obviously a lower factor of safety than a newly installed rope. A rope in an upcast shaft which normally has higher moisture content and higher temperature compared to a. d. c. shaft has a low factor of safety.

5. Bending of ropes : Bending of winding ropes lowers their strength and encourages fatigue. This reduces the factor of safety and bending on the rope should therefore be minimum. Drums and pulleys on smaller installations are of about 60 times the rope diameter but for mine winders the ratio is nearly 100. The tower mounted koepe winder sheave which is subjected to less bends has an advantage, from this point of view, over the ground mounted koepe winder.

6. Man winding or material winding :- A rope used for man winding should have more factor of safety than in a rope used for material winding.

7. A winding rope is subjected to shock loads.

i. When the ascending cage strikes the gates or the air lock devices at the end of a wind.

ii. When the ascending cage strikes rigid guides or receivers having faulty alignment.

iii. If the cage suspension chains are slack and the cage is set into motion.

If keps of the ordinary type are used at the pit top, the winding engineman has to lift the pit top cage at the start of the wind so that the keps can be withdrawn. If the pit bottom cage is resting on joists of pit bottom landing, the cage chains become slack and at the commencement of the wind they experience a jerk. This transmits a shock stress to the winding rope. The end of the rope entering the capel suffers the most severe shock.

For ropes of drum winders, used for men winding, the factor of safety is usually 10, but for friction winders it is between 6.5 and 8. A lower safety factor is used for friction winders as the skip/cage must hang on the rope all the time since no keps are provided, and no shock stress, as described above, transmitted to the winding rope at that start of the wind.

Under the Coal Mining Regulations, the factor of safety should be atleast 10 for a winding rope though the D.G.M.S. can give relaxation in deserving cases. The winding rope should be replaced after three and a half years of use.

Recapping of winding ropes :

The winding rope is subjected to wear and stresses at the following points in particular :

i. near the rope capel. Oscillations of the rope are damped down at this point.

ii. rope resting on the headgear pulley when cage/skip is at the pit top and pit bottom. This stretch of the rope is subjected to bending stress and the strands tend to open out permitting moisture or water to enter the core of the rope.

iii. points on the rope where they leave the winding drum when the cage is at the pit top or pit bottom.

iv. Point of the rope which leaves the drum surface to enter inside of the drum for attachment to the drum shaft.

These points of stresses have to be shifted periodically for safe working and with this object the rope near the capel, minimum 2 m long, has to be cut once at least every six months and the rope recapped. Such recapping is required after every overwinding incident also. The extra rope for such recappings is provided for whenever a new rope is first installed and the extra length is kept coiled on the drum shaft in reserve. Splicing of winding rope is not permitted. The Regulations require that there shall be at least two turns of the winding rope on the drum when the cage/skip is at its lowest working point in the shaft. This rope length for minimum two dead turns is also to be provided for when a new rope is installed. Normally 5-6 turns are provided when a new rope is fitted.

A winding rope, after its first installation and after every recapping, is required to make at least 5 trips up and down the shaft with loaded before using it for man winding .

The legal requirement of recapping once every six months is not necessary in the case of koepe widers.

The maximum life of a winding rope under the Regulations is 3½ years, but the Director of Mines Safety may give an extension or restrict the life.

For recapping a rope, the cage is first made to rest on 2-3 girders placed across the shaft top and sufficient slack rope is provided from the spare coils on the winding drum to enable a length of rope to be cut off. The rope capel is released from the cage and drawn to one side. The recapping process then follows the procedure described earlier for fitting a capel.

Method of automatically tipping skips in an inclined shaft :

There are a number of different ways by which a skip in an inclined shaft can be made to tip automatically when it reaches the ore-bin. One of the most effective methods is as follows.

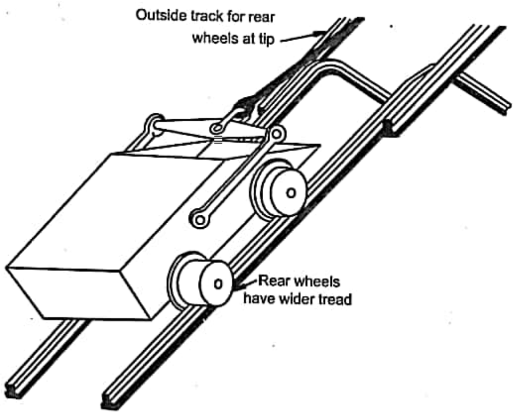


Fig. 11.15 A skip for automatic tipping in as inclined shaft at Mosabani

The main track is bent inwards towards the footwall, and the front wheels of the skip conform to this bend, at the same time, the rear portion of the skip must be guided so that it proceeds straight up the shaft. In the example shown, this is brought about by fitting wheels at the back which have a wider tread, and these engage a special track of wider tread, and these engage a special track of wider gauage outside the main track. As the skip ascends, the back wheels continue on the outer rails up shaft and the skip tips when its front wheels leave the main track.

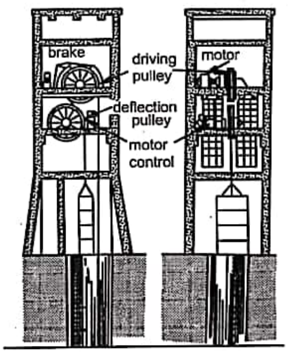


Fig. 11.16 Tower- mounted Koepe winder

QUESTIONS

1. Compare cage winding with skip winding.
2. Describe a safety detaching hook stating how it functions in case of an over-wind. How should the cage be recommissioned after an overwind ?
3. Describe :
   (a) safety catches
   (b) safety devices for a descending cage if it overshoots the pit-bottom decking level.
4. Compare the advantages and disadvantages of rigid guides with flexible guides.
5. With a sketch describe in brief the manner of fixing rope guides in a shaft and keeping them in proper tension. What should be the weight on a rope guide in a shaft 300 m deep ?

CHAPTER – 12

WINDING : DRUM WINDERS & FRICTION WINDERS

Winding systems are classified in the following two main groups, based on the device employed to hoist the cage/skip to the surface.

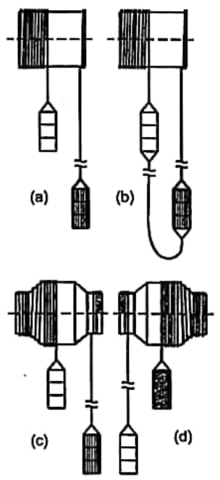
(A) Drum winding, employing cylindrical drum or a variation of it, the bi-cylindro-conical drum.

(B) Friction winding, also called Koepe winding, named after its inventor, Fredrick Koepe, employing a friction pulley.

DRUM WINDING :

Drum winding employing a cylindrical drum is the earliest and simplest system of winding adopted in mines. One end of the winding rope is secured to the hoisting drum and from the other end the conveyance 9cage, skip or bucket) is suspended. Practically all the mine winders are balanced winders in that the drum accommodates two ropes, one for the hoisting cage and the other for the down-going cage, and the travel of the two cages is simultaneous - loaded cage coming up and the empty cage going down. (Fig.12.1). The drum in most of the cases is rigidly keyed to the drum shaft though drums with provision of clutch are used in some cases With cylindrical drums both cages travel at the same rope speed. If the shaft is of limited cross-section and incapable of accommadating two cages, one of the cages is replaced by a counterbalancing weight and the system then resembles the lift sin tall buildings in cities. The payload in that case is hoisted once in every two winds. The position of the winding drum , connected to the winding engine through reduction gearing, is at or near the ground level. To balance the torques on the winding drum due to the loaded cage (to be hoisted) and due to the empty cage (to be lowered), as far as practicable, a balancing rope or tail rope may sometimes be used. A tail rope is of the same diameter as the hoisting rope, has the same length as the depth of the shaft, and is attached to the bottoms of the two cages. It simply hangs in the shaft and is not guided by any "return" pulley. Though it involves an extra cost and extra load that has to be set in motion at the start of a wind, the tail rope is an advantage in deep shafts

to reduce the cost of power in winding as will be clear from the solved examples in the following pages in this chapter. The tail rope also results in better control of the winding engine and in smoother winding.



Another method of nearly balancing the torques on the winding engine is the use of bi-cylindro-conical drum which is sometimes preferred for deep shaft winding. A tail rope is not employed when such drum is used. The bi-cylindro-conical drum (fig. 12.1) is larger than the cylindrical drum for the same duty and is so constructed that there are 3 drums on the same shaft ; one large diameter drum in the middle and two small diameter drums of equal diameter on either side rigidly connected by a conical surface with spiral grooves for the winding rope. The loaded cage is pulled by the rope that coils on the small drum in the beginning of the wind, then on the spiralling groove of the conical surface, and finally (for nearly 60 % of the duty cycle) on the large middle drum. During the same wind the empty cage going down the shaft, uncoils its rope from the large diameter drum in the middle, and the rope, after following a spiral path on the conical surface, uncoils itself from the small diameter drum during the last stage of the wind.

Fig. 12.1 Different systems of drum winding.

(a) Balanced cylinder ical drum without tail rope,

(b) Balanced cylindrical drum with tail rope,

(c) bi-cylindro-conical drum ; start of wind.

(d) bi-cylindro-conical drum; end of wind.

Power and torque diagrams for winding engines :

Winding of men/mineral is essentially a question of lifting men/mineral through a given distance in a given time. To find out the torque on the winding drum shaft and to calculate the power of the motor/engine is a a problem of simple mechanics keeping in mind a few assumptions based on experience.

When a winding engine starts from rest, there is

(a) an acceleration period during which the winder reaches the maximum speed.

(b) an acceleration period during which the revolutions of the winder are constant.

(c) a retardation period during which the winder speed is brought down from the maximum to zero, bringing the system to rest.

The above three periods constitute the winding cycle of a winder

The duty cycle of a winder comprises the above three periods plus a decking period during which loaded mine tubs are removed from the cage at the surface and empty mine tubs are pushed inside the cage for sending them down. The duty cycle of a winder is shown on a graph by showing its angular velocity or torque on the Y-axis and the time on X-axis.

The power rating of a winding speed.The torque required to raise the load and the winding speed. The torques required during various stages of the winding cycle may be considered under four heads as follows :

(a) torque due to cages, tubs and mineral

(b) torque due to unbalanced rope in the shaft

(c) torque due to frictional force

(d) torque due to forces causing acceleration and retardation.

The first three of these constitute the *static torque* and the last one constitutes the *dynamic torque*. The dynamic torque consists of : (i) torque due to the force causing acceleration or retardation of masses moving in a linear direction such as the two cages, their contents and the two ropes (ii) the torque required to accelerate or retard the rotating masses such as the winding drum and the 2 headgear pulleys. The calculations to find the torque may be understood from the following example.

Example :

A winding engine hoists per wind 3 tef of pay-load of copper ore in 2 mine cars up a vertical shaft, 600 m deep. As the loaded cage comes up, the empty cage with 2 mine cars goes down. The caged used has 2 decks, each deck accommodating 1 mine car of tare 0.75 tef. Weight of the cage, the cage chains and suspension gear is 5 tef. The duty cycle consists of acceleration 10 sec; constant speed 30 sec; deceleration 10 sec; decking period 10 sec. The winding rope weighs 5.59 kgf/m length. Length of the rope from top cage to

drum when decking is 36 m and there is also 40 m of dead rope always on each side of the drum. The headgear pulleys are 4.2 m diameter and each weighs 2 tef. Calculate the torques at different stages of winding.

(A) for cylindrical drum without tail rope, and

(B) for cylindrical drum with tail rope

Answer :

(A) *Torque diagram for cylindrical drum without tail rope.*

From the tables, a stranded rope weighing 5.59 kgf/m length is of 38 mm diameter. Drum diameter is nearly 120×rope diameter to avoid bending stress. Let us assume a winding drum of 4.4m diameter weighing 30 tef.

i. If V m/s is the speed during constant speed period the distance travelled by cages during constant speedperiod is 30 Vm. Considering acceleration and deceleration as uniform, during acceleration period distance travelled is $\frac{10V}{2}$ m and during deceleration period also it is $\frac{10\,V}{2}$ m.

$$\therefore \quad \frac{10\,v}{2} + 30V + \frac{10\,V}{2} = 600\text{ m}; \text{or } V = 15\text{ m/s}.$$

ii. Torque due to cages, mine cars and mineral is constant throughout the wind. It is = unbalanced load ×drum radius. It is therefore

$$3 \times 1000 \times 9.81 \times 2.2 = 64746\text{ Nm} = 64.746\text{ kNm}$$

iii. Torque due to unbalanced rope varies throughout the wind. At the start of the wind it is

$$5.59 \times 9.81 \times 600 \times 2.2 = \text{Nm} = 72.386\text{ kNm}$$

iv. During acceleration travel of rope is 75 m. Loaded cage rope of equivalent length is unwound on the empty side. The unbalanced rope at the end of 10 sec. is therefore 600 - 150 = 450 m and the corresponding torque is :

$$450 \times 5.59 \times 9.81 \times 2.2 = 54290\text{ Nm} = 54.290\text{ kNm}$$

v. At midwind the ropes will balance each other and the torque will be zero. After that, length of ascending rope is smaller than the descending rope and there will be a negative torque gradually increasing in value. At the beginning of retardation period the torque will be :

$450 \times 5.59 \times 9.81 \times 2.2 = -54.290$ kNm

and at the end of the wind it will be

$5.59 \times 9.81 \times 600 \times 2.2 = -72.386$ kNm

vi. Friction torque, based on experience, is taken to be equal to that exerted by a force which is $\frac{1}{16}$th the weight of the 2 cages and their contents, acting at the drum radius. This will therefore be

$$\frac{(5+5+1.5+1.5+3)\,1000\times 9.81}{16}\times 2.2\,Nm$$

$=21582$ Nm $= 21.582$ kNm

vii. The force required to accelerate the moving masses can be divided into 2 parts : (a) force required to accelerate masses moving in linear direction, viz. the 2 cages and their contents and the 2 ropes (b) force required to accelerate the rotating masses, viz. the winding drum and the headgear pulleys.

The linear acceleration of the travelling load is $\frac{15\text{ m}}{10}$

$= 1.5$ m/s^2 Force for acceleration in linear direction is given by Force = Mass $\times$ acceleration. This force

$= [\,(10+3+3)\,1000 + (2 \times 672 \times 5.59)\,] \times 1.5$ N

$= (16000 + 7513) \times 1.5$ N

$= 35270$ N

and the corresponding torque is

$= 35270 \times 2.2 = 77594$ Nm $= 77.594$ kNm

viii. A similar torque of 77.594 kNm will be required for retardation but will of course be negative.

ix. Torque necessary to accelerate the rotating masses

= moment of inertia $\times$ angular acceleration

= mass $\times K^2 \times \alpha$

where K is radius of gyration and α is angular acceleration. Full speed angular velocity of the drum is given by

$$\omega = \frac{V}{r} = \frac{15\text{ m}}{2.2} = 6.82\text{ rad/s}$$

$$\text{angular acceleration} = \frac{6.82}{10} = 0.682\text{ rad/s}^2.$$

Normally radius of gyration is taken as 0.8 of the full radius of cylindrical drums. Here we shall use this figure for radius of gyration of the drum and the headgear pulleys.

Torque required to accelerate the winding drum neglacting weight of dead rope is :

$30 \times 1000 \times (.8 \times 2.2)^2 \times 0.682 = 63385$ Nm $= 63.385$ kNm

Torque required to accelerate the two headgear pulleys is :

$(2 + 2) \times 1000 \times (.8 \times 2.1)^2 \times 0.682 = 7698.4$ Nm.

$= 7.698.4$ kNm

Total dynamic torque $= 63.385 + 7.698 = 71.083$kNm.

Having calculated the torques (in kNm) at different stages in the winding cycle of the winder, they are now tabulated as shown in the following table and a graph is drawn. (Fig. 12.2)

Time, seconds from start of wind	0	10	10	40	40	50
Stage	acceleration		constant speed		retardation	
Load Torque	64.75	64.75	64.75	64.75	64.75	64.75.
Rope Torque	72.39	54.29	54.29	54.29	–54.29	–72.39
Friction Torque	21.58	21.58	21.58	21.58	21.58	21.58
Travelling Torque	77.59	77.59	0	0	–77.59	–77.59
Rotating Torque	71.08	71.08	0	0	–71.08	–71.08
Net Torque	307.39	289.29	140.62	32.04	–116.63	–134.73

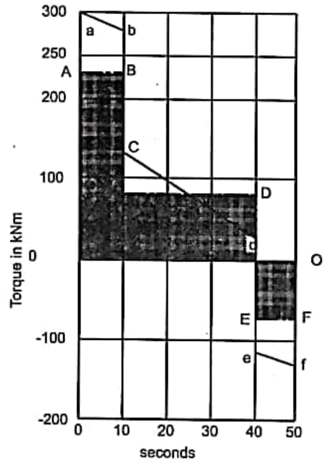


Fig. 12.2 Torque time diagram for balanced cylindrical drum winder.
abcdef-cylindrical drum without tail rope.
ABCDEF- do with tail rope.

(B) Torque-time diagram for cylindrical drum with tail rope :

If a tail rope having the same diameter as the main hoisting rope is used and the conditions and data remain the same as before, the load torque will remain the same as before with a value of 64.75 kNm throughout the wind but the torque required to accelerate the travelling masses in linear direction will be increased by that required to accelerate the extra mass of tail rope. The weight of the tail rope will be 600 × 5.59 =3354 kgf =32902 N and the extra force required to accelerate this extra travelling mass of tail rope =3354 × 1.5 = 5031 N. The torque on the drum shaft due to the extra force is 5031 N × 2.2 m = 11068 Nm *i.e.* 11.07 kNm.

The torque for the rotating masses remains unchanged and the friction torque is assumed to remain unchanged. The earlier table prepared for the cylindrical drum (without tail rope) can now be recast to incorporate the additional torque for tail rope during acceleration and the extra negative torque for tail rope during retardation.

Time second from start of wind	0	10	10	40	40	50
stage	acceleration		constant speed		retardation	
Load torque	64.75	64.75	64.75	64.75	64.75	64.75
Rope torque	0	0	0	0	0	0
Friction torque	21.58	21.58	21.58	21.58	21.58	21.58
Travelling torque	88.81	88.81	0	0	– 88.81	–88.81
Rotating torque	71.08	71.08	0	0	– 71.08	–71.08
Net torque	246.22	246.22	86.33	86.33	– 73.56	–73.56

The resultant torque diagram is given in fig. 12.2. A comparison of the two torque diagrams indicates that the effect of the tail rope is to reduce the starting torque required from 307.39 kNm to 246.22 kNm, a saving of almost 20%. The negative torque at the end of the wind is also subsantially reduced, calling for comparatively less breaking effort.

Torque-time diagram for a bi-cylindro-conical drum hoist

A bi-cylindro-conical drum is sometimes employed for hoisting in deep mines as it reduces the net starting torque and the negative torque at the end of the wind appreciably as compared to the cylindrical drum.

In the above solved example, if we consider the cylindrical drum replaced by a bi-cylindro-conical drum having large drum 6 m diameter and small drum 3 m diameter, other data and winding cycle period remaining the same, the torque at various stages can be calculated and the torque-time diagram and the winding cycle diagram will be as shown in Fig. 12.4.

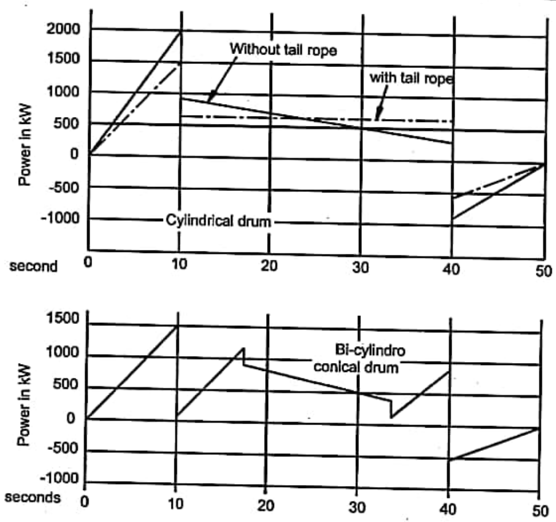


Fig 12.3 power-time diagram
(top) for balanced cylindrical drum. (bottom) for bi-cylindro-conical drum.

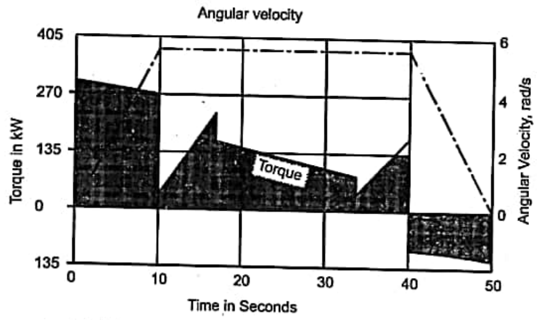


Fig. 12.4 Torque-time diagram and angular velocity-time diagram for a bi-cylindrc-conical drum.

The motor size for the winder is determined by the root mean square torque. As the current is more or less proportional to the torque, for all speeds the r.m.s. torque will be proportional to the mean heating effect of the motor currents.

$$\text{R.M.S. torque (Nm)} = \sqrt{\left[\frac{\Sigma\frac{t}{3}\left(T_1^2+T_1T_2+T_2^2\right)}{\frac{2}{3}(t_a+t_r)+t_c+\frac{1}{3}t_d}\right]}$$

where T_1 is net torque (in Nm) at zero seconds (starting).

T_2 is net torque (in Nm) after t seconds, the time of duty cycle

t_a is acceleration period, seconds,

t_c is constant speed period, seconds,

t_r is retardation period,

t_d is decking period, *i.e.*, for loading and unloading.

t is duty cycle period,seconds.

In the above formula of R.M.S. torque, the value

$\frac{2}{3}(t_a + t_r) + t_c + \frac{1}{3}t_d$ is called equivalent time t_e.

The r.m.s. torque is used to calculate the equivalent motor power, using the maximum speed of the motor. The power so calculated is the r.m.s. value 5 % is added to cover possible discrepancies between motor design and performance.

Example :

If the r. m. s. torque for a winder is 109 kNm, the winding drum (cylindrical) diameter is 4.4 m and the maximum rope speed is 7.0 m/s, calculate the power of the motor.

Answer :

$$\text{Power of motor, kW} = \frac{109\text{ kNm} \times 7\text{m/s}}{2.2\text{ m}}$$

$= 346.8$ kW

add 5 % ; Total motor power is, therefore, 364 kW.

Conclusion from the torque power diagram :

1. If a cylindrical drum without tail rope is used the peak demand of power is large and much in excess of what it would be if a bi-cylindro-conical drum is used.

2. The negative power requirement at the end of a wind is also large in the case of cylindrical drum without tail rope. If electrical winder with regenerative breaking is used, most of the negative power can be fed back to the electric supply system if Ward-Leonard system is employed. But if a steam winder is used the large negative torque calls for heavy application of breaks, or alternatively, application of steam pressure against the motion resulting in increased steam consumption.

3. The diagram highlights an important advantage of electrical winding. An electric motor can withstand momentarily a load nearly 100% over and above its normal power. If the peak power demand (e.g. at the end of acceleration period) is 2,000 kW a motor of only 1,000 or 1,200 kW can be used for winding . In the case of a steam engine however the steam engine should be capable of developing the peak demand. Not only this if atwin cylindrical engine is used, each cylinder should be capable of producing peak requirement of power as one of the cylinders may be having its crank at the dead centre.

FRICTION WINDING (KOEPE WINDING)

The friction winder, which is also called Koepe winder after its inventor, Fredrick, Koepe, consists of a steam or electrically driven sheave fitted with renewable friction lining which is grooved to suit the main winding rope whose arc of contact varies between 185° and 230° according to the design of the winder. A cage or skip is suspended at each end of the rope which is not secured to the sheave. A tail rope whose ends are attached to the underside of the cages is always used in the Koepe system of winding. The tail rope is arranged to hang freely in the shaft below the lowest winding level. The winding rope along with the attached cages is raised or lowered by power transmitted through the friction between the winding rope and lining of the sheave. Hence the name friction winding. The cofficient of friction between the sheave lining and the rope is a minimum of0.2. The static torque is constant as all the weights other than the useful payload are balanced and therefore the power required approaches the practical minimum and varies only slightly as the winding levels are deepended.

The basic principal underlying the working of a friction winder is the fact that a belt or hoist rope passing over a wheel driven by a prime mover, has different tensions at points where the belt or rope enters the wheel and where it leaves the wheel. The relationship of these tensions is given by the equation.

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

Where,

T_1 = tension in the rope entering the wheel i. e., the rope being hoisted.

T_2 = tension in the rope leaving the wheel.

μ = cofficient of friction between tread material on the wheel and the rope, the value ranging between 0.35 and 0.5, depending upon materials.

θ = angle of wrap of rope around friction wheel in radians.

e = base of Napierean logarithm.

The rope may slip over the wheel when the ratio $T_1 : T_2$ exceeds a value 1.5 to 1.6. In general the lower the $T_1 : T_2$ ratio, less is the possibility of rope slippage over the friction wheel. Also, the lower the $T_1 : T_2$ ratio, less the out-of-balance torque the motor has to drive against.

Unlike in drum winding the friction winder may employ 2, 4 or more ropes but the tail rope is only one. A tail rope is generally of such a size that it has the same weight per metre length as the weight per metre length of the combined number of main ropes; for example, in a four-rope friction hoist using four 2.5 cm dia flattened strand main ropes of 2.8 kgf/m, the correct tail rope would weigh approximately 4×2.8 = 11.2 kgf/m.

As the friction between the winding rope and the wood of the tread has to be high for effective working of the koepe system, the locked coil rope, because of its smooth surface, was long considered to be unsuitable, and there was a general tendency to use stranded ropes. The flattened strand rope appeared to be much favoured. Tests have shown that the cofficient of friction of locked coil rope with oak wood is 0.388 and of stranded Lang's lay rope of the same diameter with oak wood is 0.336. The prejudice against locked coil rope is therefore disappearing and there are a number of installations in Britain and other European countries where locked coil rope is used for koepe winders. The high capicity factor of locked coil rope is another reason for its preference. The wear of the wooden tread is much less due to the plain and smooth surface of locked coil rope.

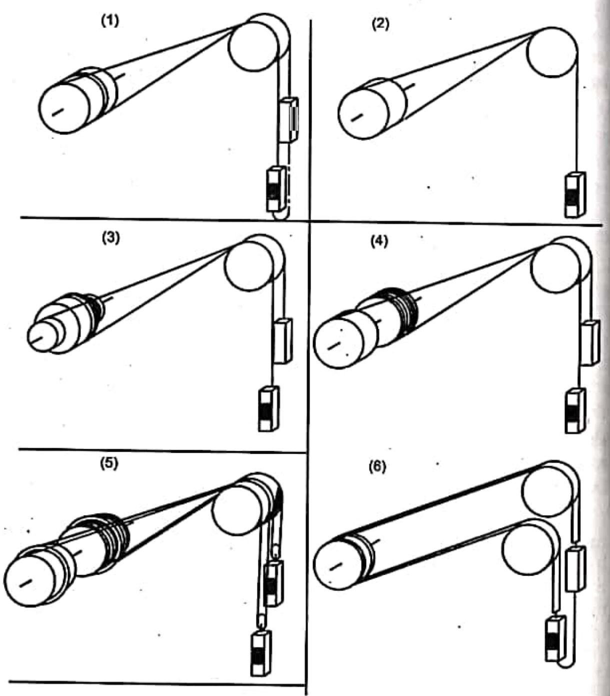


Fig. 12.5

1. *Single drum winder ; double cages/skips*
2. *Single drum winder ; single cage/skip*
3. *Bi-cylindro-conical drum winder*
4. *Double drum winder*
5. *Double drum winder ; Blair type*
6. *Two-rope friction winder winder ;ground mounted*

There are two types of Koepe winders :

1. *the ground Koepe ;* 2. *the tower Koepe ;*

In the ground Koepe the winding engine is installed at the ground level and the head gear sheaves are situated one above the other on the head gear as shown on P. 245. The rope operates in the plane of the Koepe driving wheel without any angle of fleet.

In the tower Koepe the winding engine is erected on a tower over the shaft. This type of Koepe winder possesses many advantages over the ground Koepe, these being 1. a large angle of rope contact on the driving sheave can be obtained, normally 200 degrees to 230 degrees ; 2. the winding rope is protected against adverse weather conditions ; 3. the headgear structure need be no stronger for a given only than some of the head gears that are in existence today for normal drum winding ; 4. it eliminates any obstruction by a winding engine house in the neighbourhood of the pithead. 5. The rope is subjected to less number of bends on a tower mounted Koepe.

Advantages of Koepe winding system (*i.e.* friction winders) over the drum winding system :

1. Koepe system is most suitable for winding heavy payloads from larger depths. In drum winding system the drum has to be of large diameter for deep shafts and multilayer coiling on drum results in reduced rope life on account of crushing and heavy wear
2. Koepe winder is simple for manufacture, compact and lighter than the drum winder. Intial cost is therefore, less for similar duties.
3. Less costly engine foundations are required for koepe winder due to lighter weight and compactness.
4. In Koepe system the inertia of rotating parts that have to be set in motion is less compared to a drum winder. This is however partly offset by greater inertia of ropes. Electric motor of smaller size is required. Less inertia, coupled with the balance tail rope, lowers peak demands on electric supply system. This results in low operating and maintenance cost.
5. There is no fleet angle in koepe system and wear on the winding sheave is reduced.

6. Koepe system lends itself to adoption of multi-rope winding which has important advantages listed later in this chapter.
7. With a drum winder it is not possible to use an existing drum when winding is necessary from greater depths, unless the drum is replaced by a larger diameter drum to accommodate more substantially changed. It is however, possible to replace the drum by a friction sheave and use the existing winding equipment for winding from larger depths.

   It is suitable for horizontal mining system where both cages wind from one level. If loading has to be done from a deeper level on regular basis the change-over is possible by changing the koepe pulley or other winding equipment (assuming that winder motor has adequate power).
8. The skip/cage does not rest on the keps and therefore, to start it, no shock loads are transmitted to the ropes, and a smaller factor of safety, 6 to 7, is adequate.
9. A smaller length of main rope is required compared with drum hoisting using a balance rope, as there are no extra coils.
10. Operating costs are less due to the smaller rated output of motor.

The disadvantages of koepe system of winding are :

1. Winding possible from one level only if the two skips/cages are nearly balanced. For multi-level winding, clutched cylindrical drum winders have a definite advantage. With a Koepe winder multi-level hoisting can be only with a single conveyance (*i.e.* cage/skip) and counterweight. Alternatively, for winding from 2 loading stations in one shaft two independent winding engines are sometimes used.
2. Koepe system can be used for only vertical shafts and not for inclined as guiding and tensioning of balance rope poses problems.
3. Koepe system cannot be used during shaft sinking.
4. Koepe system is not suitable for shallow shaft as the cylindrical drum size, if drum winding is adopted, would be nearly the same as the Koepe sheave size.
5. A deeper shaft sump is required to accommodate the tail rope loop.

6. The rope changing equipment costs more and the balance rope requires heavier suspensions gear and stronger shaft conveyance.
7. Seperate run with one cage is impossible.
8. If the rope breaks, both cages fall in the shaft.

   Ground mounted friction winder ropes aré, however, subjected to effects of climate.

Multirope system of winding :

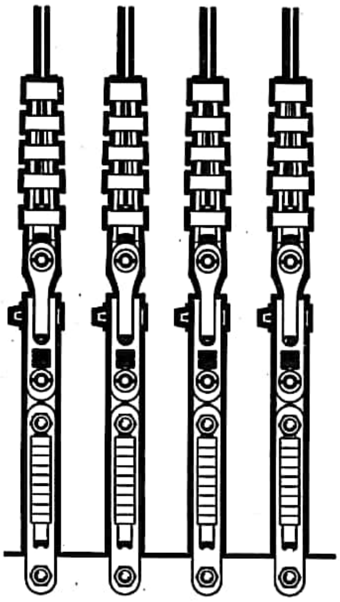
The multirope koepe winder is an improvement over the single rope Koepe winder and is essentially a friction drum has as many parallel grooves as the number of ropes. These grooves are 30 cm apart, centre to centre, and are as deep as the radius of the rope. The tread material, in which the grooves are made, is wood or a kind of plastic and it is attached to the drum plate by countersunk bolts. The number of ropes for the cage is even, usually 2 or4. The reason for these even number of ropes is that adjacent ropes on the drum are of opposite lays i.e. one rope of right hand lay and the other of left hand lay, as such arrangement avoids the rope tendency to untwist. In Russia 8-rope hoists are in use for 50-tenet payload. On the European contient 4-rope friction winders are popular though there are isolated installations of even ten-rope friction winders. In India, Jaduguda mine is equipped with a multi-rope friction winder (2 ropes) with a payload of 5 te in the skip. The tension in all the ropes should be equal, as far as practicable.

Adjustment for variation in rope tension :

In the case of friction winders if two or more ropes are used for a skip, the stretch of all the ropes may not be the same and this results in unequal tension in the ropes. With the object of ensuring equal tension the following arrangements are sometimes adopted.

1. Inserting or removing links of varying length in the suspension unit between the capel and the skip. There is a limit to the number of links which can be included in the suspension unit and recapping is a lengthy process which can usually be carried out only at the week-end. The German type of rope capel (GHH type) provides for rapid adjustment of rope length but it can be used with stranded ropes and not on locked coil ropes.

Balancing of the tensions may be carried out by movement of deflecting sheaves as an alternative to linkage adjustment in the suspension.



Rectangular blocks for tension adjustment

Fig. 12.6

2. Using a series of rectangular blocks which can provide coars or fine adjustment. (Fig. 12.6)

3. Adoption of a linkage arrangement with provision for the insertion of hydraulic cylinder for each rope. By simply inserting a cylinder in the central slots and applying pressure from a portable motorized pump, the rope tensions are automatically balanced rapidly without the necessity for providing clamps to remove the load from the suspension during adjustment. The tension imposed on individual ropes can be measured by insertion of split pressure rings between the rope capel and the skip as shown in Fig. 12.7.

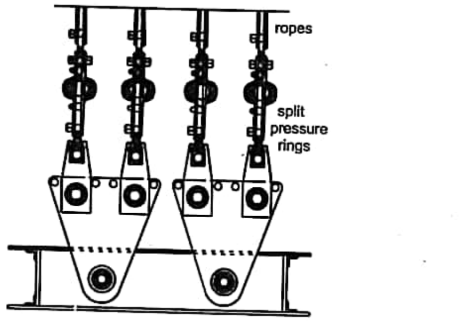


Fig. 12.7

Fig. 12.8 shows a simple link comensating gear where four ropes are attached to a skip. The advantage of using link compensation gear with a single connection to the cage/skip is that the lifting forces are constantly applied in line with the designed centre of gravity of the load.

The multi-rope winders are generally tower mounted and guide sheaves or deflection pulleys are used for centering the ropes over the cages.

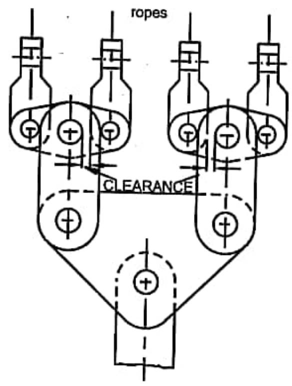


Fig. 12.8 A simple link compensating gear

The advantages of multi-rope winders are :

1. Each rope has to be of small diameter compared to one large diameter rope which is difficult to manufacture and handle; the weight of the single rope itself willbe excessive in deep shafts and the safe payload it will be small. Small diameter ropes require friction drums of small diameter, e.g,, the diameter of the friction drum obtained with a 4-rope hoist is half the diameter for an equivalent single rope system. (See Blair winder later)
2. It results in better safety. If one rope developes fault, the other ropes can share the load
3. The reduction in size and weight of the friction drum results in
   (a) reduction in the size and weight of the hoist.
   (b) reduction in static frictional torque and inertia of the rotating parts and, hence, in motor power.
   (c) drum speed can be increased.
4. There is saving in space at the pit top.
5. The capital cost of the installation is less.
6. When the diameter of the friction drum is equal to the distance between centres of the hoisting cage/skip, no deflection sheaves will be required.
7. The ropes are protected from the atmosphere as multi-rope winders are generally tower mounted and the entire installation is accommodated in a tall building, equipped with a lift for access to the friction pulley/drum.

Having grasped the working of friction winders, we can now work out the torques at various stages in the winding process by a friction winder.

Example :

Calculate the torques at different stages for a tower mounted friction winder with the following data.

Loaded skip weight – 8 tef
Empty skip weight – 4.5 tef
Rope weight – 5.78 kgf/m length
Friction drum diameter – 2 m
Acceleration time, ta – 16s
Constant speed time, tc – 30s
Retardation time tr – 10s

Decking time, td – 15s
Maximum rope speed – 8.15 m/s
Shaft depth – 350m
Tower height – 30m
Bottom rope loop – 10m

Moment of interia of the friction pulley and the motor geared to it, referred to the axis of friction pulley, is 24 tonne-m^2.

Assume static torque due to friction 0.08 times the torque due to loaded skip plus empty skip.

Answer :

The total torque consists of (A) static torque and
(B) dynamic torque

(A) ***The static torque.*** It is due to

i. the unbalanced load, and
ii. due to friction

i. static torque due to to unbalanced load
$= (8\text{-}4.5)$ tef $\times 1000 \times 9.81 \times 1$ m Nm
$= 34.3$ kNm

ii. static torque due to friction
$= 0.08\,(8+4.5)$ tef $\times 1000 \times 9.81 \times 1$ m Nm$=9.81$ kNm

$\therefore$ total static torque $= 34.3+9.81= 44.11$ kNm

Since length of rope on the empty side and load side is equal there is no unbalanced rope and no rope torque.

(B) *Dynamic torque :*

The total equivalent inertia is due to following masses

Loaded skip $8\times1000 = 8000$ kg.
Empty skip $4.5 \times1000 = 4500$ kg.
total rope length is $390\text{ m}\times2 = 780$ m
mass of the total rope length is $780\times 5.78 = 4508$ kg.
Mass of the friction wheel, rotating parts of motor, gearing, etc.

$$\text{is} = \frac{24\text{ tem}^2}{(1\text{ m})^2} = 24000\text{ kg.}$$

The total masses (kg) for dynamic torque = 8000
+ 4500
+ 4508
+24000
41008

T_a, dynamic torque during acceleration

$$= 41008\ kg \times \frac{8.15 m/s}{16 s} \times 1\ m\ Nm$$

= 20.88 kNm

T_r, dynamic torque during retardatio

$$= 41008\ \text{kg} \times \frac{8.15\text{m}/\text{s} \times 1\text{m Nm}}{10\text{s}}$$

= 33.42 kNm

The torque in kNm can now be tabulated as follows :

	T_a	T_c	T_r
Static torque	44.11	44.11	44.11
Dynamic torque	20.88	0	–33.42
Total	64.99	44.11	10.69

These are depicted in Fig. 12.9.

It may be noted that there is no torque variation in each period of acceleration, constant speed period or retardation

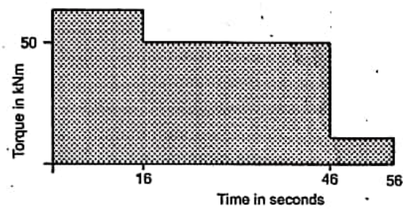


Fig.12.9 Torque-time diagram for a koepe winder.

Blair winder

It is impractical to wind substantial payloads from extreme depths using a single rope attached to each cage/skip due to the size and weight of the rope required. The Blair winder is a special form of double drum winder which overcomes this problem by suspending each cage/skip from two ropes, thus minimising both the rope and drum diameters. This type of winder is used for very deep mines.

All drum winders incur high rope coiling loads on the drums. The effects of single layer coiling on drum designs are well understood but the effects of multilayer coiling are more complex, necessiating careful consideration of drum design and strength based on experimental data.

QUESTIONS

1. What are the different methods of balancing the torque on the winding drum.

   Describe the bi-cylindro-conical drum for winding and state how it helps in balancing the torque.

2. What is meant by R.M.S. torque and state its significance for calculating power of a winder motor. Give a formula for the R.M.S. torque.

3. Compare drum winding with friction winding.

4. State the advantages and disadvantages of multi-rope friction winder. What methods are adopted to equalise the tensions on the winding ropes when multi-rope winding is employed ?

CHAPTER – 13

WINDING : STEAM & ELECTRIC WINDERS, SPEED CONTROL & SAFETY DEVICES

In a number of coal as well as metal mines the winding engines are powered by steam though there has been an increasing tendency during the last 4 decades, to install electrically powered winding engines due to the establishment of high capacity central thermal power stations.

Steam operated Vs. electrically operated winders :

1. Initial cost of steam raising plant is high for drum winders designed for same duties. The steam plant is going out of date in most of the mines.
2. The steam winder should be capable of providing the maximum power from one steam cylinder at the start of the wind. The cylinder and the engine, therefore, have to be fairly large. In the case of electric winder the peak power during acceleration period can be taken by the motor as temporary overload, so that the motor has to be selected as one dealing with the average power required during the winding.
3. Operational and maintenance cost of steam plant are higher. Extra work has to be provided for periodical cleaning, repairs, etc. Electrical winder demands less attention to maintenance. Where only occassional winding is necessary and full shift duty is not required of the winder, electric winder is advantageous as fuel standby losses are considerably reduced.
4. Electric winders provide uniform torque compared to the pulsations of reciprocating steam engine. During acceleration and retardation periods the rate of change of torque is smooth and uniform with electric winders. This results in smoother winding and less wear on the rope, guides and other winding gear. The uniform torque is specially advantageous in the case of Koepe winding to prevent rope slip.Where a cylindrical drum winder is used a balance rope for equalisation is not as satisfactory with a steam engine as with an electric motor. The cyclic variations in torque during each revolution

tend to set up a pulsation in drum speed which may cause serious trouble if it should synchronise with the natural frequency of the balance rope. In any case the pulsations in torque cause the rope to oscillate.

5. Higher winding speed is possible with an electrical winder and therefore, larger output can be handled.
6. Electric winder permits installations of more safety devices to deal with over-winding, excessive speed, *etc.*
7. Speed control of electric winder is smooth and more precise.
8. Regenerative breaking or some other form of electrical breaking is possible on an electrical drive and this results in great saving of wear and tear on the mechanical breaks.
9. The electrical winding engine room is cool, compact and clean as also the surroundings as there is no problem of stocking coal for boilers or of ash disposal.
10. Efficiency of electric motors is more than of steam engines and electric power is used with much more efficiency for winders than in the case of steam power where cut off is at 0.9 of the stroke during the acceleration period and 0.25 to 0.3 of the stroke during the constant speed period.
11. A steam cylinder, however simple and excellent, is a more difficult machine to control and to run on an intermittent duty like a control than that required for an electric winder. Unlike a steam engine,an electric motor can be furnished with mechanisms which simplify the work of the driver at the begining and the end of a wind and which makes the abuse of the machine impossible.

One major advantage of steam winding is its non-dependence on an external power source and the consequent reliability. In the coal mines free and easy supply of fuel is another advantage. With the grid pattern of electric supply from large capacity electric power houses these days, electric power is quite reliable. Some mine owners, however, may prefer to have a diesel engine set as a standby power supply in case of failure of electricity.

Winder motors in most of the cases operate at 3.3 kV. For A.C. 3-phase, 50 cycle electric supply they normally do not have more than 16 poles. The synchronous speed of an electric motor for 3 phase, 50 cycle electric supply is given by the formula

$$\frac{3000}{\text{or pairs of poles}} = \text{r.p.m.}$$

and with a 16 pole motor the synchronous speed is 375 rpm. With a single speed reducing gear of 1 : 8, the shaft of a winding drum will have a speed of normally 45 rpm which is usually the maximum speed of winder drum-shaft.

As the efficiency and power factor of an induction motor decreases with an increasing number of poles, there are power losses if a slow speed motor with large number of poles is selected for winding. For this reason a winding motor usually has less than 16 poles (generally 8, so that the synchronus speed *is* 750 rpm) and it is of slip ring type.

SPEED CONTROL OF ELECTRIC WINDERS :

Speed of AC winding motors, slip ring type, is varied by regulating resistance in the rotor circut. This is known as rheostatic control. For small motors of upto 100 kW electric resistances in the form of cast grides are employed. The resistances are placed in rotor circuit and their magnitude varied by contractor switches.These resistances have to dissipate large amount of heat when the winder is accelerating and they have to be of large size for high H. P. motors. At the start of the wind the entire resistance is in the rotor circuit and connected to the slip rings; as the winding proceeds the resistances are gradually cut out so that during the constant speed period, after the full acceleration is achieved, the resistances are completely out of the rotor circuit and the slip rings are short circuited. The driver's control lever is arranged to vary the resistances in the rotor circuit. Metallic grid resistances are large, heavy, difficult to cool and smooth control cannot be obtained without using a large number of contactors.

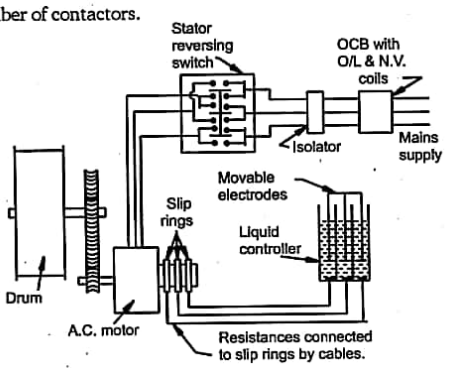


Fig. 13.1 Liquid controller for speed control and torque variation on A. C. winder.

On larger winders the speed control is effected by the use of variable resistance (Fig. 13.1) provided by the liquid controller. It consists of 3 fixed and 3 moving electrodes arranged to work together in pairs, each inside an insulated trough. The 3 moving electrodes are connected together, earthed and coupled by a rod system. The 3 moving electrodes are also connected together by a cable which forms the netural point of the rotor winding. At the base of each trough is the fixed electrode connected through a cable gland to one of the rotor slip rings. As electrodes are raised, variable resistance is introduced in the rotor circuit and the resistance depends upon the depth to which the electrodes are immersed. There is a smooth control of resistance without steps throughout the working range. The electolyte is a solution of sodium carbonate 3 % to 10 % in distilled water. It is cooled by water circulating through pipes immersed in the tank. The resistance of electroyte falls with increase in temperature : 50 % fall may be expected for temp. rise from 27 ° C to 54 ° C. In course of use over a period of a few weeks there is some formation of scale on the electrodes which have to be periodically cleaned, as the scale is of insaulting nature.

The liquid controller can be operated by an oil operated servo mechanism which is so designed that when it is necessary to start the winder against full load the driver moves the operating lever to the full speed position for the required direction, the servo device then automatically moves the electrodes at the pre-set rate. The first part of the movement is made quickly to build up the required accelerating torque, after which the stroke is completed in the designed time period. This arrangement of liquid controller is suitable for use with motors upto 75 kW. For larger motors the horizontal Weit Controller is used in which the electrodes operate horizontally in throughs and the electrolyte is supplied by a circulating pump.

The electric supply to the winder motor is controlled by an oil circuit breaker fitted with

(a) Low volt protection
(b) Over current trip coils and
(c) Earth fault protection.

An air break contactor switch is employed for reversal of direction of motor rotation. To effect reversal, the switch interchange 2 of the 3 stator leads.

Ward Leonard method of speed control :

This method of speed control of winders is adopted 800 kW. The main advantage is that it is economical in power consumption only for large winders usually of above and very sensitive for speed control; and for these reasons it is widely used for large winders during the last 50 years.

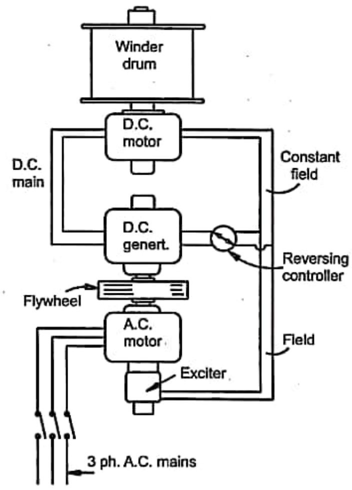


Fig. 13.2 Ward Leonard control of winder.

In this system (Fig. 13.2) the winding engine drum is driven by a D.C. motor either direct coupled or through gearing. The D. C. motor receives its power from a D.C. generator at a voltage which can be varied and also reversed. But the field of the D.C. motor is always constant. The D.C. generator is shunt wound and is driven by a constant speed, 3 ph. A. C. motor (direct coupled) receiving its power from A. C. mains. On the same shaft of A. C. motor is mounted an exciter (small D. C. generator) which

(a) supplies direct current of a constant value to the field circuit of D.C. motor of winder

(b) supplies direct current to the field circuit of D. C. generator. But this current can be varied in magnitude and direction by a reversible controller.

It will, therefore, be seen that the system essentially consists of the following equipments instead of a single winder motor and its controller :

(a) 3 ph A. C. motor directly coupled to

(b) A. D. C. generator (shunt wound), and also to

(c) An exciter which is a compound wound small dynamo,

(d) A DC motor (shunt wound) coupled to the winder drum,either direct or through gearing.

(e) A reversing controller in the field circuit of the large DC generator.

The A. C. motor - D.C. generator set runs continuously during the shift. But the winder motor is operated only as and when winding is to be done.

The e. m. f. generated in a D. C. generator is given by the formula

$$E = \phi \times P \times \frac{Z}{a} \times \frac{N}{60} \times 10^{-8} \text{ volts}$$

Where, E = the e. m. f. generated

ϕ = flux per pole

P = No. of poles

Z = No. of armature conductors

N = speed in r. p. m.

a = the number of parallel paths in the armature. It is equal to two for a wave winding and equal to the number of poles for lap winding.

It can be seen that everything excepting flux remaining the same, the voltage of the D. C. generator is proportional to the flux.

The output of DC generator can be varied from zero to the full voltage by varying its excitation ; and by reversing the position of the controller in the field circuit of the D. C. generator, the direction of rotation of the winder motor can be changed. The voltage of the D. C. generator can, therefore, be varied by controlling the current in its field circuit and this current is small. The reversing controller has, therefore, to deal with only a small current and its position decides the voltage and output supplied to the winder motor and consequently its speed. There are, therefor, eno heavy resistance losses as in other methods of speed control employing an A. C. motor coupled to the winder, but only small electric losses in the field circuit of D.C. generator. The only power wasted is in reversing controller resistance and this amounts less than 5 % of the power taken by the winder motor.

Ward Leonard system requires a heavy intial cost as it requires 4 electrical equipments (1 A. C. motor, 1 D. C. generator, 1 D. C. motor and 1 exciter) instead of a singal A. C. motor of a winder in the normal methods of winding. The space requiired for installation is also large and winders withWard Leonard system can therefore be installed only on the surface and not underground. The motor generator set has to be run all the time during the shift, though the winder and its motor operate interimittently. Inspite of these disadvantages the system is economical in power consumption and gives very fine speed control.

It has to be appreciated that for the D. C. shunt motor of winder the basic formula is

speed $\propto$ to E (approx.)

where flux of the field circuit of motor is constant, and E is the applied voltage.

Ward Leonard Ilnger System :

The Ward Leonard system suffers from the disadvantage that inspite of the inertia of the moving parts sharp current peaks are transmitted to the power supply system causing wide fluctuations of voltage. This defect is remedied to a substantial extent by using slip ring induction motor equipped with a slip regulator for driving motor-generator set and mounting a flywheel at one end. The flywheel stores energy by speeding up during periods of light load and delivers it by slowing down during the heavy load period.

Overspeed preventing and slow banking devices :

Mining regulations require that for every shaft exceeding 100 m depth. automatic devices should be provided (i) to prevent over-speed (ii) to prevent overwinding, and (iii) to ensure "slowbanking" i. e. as the cage approaches pit-top or pit-bottom decking level, its speed should not exceed 1.5 m/s when men are being hoisted or lowered. The majority of the automatic devices or contrivances for overspeed prevention and slow-banking used in this country are of the following makes.

1. Whitmore Controller.
2. Automatic Contrivences and speed recorders
   (Bharat Gold Mines Limited) (See Appendix for details).

The Whitmore Overwind Controller :

Two screwed spindles (Fig. 13.3), one for each direction of wind, are driven through bevel and straight-cut gears from the drum shaft and carry overwind nut B and overspeed nuts C, which move up and down simultaneously with the movement of the cages or skips in the shaft. The overwind nuts have notched fingers normally engaging with a fixed plate D, but near the end of the wind they run on to a spade lever E, which operates the trip lever F by the rotation of the nut, releasing the spring G which cuts off power and applies the breaks.

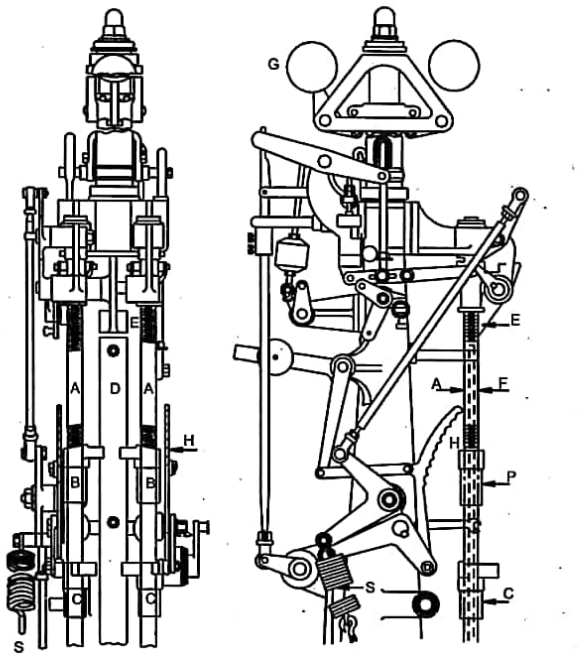


Fig. 13.3 Whitmore automatic controller.

The governor controls the position of a notched quadrant H; if overspeeding takes place the wing on nut C engages with a tooth of the quadrant and through the linkage shown, applies the breake gradually and shuts off the power supply. The speed must be progressively reduced at the end of the wind owing to the shape of the quadrant and overspeed protection continuous for the rest of the wind, when control is once established.

In order to comply with the slow-banking requirement of a landing speed of less than 1.5 m per sec., a hydraulic device is added to take charge near the end of the wind when the governor becomes too insensitive. This works on the dashpot principle in which the speed of descent of the dashpot piston is proportional to the cage speed and, unless the predetermined retardation is adhered to pressure builds up behind an orifice and tripping takes place.

The centrifugal governor has proved, to be too insensitive to cope with a speed variation from a full speed of as high as 20 m per sec, to practically zero. An auxiliary device is generally arranged to come into operation only during the later part of the retardation period when men are being wound and this constitutes the *slow-banking* portion of the automatic appliance.

On steam-driven winders an additional device may be provided by which the amount of steam that can be admitted to the cylinders is limited as the cages approach the end of the wind. Movement of the steam throttle lever in the forward direction is automatically curtailed, but if the control gear is put into reverse, full admission of steam is possible. If the driver, therefore, wishing to retard the engine, inadvertently admits steam to accelerate it, the rate of acceleration is minimized ; this also takes place if the wind is commenced in the wrong direction. Reeief valves may also be provided to release steam trapped between the cylinder and the throttle valve when the latter is closed automatically in emergency.

Mechanical breaks on winders

Mechanical breaks, acting directly on the winding drum, must be provided for all winding engines. For a winder with two cages the breaks must hold the maximum torque of the engine, in either direction, when the loads are in balance. The standard practice is to provide two pairs of break shoes which act directly on brake shoes which act directly on brake paths constructed on either side of the drum. The brake shoes are connected by rods and levers to the operating pedal of the winding engineman. Fig. 13.4 shows a suspended caliper type and Fig. 13.5, an anchored post brake. The anchored post brake, which has been in use for many years, suffers from the disadvantage that the pressure on the brake path varies from the maximum at the top of the

brake blocks to a minimum at the bottom. This causes unequal wear of the lining of the brake blocks. The centre suspended caliper brakeis preferable to the anchoed post brake as it gives a more uniform pressure over the area of brake blocks resulting in uniform wear and greater braking power. The wooden brake blocks are lined with bonded asbestos or fibre such as "Ferrodo" brake lining. The coefficient of friction for bonded asbestos lining is nearly 0.4 but it is reduced by presence of oil or water on the brake path. An adjuster on the tie rods is used to adjust the position of brake blocks relative to the drum as the lining wears in courese of use. Normally the brake blocks grip the drum and hold it when it is stationary or not required to rotate and this is possible by the use of dead weights suspended from the brake lever. For lifting the weights a brake engine, worked by steam, oil pressure or compressed air, is arranged. The brake lever is connected by a system of rods to the brake engine control valve. The control valve must be designed to move to the "brakes on" position if the operating gear becomes detached. In the event of failure of steam, oil or air supply or when an emergency trip occurs, the breaks are applied automatically by the falling weight.

A brake engine for an electric winder is generally operated by oil pressure or compressed air.

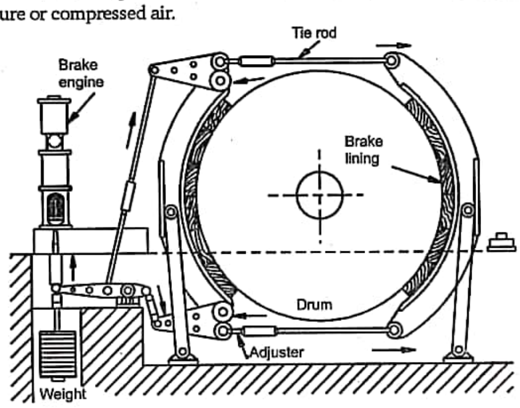


Fig. 13.4. Suspended caliper brake

A refinement of the orthodox weight-loaded type of mechanical brake is the compound spring brake which applies the braking pressure through springs, thus avoiding excessive shocks to the rope and cages due to the momentum of the falling brake weights. This system employs two oil-operated brake engines, one of which controls the spring braking and the other, the raising and lowering of the brake weights, this latter engine being heavily damped. Each brake engine is controlled by its own control valve and these are so connected to the driver's lever that they operate in series. Thus the spring brake operates first and applies almost instantaneously braking torque upto about 50 % of the maximum possible. Further movement of the driver's brake lever then applies the brake weights to augment the breaking force, thereby bringing the winder quickly to rest without stock to the system. For normal manoeuvring the spring brake only is required, but the arrangement permits the weights being held in reserve and they are only brought into action when heavy braking is required, but the arrangement permits the weights being held in reserve and they are only brought into action when heavy braking is required or the drum is to be held stationary after the winder has been brought to rest.

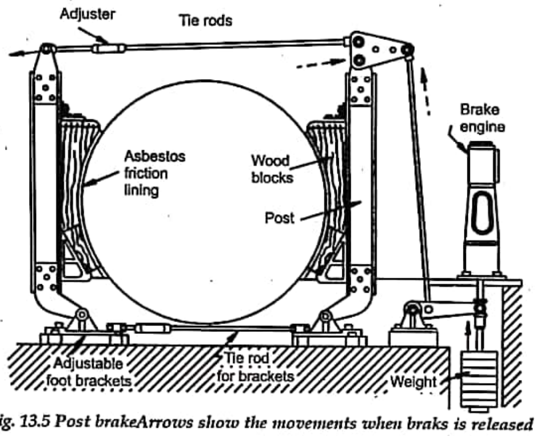


Fig. 13.5 Post brakeArrows show the movements when braks is released

Thrustor brake :

Some of the recent installations of winders use disc brakes in conjunction with a solenoid thrustor *e.g.* at Jaduguda mines. Fig 13.6 shows a solenoid thrustor brake. The solenoid is electromagnetic in action. The brake is set by adjusting the spring when the current does not flow through the solenoid. When the current flows through the electric motor of the winder (or the hauling/hoisting engines, the solenoid is energised and if exerts a

force pull and the broke blocks and releases drum to rotate. If there is failure of electric power there is no pull in the solenoid and the brake is automatically applied by the tension in the spring. Current to the solenoid is regulated by the operation of the hand or foot lever operating the brake.

In some other type of thrustor brakes, the thrustor is worked by oil pressure generated by an impeller driven by an electric motor. On the stoppage of electric current to the hauling/hoisting motor solenoid is de energised, the oil pressure reduces and the brake is applied by action of dead weights or by spring tension. The action of an oil pressure thrustor, while swift, is gentle, whereas the action of a solenoid may need to be damped by using a dashpot.

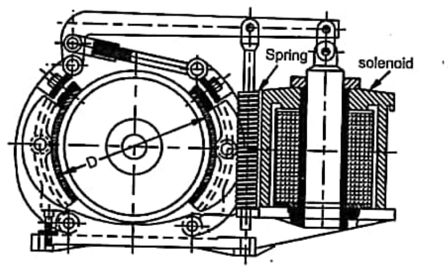


Fig. 13.6 A solenoid thrustor brake.
Also see "Thrustor Brake on conveyors", Chapter 17.3 Figure.

Electrical braking of winders :

The method of braking to be adopted in a particular case will depend upon the energy to be absorbed in the retardation period. If the energy to be absorbed is only small the power may be cut off a few revolutions from the end of the wind and the wind allowed to slow down, using the mechanical breaks to bring cages gently to rest. This is the only method possible on steam winders upto a depth of usually 200 m. At higher depth the energy to be absorbed during the retardation period is high owing to the down-going load over-balancing and the method of using only mechanical brake results in generation of heat at the brake linings, their heavy wear and the cost of their frequent renewal would be excessive. For deeper shafts, therefore some form of electrical braking is normally adopted for electrical winders in addition to the mechanical braking which is required under the Mining Law. The types of electrical braking for winders are :

1. Counter current braking
2. Dynamic braking
3. Regenerative braking
4. Eddy current braking

Of these the first 3 are commonly adopted.

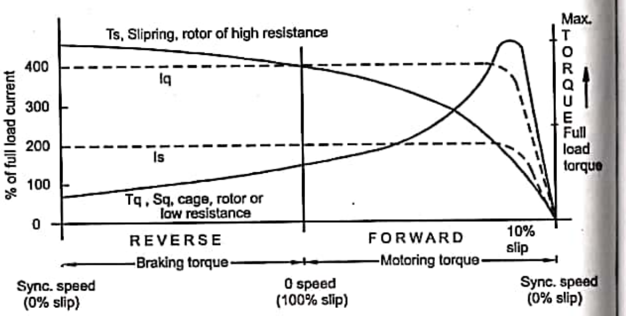


Fig. 13.7 Reverse current braking characteristics
Is–current of slip-ring motor
Ts–torque of slip ring motor
Iq–current of sq. cage motor
Tq–torque of sq. cage motor

1. Counter current braking :

This is effect by reversing the electric supply to the stator. Two phases of the stator supply are interchanged by first bringing back the winder operator lever to the "off" position and then to that position which runs the drum in opposite direction. The amount of braking depends upon the position of the lever since the lower the resistance in the controller (and therefore in the rotor circuit), the greater the rotor current and the braking torque produced. This method of braking is simple and commonly adopted, though it suffers from following disadvantages :

(a) A large amount of electrical energy converted into heat is to be dissipated by the rotor resistences and it is thus wastful of power.

The liquid controllers are preferred for dissipation of heat.

It is necessary to use a motor which is suitably rated to cater for the high internal heat losses experienced during reversal.

(b) It involves a high line current, and cables to carry such high line current have to be provided.

(c) The reversing switch is subject to severe wear and tear.

(d) When the direction of rotation of the stator magnetic field is reversed the voltage between stator and rotor is doubled and the insulation of both must be adequate to prevent brak-down.

(e) The method has been proved to be bad for the winding ropes.

2. Dynamic braking :

The principle of dynamic braking for A. C. winders is perhaps the most important development of economical braking which has yet been employed. The principle adopted is : by exciting the stator of the asynchmous Induction winder motor with D. C, and connecting the rotor windings to a variable resistance, the motor can be made to act as a dynamo, thus applying braking torque to the motor shaft.

A. C. supply is cut off from the stator of the winder motor and in its place D. C. excitation is applied to part of the stator winding. The slip rings of the rotor continue to remain electrically connected to the starting resistances. The D.C. supply in the stator winding results in a stationary magnetic field in which a rotor is revolving and causes the winder motor to run as an A. C. alternator. But the output of this "A. C. Alternator" is absorbed by the rotor resistences. By varying the rotor resistance the breaking torque can be varied and further control of the braking torque is possible by varying the direct current to the stator. By a combination of these controls effective braking down to 4 % of full speed is possible and below that speed mechanical brakes are applied. For short periods extremely high braking torque may be achieved by increaseing the direct current : this is usually referred to as *field forcing*

This system is particularly useful for heavy lowering windrs and for slow speed shaft inspection which would otherwise involve expensive mechanical braking, but it is used on an increasing scale for normal winding because of the saving in power and ease of control approaching the smoothness of Ward Leonard D. C. winder control thereby eliminating the coarse and costly use of reverse current braking or mechanical braking. It has a wide field of application apart from winding but the control gear required and the provision of a D. C. supply involves extra cost.

A Survey of dynamic braking braking systems used for a. c. winder motor indicates that they broadly fall into four categories :

1. Co-ordinated control
2. Compensated control
3. Torque control
4. Co-ordinated and compensated control.

Again these system differ in method of application depending upon whether the winder is grid resistance controlled or liquid rotor resistance controlled.

A clear understanding of the principles involved in these schemes requires study of induction motor behaviour under dynamic braking conditions.

The dynamic braking torque $\propto$ stator flux $\times$ rotor current

$$i.e. \quad \propto \text{d.c. excitation current} \times \frac{\text{rotor voltage}}{\text{rotor resistance}}$$

In the co-ordinated system of dynamic braking the required braking torque is obtained by co-ordinated action of increasing d. c. excitation simultaneously reducing rotor (external resistance. This system has the disadvantage that at reduced speeds the braking torque is less and in case of liquid rotor type of resistance density and temperature variation of electrolyte will affect the rotor resistance. The disadvantages of mechanically co-ordinated systems have led to compensated systems. In these systems the braking torque is made independent of speed by controlling stator excitation as a function of rotor current. In practice, these systems use rotor current as a medium to control the output of a unit in which stator current is desired. These units include rotary excitors, power transductors (magnetic amplifiers) and thyristors all of which act as amplifiers responding to rotor current signal. Thus the stator excitation current is related to rotor current and by this means compensation obtained for demagnetising effect of rotor current.

The torque control scheme developed by GEC claims advantages over compensated and co-ordinated system. The circuit is designed on the basic consideration that it is most desirable to maintain maximum torque at any excitation independent of speed. In this type of control the ratio of rotor resistance rotor speed is maintained constant. The latest development in dynamic braking is the use of thyristor with transistorised firing circuits to replace power amplifier supplying d.c. excitation.

3. Regenerative braking :

If the torque due to down-going cage/skip is more than the torque to be overcome by the winder for raising the asending cage/skip, the down-going load tends to accelerate the winder and its connected motor. If D. C. motor with fixed field strength is used for winding, as in the case of Ward Leonard system, the D. C. motor functions as a dynamo when the downgoing load tends to accelerate the winder and the power generated by such "dynamo" is returned to the power supply line as energy. The net effect is that the winder motor receives less effective power and this amounts to a braking action on the winding drum. This is known as regenerative braking. It can take place at any speed in the case of Ward Leonard system. It can take place at any speed in the case of Ward Leonard system. In the case of A. C. winders, however, regenerative braking can occur only beyond the synchronous speed of A. C. motor. The braking torque is proportional to the speed above synchronous speed upto about 175 % of full load torque. Change pole motors (A. C.) give regenerative braking after reconnection to a lower speed winding until lower speed is reached.

List of safety devices on winders :

The safety devices used for a winding system are the following and some of these have been described earlier in this chapter.

1. *Mechanical brake or friction brake* : When considering safety devices this is the first device that comes to mind, and is required by mining regulations.
2. *Additional mechanical brake* : (friction type) acting on the brake rim of the flexible coupling between the motor and the gear box. This is sometimes referred to as brake. The performance of this brake is to brake the inertia of the motor armature bringing it to immediate stop. This eliminates the harmful oscillations of the armature which would have otherwise occurred when the current to the motor is switched off. The motor brake operates synchronously with the service and safety brake mentioned at (1) above.
3. *Automatic contrivance* which (i) prevents oversending, (ii) prevents overwinding and (iii) ensures slowbanking at a speed during slow banking is 0.5 m/s.

4. *Reverse direction prevention switch :* It trips power if the winding engienman through mistake operates the motor in wrong direction.
5. *Time limit switches at the pit top* mounted on the headgear which trip electric supply to the motor if ascending cage overshoots the decking level. the cage strikes a lever which operates the switch. Such switch is sometimes provided for the down-going cage also.
6. *Rope deviation limit switch* used on multi-rope Koepe winder. If any one of the ropes deviates from its normal path, it strikes a small pulley which actuates the arrangement for tripping power to motor.
7. Limit switch for tail rope is reduced for any reason or if the loop is obstructed for any reason, a limit switch actuates the power trip switch.
8. *Tachometer generator on the gear box :* If the gear box is faulty and the winding drum shaft does not run at its normal speed the tachometer generator will not generate sufficient direct current. This shortfall in the direct current results in tripping of power to motor through electrical linkage. If the speed of the winder is exceeded by 15 % of its rated value, or if it is higher than 50 % shortly before the end-stopes,safety braking is initiated via auxiliary contractors by the tacho-generator the pantam measuring contactor and the cam switches on the drive controller.
9. Wedge arresters for down-going cage
10. *Safety catches* mounted on the headgear for the ascending cage.
11. *Safety detaching hook* for ascending cage.

A depth indicator indicates to the operator the depth of the cage/skip from the surface and cannot be considered as a safety device.

All the safety devices mentioned here are not installed for every winder installation.

Arrangements for monitoring of the winder motor temperature are provided on some of the recently acquired winders of Coal India Ltd. At the spots of the motor which becomes very hot in operation, thermo-couples are placed and these, in conjunction with a tripping and warning unit and auxiliary contactors, block the voltage supply to the motor if overheating occurs in the motor winding.

QUESTIONS

1. Describe a liquid controller for A. C. winder and the manner in which it controle the speed and torque of the winder.
2. Describe :

   1. Post brake as used on a winder for deep shaft

   2. A thrustor solenoid brake.
3. What are the different methods of electrical braking of winders? Describe in brief one of them.
4. What are the various safety devices used on electric winders Ω and describe any three of them in brief.
5. Explain the basic principle of working of Ward Leonard system as used for winding and describe with sketch its manner of speed control.

O O O

CHAPTER – 14

WINDING : PIT-TOP & PIT-BOTTOM LAYOUTS WITH CAGE WINDING & SKIP WINDING

Some of the layouts for pit-top and pit-bottom employing cages/skips for winding have been described in Vol. 1 of Elements of Mining Technology. Only a few of them have been reproduced in this volume and the reader may refer to Vol. 1 for other layouts.

The raising capacity of a mine depends on the shaft capacity which in turn depends on the manner in which tubs or mine cars are handled at the pit-top and pit-bottom layout is done with the following objects in view :

1. Use of the shaft to its full capacity.
2. Use of minimum number of tubs in the circuit.
3. Use of minimum number of operation.
4. Maintaining steady flow of tubs.
5. Minimum decking time.
6. Lowering of materials.
7. Handling of ores or coals of different grades.
8. Avoiding large excavations near pit-bottom.

In any pittop arrangement it is essential that the loaded tub or mine car, raised from the pit, discharges mineral close to the shaft and returns to the cage, so as to require the least number of tubs in circuit. It is also necessary that mine cars are not allowed to run freely under gravity over long distances.

Run round arrangement at pit-top (cage winding) :

A run-round arrangement is shown in Fig. 14.1. From the decking level, the loaded tubs are taken to the tippler T via a weighbridge W and empties travel by gravity to a creeper (which elevates them to a little above the decking level) and gravitate to the other side of the cage. A creeper on load side is not desirable and the usual arrangement therefore is to have the

decking level 4 to 6 m above the ground level on gantry. A weigh bridge for all the mine cars raised from the pit is a good practice but is uncommon in our mines.

If the quality of mineral raised from the pit is not the same, sometimes due to working of two or more seams of coal (or ore from 2 different levels) by the same pit, two or more tipplers have to be provided for the various grades. The figure shows coal of two grades coming from the different seams, each raised by a seperate pit. One tippler T_1 has been provided for the loaded tubs, containg shale or stone, which may be disposed of by a belt conveyor. Provision may be made for alternative arrangement to unload the coal tubs when the usual tipplers cannot be used due to breakdown stoppage of screening plant. Such arrangement consists in providing one or two travelling tipplers, depending upon the output, for tipping the coal into a dumping yard.

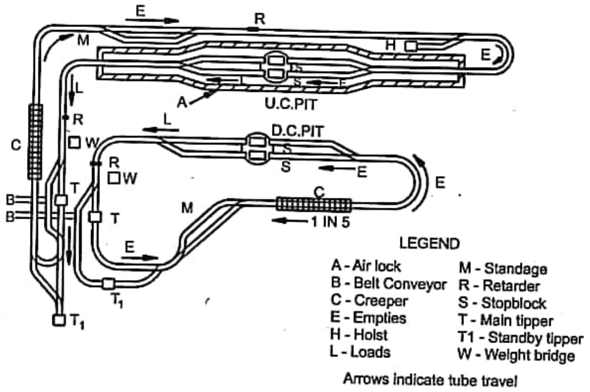


Fig. 14.1 Pit-top layout with run-round arrangement.

When the decking level is above the ground level, the materials are lowered into the mine by loading them into the cage at ground level and an opening in the shaft walling, equipped with a gate and a track, is provided for this purpose; alternatively, a hoist is used for taking materials to the decking level.

The main disadvantage of a run-round arrangment is the large space required and the long circuit which the tubs have to pass specially with long wheel-base mine cars which require large radius curves.

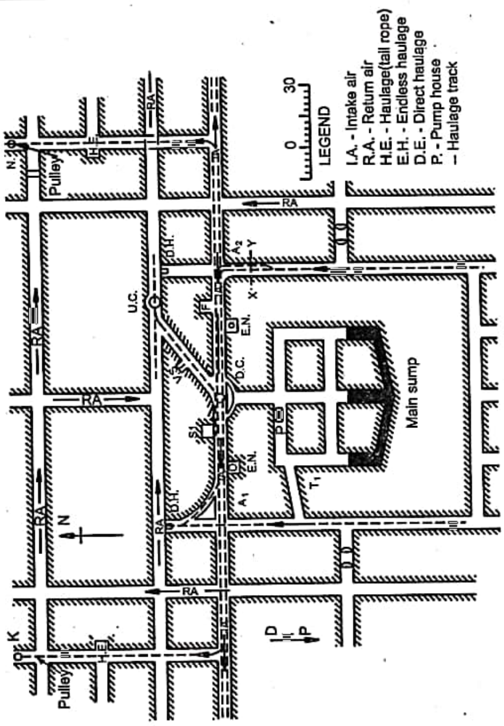


Fig. 14.2 Pit bottom layout. (cage winding). KLMN is shaft pillar.

The type of pit bottom layout to be followed depends upon the types of transport system used in the vicinity of the pit bottom and the method of winding, whether skip winding or cage winding. The pit bottom layout lasts the whole life of the pit, and has to be designed to meet the maximum production likely to be handled by it, as re-arrangement of pit bottom is expensive and may involve costly excavation in stone over a wide area, resulting thereby in weakening of the shaft pillar. The re-arrangement takes a long time and hampers normal production.

Though a pit bottom layout essentially depicts the transport arrangements near the pit bottom to deal with a targeted output, ventilation, drainage and support arrangements have to be considered in designing it. Fig. 14.2 shows the usual pit-bottom layout with cage winding and underground rope haulages in Indian coal mines.

Lofco System :

In this system there are two double tipplers, with one pair of tipplers on each side of the shaft. Referring to the Fig. 14.3 empty car at A is rammed into cage I, pushing a loaded car from the cage to tippler C. During the period of subsequent wind this car is tipped. When cage 2 comes up, empty car from D is pushed in the cage 2 and loaded car from the cage 2 runs into position B in the tippler, and the cycle is repeated. The original installation of this type was at Lofco house colliery in Britain (Hence the name Lofco). The mine cars, it may be noted, never leave the neighbourhood of the shaft. Efficient dust suppression arrangements have to the adopted as the dust raised during tipping may be carried down the D. C. shaft.

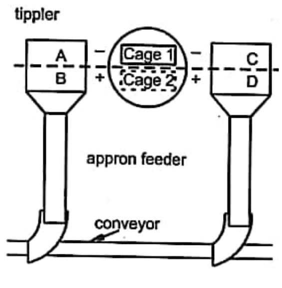


Fig. 14.3. Lofco arrangement at pit top

Back shunt circuit

The back-shunt circuit arrangement is adopted in the pit-top layouts at Chinakuri and Girmint collieries using 3.5 te capacity mine cars with a guage of 1.1 m. It is a cheap, efficient and simple arrangement of reversing cars, but a spaced feed is necessary to allow sufficient time for each car to clear the back-shunt before the next one enters. Clearance can be speeded up *(i)* by making the back-shunt rails very steep from the position at which the rear wheels of each car are clear of spring points, or *(ii)* by installing a spring buffer in the back shunt which will arrest the car as soon as it is clear of the spring points and expel it rapidly.

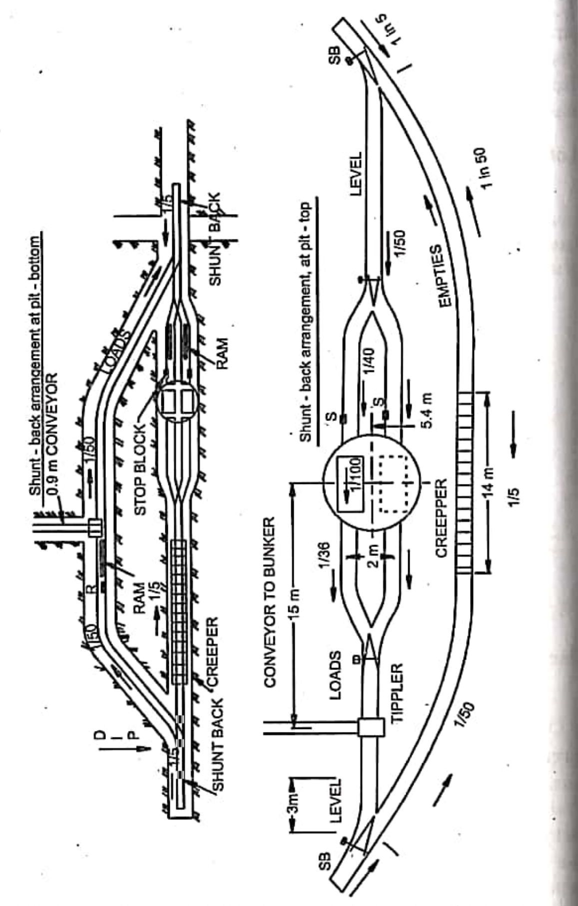


Fig. 14.4. Shunt back layout at pit bottom and pit top. 1/50 indicates a gradient of 1 in 50 Arrows indicate down-slope.

The arrangement is good where long wheel-base cars are used. The width of the circuit is reduced though the length required may sometimes cause difficulty if the winding engine room is very near the shaft. As the tippler is near the shaft suitable steps have to be taken to prevent coal dust from entering it, if down-cast.

The back-shunt circuit arrangement is shown in fig. 14.4. Most of the operations are automatic and only one car is pushed into the cage at a time. The empty car,leaving the shunt back, enters the cage and the points at the crossing are automatically made by the passing of the car for travel of the next car to the other cage. The tippler is electrically operated and only three men are required for the control of the pit-top ; one banksman, one tippler operator and one helper to assist the banksman. The arrangement is capable of dealing with an output of 50,000 te per month (coal)

Turntable circuit :

A turntable circuit ensures continuous feed of cars which need not be delivered to the turntable at regular intervals unlike the back-shunt. The reversal of car is accomplished within a restricted space. The turntables for outputs exceeding 500 te/day are usually power operated. The length of pit top required for turntable circuit is smaller than that for back-shunt circuit. Fig. 14.5 illustrates a turntable circuit. Only three men are required at the pit-top. The track on the empty side is curved because of the short distance between the shaft and the winding engine house. Turntable circuits with power operated turntables at Kunstoria colliery provides the most compact pit top arrangements with only 3 men at the pit-top in a shift for dealing with an output of 30,000 te per month (coal).

Traverser circuit :

Traversers offer very compact circuit, shorter than described above, where cars have to move from one side of the shaft to another. A traverser is a platform, running on rails laid at right angles to the car tracks which are parallel to the length of the cages. Mine cars, emptied at the tippler, to the length of the cages. Mine cars, emptied at the tippler, travel to the cage-side traverser which receives them, and the traverser is then pushed and positioned in front of a cage for ramming the cars into the cage. The traverser is powered by electricity, compressed air, by hydraulic means and sometimes by manual labour as in some mines of Jharia and Rangiganj fields. As traversers save a considerable space available for car circuit, they are advantageously used where space is limited, specially on the engine side. They are ideally suited for single deck cages. Tipplers are sometimes incorporated in traversers, making further saving of space and man power. Fig. 14.5 shows a traverser circuit.

As the traverser has to carry two cars when a tandem cage is used the track for traverser-travel is of wider gauge than the normal car track.

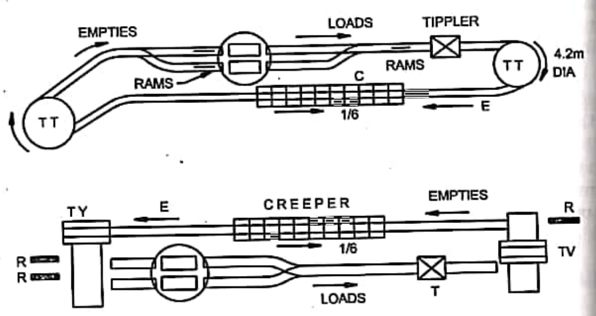


Fig. 14.5. Top-pit top layout with turn tables. Bottom-pit top layout with traversers. TV-traverser, R-ram, T-tippler 1/6 indicates a slope of 1 in 6.

The traverser circuit shown here employs only one creeper, with the results that the traverser near the cages has to travel less when feeding one cage, but more when feeding the other cage. This defect can be removed by the use of two creeper, one on either side of the load track, so that each creeper supplies empties to only one particular cage.

Unlike back shunts or turnable circuits, the capacity of a traverser circuit cannot be increased once it is installed and the installation should cater to the maximum output expected from the mine. A traverser can deal with 45 to 60 winds per hour and only three men control the pit-top. The traverser circuits, adopted in some mines of Jharia and Raniganj fields, use traverser only on the engine house side employ only one creeper.

In some modernised mines, the cabin of the banksman or the onsetter is on the traverser itself, which is electrically operated and equipped with pusher rams. This enables better control of the traverser by the operator.

Creeper Layout :

Fig. 14.6 shows another layout in a thick seam using creepers for handling empties and is adopted in some mines of Jharia and Raniganj fields the length of cages is in dip-rise direction.

Shunt back layout :

A pit bottom layout with traverser arrangement and a belt conveyor delivering the output of the mine to the pit-bottom is shown in Fig. 14.4.

A pit-bottom layout with traverser arrangement in conjunction with a belt conveyor feeding the pit-bottom is not used at any of our mines.

Locomotive layout :

In a layout with locomotive haulage designed for oneway traffic loaded cars are pushed into the cage on one side only while empties are taken out at the other side from where they are sent out to various districts. Each haulage track serving underground district must be connected with both the full and empty side.

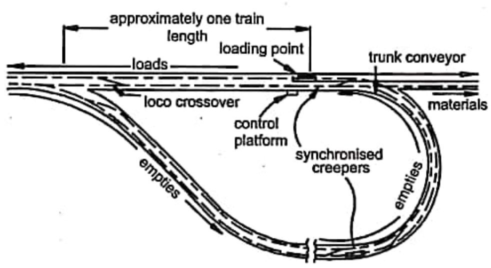


Fig. 14.6 Layout top semi-permanat loading session.

For locomotive haulage at the shaft bottom, there are two main types of layouts which are modified according as the shaft is situated in the axis of the main haulage, at right angles or at an angle.

1. Loop type
2. Reversing track type

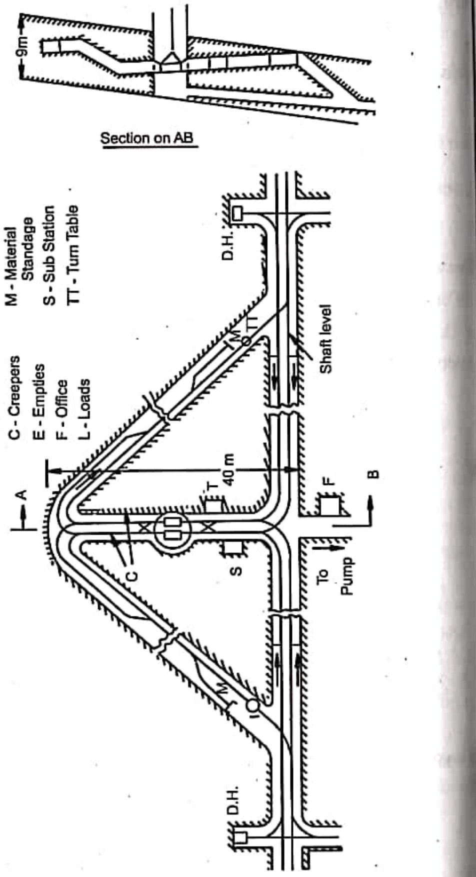


Fig. 14.7 Pit bottom layout in a thick seam with shaft axis along dip and rise

In the first type, a loop is provided for bringing the load on one side f the shaft and taking the empties to the districts. Larger loop will provide ore standing space.

The second type avoids the loop and brings the empties to the full side f the cage with the help of traverser, turntable, or shunt back. This eliminates long run round and reduces the idle travel of a locomotive to an absolute inimum; however, its capacity to deal with increased output is limited and it ecessitates greater width at the pit-bottom.

ayout for skip :

Pit bottom arrangements for loading the skip usually take three forms :

1. Mines cars tipping direct into measuring pockets,
2. Cars tipping on belt which delivers mineral into pockets,
3. Mineral discharged into storage bunker and fed to the measuring pockets.

The arrangement of tipping direct into pockets is not considered esirable for the following reasons :

(a) As mine cars are to be led to the top of the measuring pockets, large excavations are necessary near the shaft.

(b) If the haulage is to be in the intake a proper air-lock is to be maintained across the pocket, which interrupts unloading of cars when skip is being filled.

(c) Loading in the skip is not uniform and important control data are lost.

(d) The pit bottom becomes very congested.

In the second arrangement loaded cars pass over a tippler situated me 30-50 metres away from the shaft. Mineral is discharged into vibratory der. It feeds a conveyor delivering into a chute which deflects mineral into e of the measuring pockets fitted with antibreakage device. When the pocket filled with a skipload weight of mineral the weighing beam operates a lve which turns the deflecting plate of the chute to the other pocket and es top of the loaded pocket. The time taken for loading in a pocket nchronises with the time required for emptying the loaded pocket and winding of the loaded skip; when the arriving skip is delayed, conveyor and tippler automatically stopped by an interlock system.

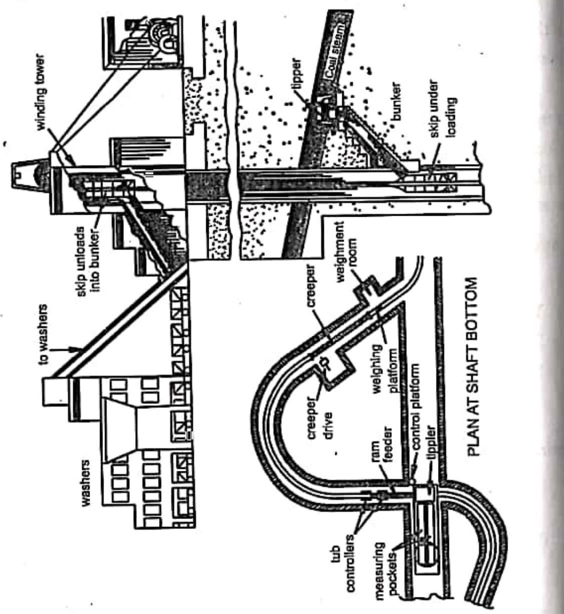


Fig. 14.8 General arrangement at the pit top and pitbottom loading point

This method ensures correct loading of the skip and eliminates other drawbacks of the earlier arrangement.

In the third method a trunk conveyor discharges into a concrete bunker with sides sloped at 45°. A feeder draws the mineral from the bunker and delivers to a conveyor which conveys it to the pockets as described above.

Pit-top arrangement for a skip :

Arrangements at the surface for unloading the skip are shown in Fig. 14.8. Level in the mineral hopper is known to the banksman and winding

engineman from visual indicators. As the loaded skip comes to bank, the discharge door of the hopper is closed and receiving door opened; when it is hoisted up in position, its bottom discharge door is opened automatically and lets out mineral into the hopper. As the skip is lowered, its discharge door is closed ; receiving door of the hopper is also closed and its discharging door opened automatically. Suitable system of interlocks ensures performance of all operations in proper sequence.

QUESTIONS

1. Give a suitable layout for a run-round arrangement at pit top of a coal mine capable of dealing with an output of 30,000 te per month. 1.1m³ capacity coal tubs are used and cage winding is adopted.
2. Give a pit bottom layout with shunt-back arrangement
3. Sketch a suitable layout for the pit top using (a) traversers and (b) turn tables. Under what circumstances are these layouts considered advantageous.
4. Make a suitable sketch showing the pit bottom and pit top arrangement when employing skips.

O O O

CHAPTER – 15

TRANSPORT : ROPE HAULAGES & TRACKS

Rope haulage is the dominating mode of transport in Indian coal mines though belt and chain conveyors were introduced on a large scale in mechanised mines opened within the last two decades. Underground locomotives are used on a small scale in coal mines, but the metaliferous mines have adopted them as a standard transporting medium.

The main methods of transport are as follows :

A. Rope haulage

- *i.* Direct rope haulage.
  - (a) Tail rope haulage.
- *ii.* Endless rope haulage.
  - (a) Over-rope,
  - (b) Under-rope.
- *iii.* Main and tail rope haulage.
- *iv.* Gravity rope haulage.

B. Conveyor system of haulage

- *i.* Belt conveyors.
- *ii.* Cable belt conveyors.
- *iii.* Chain conveyors.
  - (a) Scraper chain conveyors.
  - (b) Armoured chain conveyors.
  - (c) Gate end loader.
  - (d) Mobile stage loader.
  - (e) Pickaback conveyor.
- *iv.* Plate conveyor.
- *v.* Disc conveyor.

C. Locomotive haulage

- *i.* Diesel locomotive.
- *ii.* Electric battery locomotive
- *iii.* Trolley wire locomotive
- *iv.* Cable reel locomotive.
- *v.* Compressed air locomotive
- *vi.* Electro-gyro locomotive.

D. Shuttle cars

Underground transport arrangements are generally divided into 2 categories.

1. Main haulage,
2. Gathering haulage.

The main haulage arrangement is that which operates between winding shaft/incline and the main underground loading points. At the main loading point the loads are collected from one, two or more districts. The gathering haulage arrangement is that which operates between the working faces and the main loading points. In a large mine, where the working faces are far from the main loading points an intermediate transport arrangement operates and it is known as secondary haulage.

The main, secondary or gathering haulage may be by ropes, conveyars, locomotives or a combination of these.

ROPE HAULAGE :

The rope system covers the following types of haulages :

1. Direct rope haulage.
2. Endless rope haulage.
3. Main and tail rope haulage.
4. Gravity haulage.

Direct rope haulage :

This is the simplest system employing one pulling rope and one haulage drum for hauling mineral in tubs or mine cars up a gradient which is generally steeper than 1 in 10. The haulage engine is situated at the top of an inclined roadway. The train of tubs is attached to one end of the rope, the being fixed to the haulage drum. The empty tubs attached to the end of the haulage rope travel on the down gradient by their own weight and do not require power from the haulage engine. The drum shaft is, therefore, provided with a jaw clutch to disengage it from the engine. The rope speed is generally 8-12 km/h and the system can operate between any point of the haulage requirements of an advancing working face. Only one haulage track is required. The system can also serve branch roads if the gradient is suitable for down-the gradient movement of empties by gravity. For this reason, the branch road deviating at an angle of not more than 40° off the main road is convenient. A slipring motor with drum controller is used :

The main disadvantages of direct haulage :

1. High peak power demand as load starts its journey up the gradient.
2. Severe braking duty on the downward run.
3. High haulage speed demanding high standard of track maintenance.
4. Not suitable for mild inclination of roads.
5. A derailment is associated with heavy damage because of the high speed.

A backstay is fixed to the last tub of a set of tubs when it is hauled ungradient. This is a provision to arrest run-away of tubs in case of breakage of rope or tub couplings.

Direct rope, double drum balanced haulage :

This is modification of the direct haulage system. In the double drum method, two drums are provided so that when a train of full tubs is being hauled outbye, a set of empty tubs is lowered inbye. Both the drums are fitted with clutches and are mounted on the same shaft ; weights of the rope and tubs are balanced and the only unbalanced load for the haulage engine is generally of the mineral. This results in a reduced peak power demand and easier braking. The system gives highest output as each trip of the ropes brings the loads and there is a regular delivery of the loaded tubs. The system requires wider road for the haulage track and the different arrangements of the tracks are shown in Fig. 15.1.

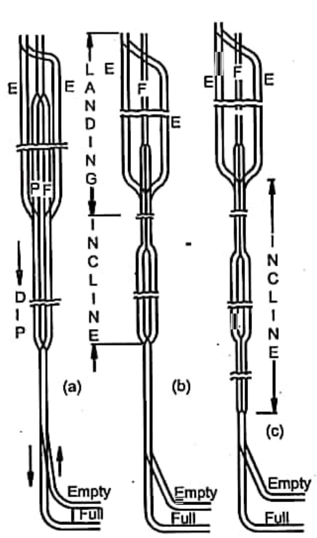


Fig. 15.1 Track layout of balanced double-drum haulage. E - track of empties; F - track of fulls i.e. loads

Endless rope haulage :

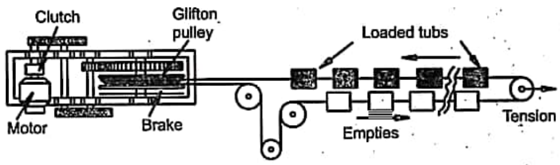


Fig. 15.2 Endless rope haulage system.

In this system, there are two parallel tracks side, one for the loaded tubs and the other for the empties, and an endless rope passes from the driving drum/pulley located at the outbye end of the haulage road to the inbye end and via a tension bogey. The tubs, loaded as well as empties, are attached to the rope at regular intervals with the help of clips so that the entire rope length has tubs on it intervals. Only one tub is attached to the rope at a time, but where lashing chain is used for attachment, the normal practice is to attach a set of tubs and the attachment or detachment is performed by stopping the rope. If, however, clips are used for singal tubs they can be attached or detached when the rope is in motion, The gradient of haulage road is mild and rarely exceeds 1 in 6. The rope speed ranges between 3 km/hr and 7 km/hr and the haulage is slow moving. The rope moves in one direction only. A squirrel cage motor is commonly employed.

Clifton pulley :

The driving pulley of an endless haulage is clifton pulley, C-pulley or surge wheel and is of a special shape. To protect the main driving wheel from wear the pulley is fitted with renewable lining of C. I. or soft steel segments having a taper of about 1 in 8. These segments are fixed on the rim of the driving wheel by counter-sunk bolts and have side flanges. The incoming rope pulling loads enter the segments, leaves them at the smaller diameter. The rope should not be loose on the pulley and in order to keep it in proper tension, due to fluctuations of the load. The common practice to ensure mounted on a special tension is to pass the rope half turn around a sheave mounted on a special tension bogey or tension carriage placed on rails. Heavy weights are attached to the bogey through a short length of chain or wire rope alternatively the bogey may be placed on a track on a sloping plane and the weights placed directly on the bogey.

The correct place for the tension bogey is at the point where slack rope is most likely to occur. On a level road or on a road rising inbye the bogey should be placed near the haulage engine ; on a road which dips inbye the tension bogey should be placed at the bottom (inbye) end of the haulage road.

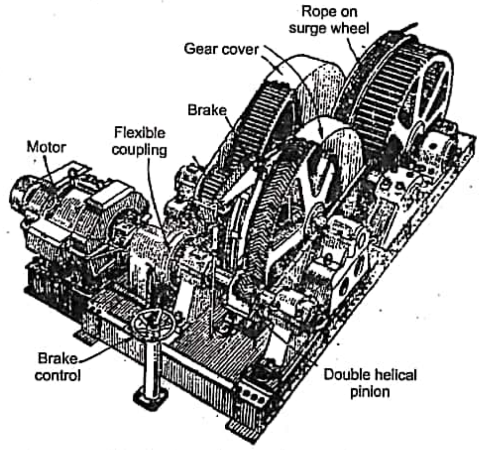


Fig. 15.3 An endless haulage engine

Endless haulages are of over-rope type or under-rope type, the latter being very common. In the over-rope type the haulage rope passes over the tub or set of tubs and in the under-rope type it passes beneath the tub or set of tubs.

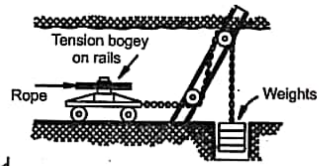


Fig. 15.4 Tension bogey

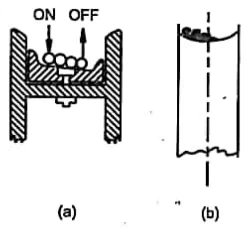


Fig. 15.5 Surge wheels (a) non-reversible (b) reversible

The design of endless haulage rope clips depends whether the haulage is of over-rope type or of under rope type. Some of the clips used are screw clips, smallman clips and cam clips (Fig. 15.6 and 15.7)

i. ***Screw clip*** : This clip is tightened on the rope by a handle and screw and the handle is coupled to the drawbar of the tub by a long steel rod hinged to the clip.

ii. ***Smallman clip*** : This consists of a pair of stee-cheeks or side-plates, loosely held together by the adjustable central bolt B which has a spring surrounding it to keep the plates apart, and kept in position by the pins supporting the lever and the coupling hook.

The bent lever is pivoted at P and carries at its upper end a wedge. A which works between curved surfaces on the inside of the cheeks. When the lever is depressed, the wedge A enters the narrower part of the space between the cheeks, so forcing them apart at the top, and at the same time causing the bottom jaws to grip the rope. The jaws are about 15 cm long and are lined with renewable soft iron bushes. When the lever is raised, the wedge A moves towards the wider part of the space between the cheeks, so releasing the rope from the jaws.

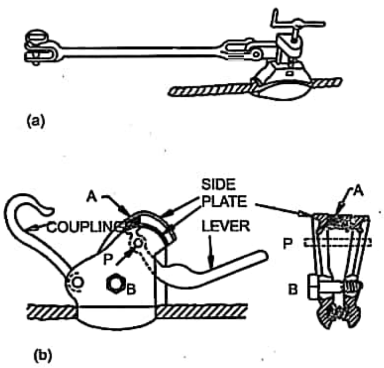


Fig. 15.6 (a) Screw clip (b) Smallman clip

The clip can be detached automatically from the rope by fixing a bridge-piece or trip-bar to a sleeper at such a height and in such a way that the rope passes underneath whilst the lever of the clip strikes against it and is thereby raised. At detaching points, the gradient should be in favour of the tubs.

iii. ***Cam clip*** : This consists of a plate C and a cam-shaped lever L which is pivoted at P and is connected by a small chain to the tub to be hauled.

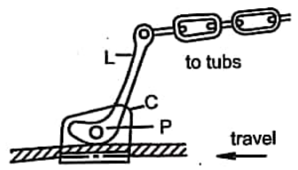


Fig. 15.7A Cam Clip.

The pull of the tub turns the lever around the pivot P so that the grip of the clip on the rope is proportional to the load. On undaulting roadways, a clip must be provided at each end of the tub.

iv. ***Lashing chain*** : The lashing chain is usually 2.5 to m long with a hook is attached to the tub draw bar. Other end of the chain is coiled 3 to 4 times around the haulage rope and the second hook is linked to the chain. On undaluting roads, one chain is attached in the front and another chain behind the set of tubs but on a gradient only one chain is needed.

Where the gradient is not mild, a tub or set of tubs is attached to the rope by two clips, one in the front and the other at the back. It is a standard practice to attach or detach tubs when the rope is in motion. If the rope is to be stopped when attaching or detaching tubs, the total timing of rope running is only a small proportion of the shift timing. Such situation arises if tubs are to attach or detach at different levels or loading points along the haulage plane and the haulage attendants are not sufficiently skilled.

Advantages of endless haulage :

1. Because of the slow speed, less wear and tear,
2. Accidents from derailed tubs do not cause much damage due to slow speed.
3. Motor of less power required.
4. It does not place heavy peak demand on the power supply.

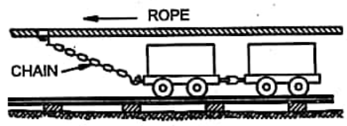


Fig. 15.8 Lashing chain for over-rope endless haulage.

Disadvantages :

1. It requires wide roads for two tracks.
2. It is not suitable for steep gradients.
3. Load on the rope is large and a rope of larger cross-section is required.
4. Large number of tubs and clips are required as rolling stock,
5. If a breakdown of any tub occurs the whole system comes to a stand still.
6. It cannot serve a main road and a branch road simultaneously unless elaborate arrangements are made to course the rope to the branch line with the help of deflection pulleys. The tubs of main-road rope have to be detached and re-attached at the branch line.

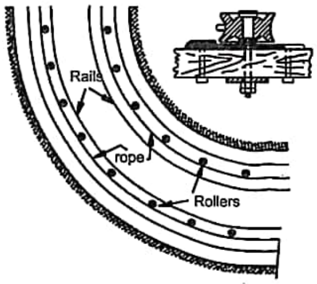


Fig. 15.9 Curve of an endless haulage showing (separately) curve pulley.

Reversible balanced haulage with endless rope :

This may be regarded as a variant of the double drum balanced haulage system whereby the full and empty trains, instead of being attached to two separate ropes, are secured by clips to an endless rope which is driven by a reversible surge wheel.

It possesses most of the advantages of the double-drum system, and, in addition more accurate positioning of the loaded and empty trains and more uniform wear of the rope can be obtained merely by adjusting the position of the clips on the ropes.

Main and tail rope haulage :

In this system hauling engine is provided with two separate drums, one for the main rope which hauls the full train out, and one for the tail rope which hauls the empty train in. When one drum is in gear, the other revolves freely on the shaft but controlled, when necessary, by the brake to keep the ropes taut. The main rope is approximately equal to the length of the plane, and the tail rope twice this length. Only one track is required.

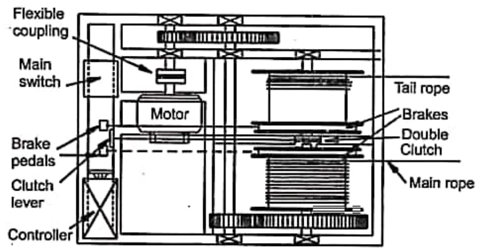


Fig. 15.10 Main and tail rope haulage

This system of haulage is suitable for undulating roadways. Where it is impossible orundesirable to maintain the double track required for endless rope haulage; it can readily negotiate curves, and it is convenient for working branches.

On the other hand, it operates at fairly high speeds and with long trains, and if a derailment occurs, the resulting damage and delay are likely to be considerable.

Tail rope haulage :

A tail rope haulage is one in which the haulage is situated at a lower level and the empties are hoisted up the sloping track. The haulage rope passes to the train of empty tubs via a deflection pulley located at the top of the roadway. The loads travel by gravity down the gradient but as the rope is attached to them their descent is controlled by the haulage driver.

Gravity haulage or self acting incline :

This is a haulage without any motor or external sources of power and consists of a cast iron pulley, 1.3 m-2 m diameter, having a brake path on one side and a strap brake.

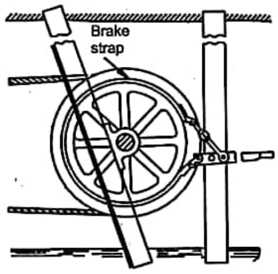


Fig. 15.11 Jig pulley of gravity haulage

It is located at the top of an inclined roadway and is employed to lower by gravity loads attached to one end of the rope while simultaneously hoisting empties attached to another end of the rope which passes round the jig pulley. The jig pulley is vertical. Only single track is required for its operation but at the mid-way of the road where the loads and empties meet, double track or a bye-pass is essential. Fig 15.12 shows the lay-out of the track and the jig pulley.

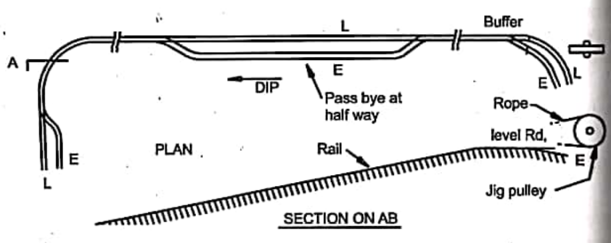


Fig. 15.12. Plan and section of layout of gravity haulage

Essentials of a good haulage track :

A haulage track underground, specially on a main roads, must be capable of carrying the loads imposed upon it with safety and security over a long period of time, with no risk of derailment and a minimum cost of maintenance and repair. The essentials are :

1. Rails of adequate weight and cross-section to carry the load.
2. Fish-plated joints, with lock washers and bolts, the joint being squares across the tracks, except on curves, where they should be stag red.
3. Sleepers of adequate length and cross-section, preferably with steel channel sleepers at intervals to give rigidly and maintain the gauge.
4. Well-rammed ballest (broken rock, gravel, slag or clinke) to provide a cushioned bearing surface.
5. Good drainage to maintain the track in a dry condition.
6. Careful alignment (if necessary with a dial or theodolite) before and after ballasting.
7. Careful grading (by a grade board and mason's spirit level for locomotive track only.)
8. Curves of adequate radius.

The weight and section of rail selected would depend on the axle loads, the proposed haulage speeds, the danger of corrosion in a wet mine, the need for adequate cross-section to resist wear, and the capital cost. Experience indicates that for locomotive haulage, the choice should normally lie between 18 kgf per m length for locos upto 10 tonne and 28 kgf per m for the larger sizes. Many rules have been suggested for the minimum weight of rail that is permissible to use, but one of the simplest states that, provided the distance between the centres of two adjacent sleepers does not exceed 0.85 m, the minimum weight of rail should be 5 kgf per metre, plus 2.5 kgf for each te on a pair of wheels. For tubs of 1.1 m^3 capacity the rail section should be of minimum weight of 16 kgf per metre length. Track gauge for tubs of 1.1 m^3 capacity is generally 0.6 m but for locomotive haulage it is usually 1 m-1.2 m in our mines.

Sleeper and ballast :

The purpose of these is to distribute the intense bearing pressure of the wheels on the rails over a sufficient area of floor so that the safe bearing pressure of the floor is not exceeded. If the floor is wet, it is essential to make provision for draining the floor on which the track rests. In the mines sleepers are usually of sal wood. The ballest consists of crushed rock like sand stone, schist, or any other hard rock, crushed to 50 mm - 60 mm size.

Curves and super-elevation (or cant) :

The minimum radius of curve on which locomotives and mine cars can operate is dependent on a number of factors including length of rigid wheel-base, rail gauge, diameter of wheels, super-elevation and speed. Sharp curves cause increased tractive resistance, excessive rail and wheel wear, derailments and possible breakage of locomotive axles.

Curve radius : It is suggested that for speeds upto 13 km/hr the radius of any curve in a main haulage road should be not less than 20 times the longest rigid wheel-base of any vehicle in the train ; and for speed overs 13 km/hr the radius should be greater than this. The plate layer or line mistry lays down the curve by experience but for locomotive haulage the surveyor comes in the picture and he gives marks on the roof of underground roadway for the path of the curve. A jim crow (Fig. 15.13) is a handy device for bending the rail to suitable curvature, in strages.

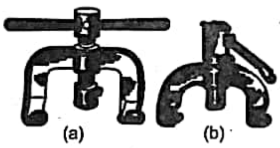


Fig. 15.13. A jim crow

Super-elevation : On a curve, centrifugel action creates a tendency for the train to leave the track and proceed along a course tangential to the curve. This throws the wheel flanges hard against the inner edge of the outer rail, causing excessive wear on the wheels and rails. To counteract this the outer rail should be raised above-all rope haulages, e. g. , with the main and tail rope system the forces in the two ropes pulling in the opposite directions tend to pull the train into the inner rail.

The amount of super elevation or cant required is dependent on the radius of the curve, the speed at which the train is travelling and the gauge of the track. If for any reason it is impracticable to super-elevate the outside rail, a check rail should be provided along the inside rail of the curve for its full length.

Super-elevation is given by the formula

$$\text{Super elevation} = \frac{Av^2}{gr}$$

Where A is gauge of track, metres

v is velocity of train, m/s

g is 9.81 m/s²

r is radius of curve, metres

Example :

A locomotive weighing 15 tef travels round a curve of 80 m radius at a speed of 30 kmph. If the gague is 1 metre what should be super-elevation of outer rail over the inner rail so that there is no thrust between the flanges of the outer wheels and the outer rail?

Ans :

$$v = 30 \text{ kmph} = \frac{30 \times 1000}{3600} = 8.33 \text{ m/s}$$

$$\text{Super-elevation} = \frac{Av^2}{gr}$$

$$\text{Super-elevation} = \frac{1 \times 8.33 \times 8.33}{9.8 \times 80}$$

$$= .0885 \text{ m}$$

$$= 88.5 \text{ mm.}$$

Widening gauge on curves :- It is advisable to widen the gauge on curves to prevent the wheel flanges binding on the rails and thereby increasing the track resistance. A widening of the gauge by 6 mm is sufficient.

Fig. 15.14, 15.15 and 15.16 show the different crossing and turnouts used on a haulage track.

Tubs and mine cars :

A mine tub is a box open at the top, and mounted on a wooden frame fitted with wheels. The axle rests in a pedestal block and is prevented from falling out of place by a clamp of mild steel flat. Ball or roller bearing is not used. Long members of the wooden frame act as buffer and a thick m. s. flat in the middle, projecting beyond the sides, constitutes the drawbar. The tubs in coal mines are usually of 1.1 m³ capacity. Their contents are unloaded manually or by a tippler.

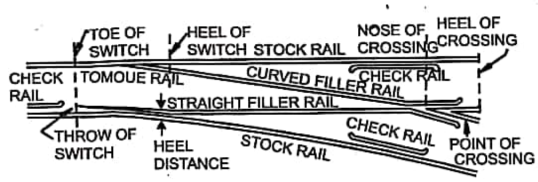


Fig. 15.14 Layout of points and crossings

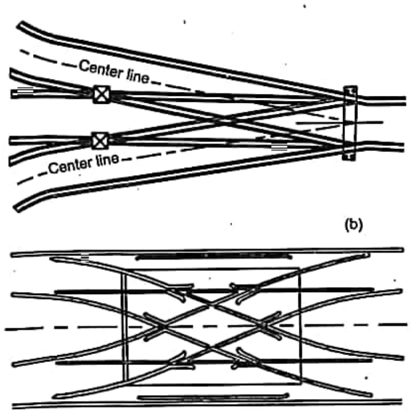
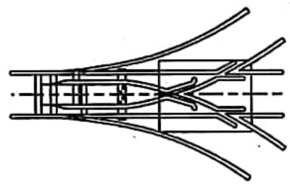


Fig. 15.15 (a) 3-way finger switch. (b) diamond crossing

A mine car is of larger capacity, usually 3 m³ or more, and the construction is like a tub with the difference that the roller bearings. Unloading is by a tippler and rarely, manually. Some mine cars are designed for automatic unloading when they pass over a bunker. One such mine car is Granby mine car.



15.16 Double turn out

It is provided with side discharge doors hinged at the top and interconnected by linkage rods. One of these side doors is fitted with a roller at mid height and when the car passes by the side of a sloping ramp (Fig. 15.17), the roller passes over the latter, this resulting in gradual opening of the side doors and the contents are discharged. The ramp is installed on the bunker. When the car is clear of the ramp the side doors close due to their weight and are held in position by catches thereby preventing any accidental opening during travel of a loaded car.

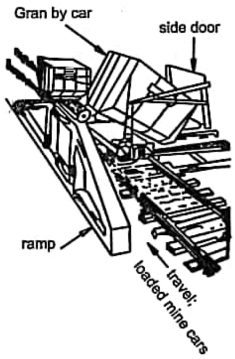


Fig. 15.17 Granby car for automatic unloading

Safety devices on haulage roads :

Monkey or back catch :

This consists of (i) a pivoted piece of steel rail placed between the track rails so as to catch the axle of a backward runway, or (ii) a wooden block pivoted at one end and pressed over the rail by a strong spring. It is used for endless haulage track for tubs moving up-gradient. (Fig.15.18)

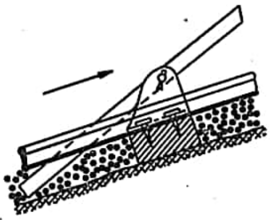


Fig. 15.18 A back catch

Stop-block :

This consists of a wooden beam or block lying across the rails, pivoted at one end and held against pivoted side block at the other. It is a good plan to have two stop-blocks some distance apart, the one forming a reserve for the other.

Backstay :

This is used behind an ascending set of tubs on a direct haulage road or on an endless haulage. It is attached to the tub axle and in the event of runaway of tubs, the pointed end of the backstay stops against sleeper of the track and the travel of the tub or train of tubs is arrested.

Drop Warwick :

This is intended for arresting forward runaways, being placed below the brow of an incline and also near the bottom and below intermediate levels. It consists of a heavy-baulk or girder hinged at one end to a specially set roof grider and held up at the other by an eye-bolt and pin. The warwick is released when required in emergency by a haulage worker pulling the wirewick is released when required in emergency by a haulage worker pulling the wire to withdraw the pin.

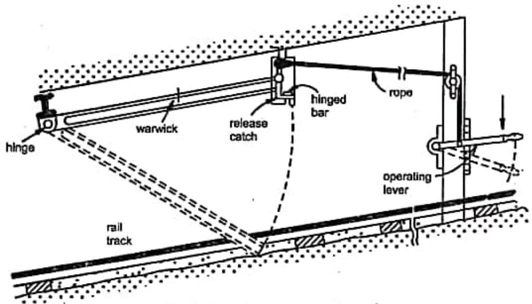


Fig. 15.19. Drop warwick.

A combined stopblock and runaway switch :

This is used at the brow of a direct haulage plane and is so constructed that at one time either the stopblock or the runway switch is effective in the event of a runaway of a set of tubs. It is manually operated by the haulage attendant when the set of tubs has to pass clear of the stopblock.

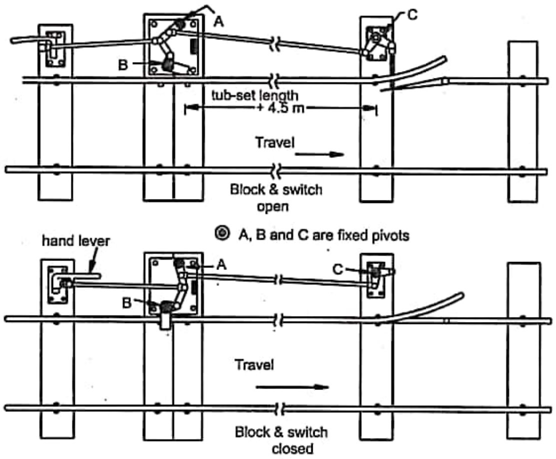


Fig. 15.20 Combined stop-block and runaway switch.

Hydraulic tub retarder :

It consists of a hydraulic cylinder containing opposed pistons. The hydraulic pressure is supplied from a 1-2 kW electric motor driven pump. The hydraulic tub retarders are used for locomotive haulages but for ordinary rope haulage where tubs of 1.1 m^3 capacity are used wooden sprags, skillfully inserted between the spokes of a running tub wheel, suffice to slow down the tub.

Signalling system for rope haulages. Overhead bare wire system is the general practice. Galvanised iron signal wires are supported on insulators fitted on long angle iron pegs which are fixed in the holes of the coal/rock pillar. The wires are connected to a bell circuit for which the electric supply should not exceed 25 volts. The power may be available from a Leclanche cell or dry cells or from transformer on the power mains. The trammer travelling along the haulage track connects the two overhead bare wires by a small piece of wire to give signal to the haulage engine driver, whenever it becomes necessary. On long haulage roads the resistance of the overhead wires becomes excessive and signals may not be audible. To overcome this problem, a relay is used as shown in the figure.

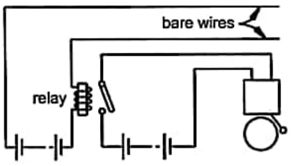
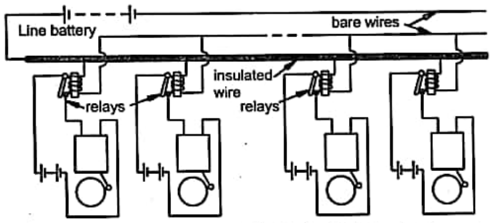


Fig. 15.21

Two-wire signal system with relay. (There are sepearate batteries for the relay and the bell circuits).

A relay is a small auxiliary device, electrically or compressed air operated, which actuates main component or part of a machine or de-energise an electric circuit for the protection of equipment. Electric relays use solenoids.



Three-wire signal system with one wire insulated. Bell in parallel each operated through a relay

Fig. 15.22

Three wire signal system with one wire insulated. Bells are installed in parallel, each operated through a realy and provided with its own source of current, such as a battery

Braking of haulage engines :

Braking of winding engines is described in the chapter on winding. In this chapter we shall consider working of brakes in general with particular reference to haulages.

A brake is a mechanism which designed to regulate the speed of a machine or to stop the motion thereof absorbing the kinetic energy of the moving parts. Brakes are of various types such as mechanical, hydrulic, electrical, compressed air and their combinations.

Mechanical brakes :

The principle of mechanical brake is to bring two members having relative motion to a state of very little relative motion or no relative motion, by utilising friction between the mating members. A friction clutch also works on this principle of utilising friction for its operation but the essential difference between the two is that the friction clutch is used to keep the driving and driven members moving together whereas brakes are used to stop a moving member or to control its speed.

The capacity of any mechanical brake depends upon the unit pressure between the braking the surfaces, the co-efficient of friction and the heat radiating capacity of the brake. The co-efficient of friction of the blocks used for braking is increased by fitting them with a lining of asbestos. The brake path is of cast iron or hard steel.

The types of brakes commonly used for mining machinery are :

1. Block brakes
   (a) *Post type brakes*
   (b) *Caliper type brakes*
2. Band brakes or strap brakes
3. Disc brakes

Internal expanding brakes are used mostly on automobiles.

A thrustor brake is of electro-magnetic type employing a solenoid but it is essentially a post brakes or caliper brak. it has been described in the chapter on winding.

The best type of brake to use is the post type brake with posts of curved caliper shape for applying the pressure over a longer are of brake path. The following example shows the method of calculating the force required to brake a particular haulage.

Example :

A train of mine cars of total 50 tef is attached to a direct rope haulage and is travelling down a gradient of 1 in 10 (average), the length of haulage plane being 1000 m. The co-efficient of friction of mine cars is $\frac{1}{50}$ and that of the rope, $\frac{1}{20}$. The rope weighs 2.73 kgf / m. The radius of drum is 1m and that of brake path is 666 mm. Width of brake path is 150 mm. There are two brake blocks, each having an angle of contact of 90°on the breake path. The break is lined with bonded asbestos with a co-efficient of friction of 0.3. Find out the radial pressure to be exerted on the brake path.

Ans. :

i. Force in rope due to descending train

$$= \frac{50 \text{ tef}}{10} = 5 \times 1000 \times 9.81 \text{ N} = 49059 \text{ N}$$

ii. Force in rope due to its own weight

$$= \frac{26.8 \times 1000}{10} \text{N}$$

$= 2680$ N.

iii. Braking force due to friction of trains

$$= \frac{50 \times 1000 \times 9.81}{50} \left(\text{as } \mu = \frac{1}{50} \right)$$

$= 1000 \times 9.81$ N

$= 9810$ N.

(iv) Braking force due to friction of rope

$$= \frac{26.8 \times 1000}{20} = 1340 \text{ N.}$$

The net force at the drum shaft is

(i + ii) – (iii + iv)

$= (49050 + 2680) - (9810 + 1340)$

$= 40580$ N.

Thus pull on rope is 40580 N and since radius of pull is 1 m, hence torque is 40580 N × 1 m = 40580 Nm.

If BF is the braking force, torque due to force in rope

Torque due to braking force.

$\therefore$ 40580 Nm = BF × o.66 m.

$$\therefore \text{BF} = \frac{40580}{0.66} = 61485 \text{ N.}$$

Frictional circumferencial pull on brake is 61851 N. Total brake surface in contact with drum is : Circunference of brake path (in m) × are of contact.

$$= 2\pi \times 0.66 \times \frac{150}{1000} \times \frac{90 \times 2}{360} = 0.311 \text{ m}^2$$

Radial pressure per m² of brake surface (asbestos lining) is, say, p Newtons

Then p × o.311 × 0.3 = 61485

$$p = \frac{61485}{0.311 \times 0.3} = 659003 \text{ N/m}^2$$

$= 659$ kN/m²

The power and efficiency of brakes and the service of brake lining depend largely on the manner in which the pressure is distributed over the whole working face of the lining. In the ordinary post type of brake the distribution of pressure is far from uniform, the heaviest being exerted near the top end of shoes and gradually reducing to zero at the bottom ; this makes for unequal wear of the brake lining and also reduces the mechanical advantage or the leverage.

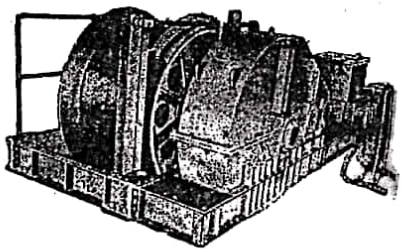


Fig. 15.23 Direct haulage

ROPE HAULAGE CALCULATIONS :

For solving examples on transport by rope haulage or by locomotives it is essential to make a few assumptions as to the effect of friction and to allow for possible increases in the load, but the problem essentially is one of moving the body on an inclined plane.

Example :

A direct rope haulage pulls 10 loaded tubs at a time up an incline dipping at 1 in 8. An empty tub weighs 400 kgf and has a capacity of 900 kgf of mineral. Length of the roadway covered by the haulage is 500 m. The rope diameter is 25 mm. Rope speed is 12 km/h. Coefficient of friction for the tubs and for rope is $\frac{1}{20}$. Estimate the power required by the rope and by the engine.

Ans. : Rope diameter is 25 mm.

$\therefore$ mass in kg per m length of rope = kd^2

Where d is diameter in cm and k is 0.35 for a fibre core rope made from steel having breaking strength of 1570 MN/m² (*i.e.* 160 kgf/mm²)

$\therefore$ mass in kg/m = $0.35 \times 2.5^2 = 2.18$ kg/m

and wt. of rope per metre = 2.18×9.81 N

$= 21.39$ N

Since the rope is continually shortening, the work done in pulling the load is continually decreasing, but for practical purpose we may consider it as constant in order to calculate the maximum power required which is always at the start of the journey of loaded tubs.

Rope speed is 12 km/h = 3.33 m/s.

The total work done may be divided into 4 groups as follows :

1. Component of the wt. of loaded tubs parallel to the haulage *i.e.* gravity component

$$= 1300 \times 9.81 \times 10 \times \frac{1}{8} N$$

$$= 15941 \text{ N}$$

2. Component of the wt. of full length of rope, parallel to the haulage plane, i. e. gravity component

$$= \frac{2.18 \times 9.81 \times 500}{8} \text{N}$$

$$= 1336.6 \text{ N.}$$

(3) Friction of load $= 1300 \times 9.81 \times 10 \times \frac{1}{20}$ N

$$= 6376.5 \text{ N.}$$

(4) Friction of rope $= \frac{2.18 \times 9.81 \times 500}{20}$ N

$$= 534.6 \text{ N.}$$

$\therefore$ Total force to be overcome

$= (1) + (2) + (3) + (4)$

$= 15941 + 1336.6 + 6376.5 + 534.6$

$= 24188.7$ N

$= 24.189$ kN.

Total work done per sec $= 24189\ N \times 3.33\ m$

$= 80549$ Joules.

$\therefore$ Power required by the rope $= 80549\ W$

$= 80.55\ kW$.

This would be the power in the rope when the loads move at full speed and it would decrease a little as the rope coils on the drum. It would be necessary to provide a motor of at least 50 % more power for the following reasons.

(a) No allowance has been made for the force required to accelerate the journey from rest.

(b) Some of the power developed by the motor is lost in the gearing between motor and the haulage drum.

(c) A reserve of power is always necessary in case the tubs derail and they have to be pulled up when the wheels are on the floor and coefficient of friction is high.

Example :

What is the least gradient for a self-acting incline 500 m long, assuming 8 tubs per train, each tub weighing 500 kgf and carrying 1000 kgf of coal ? The rope weighs 1 kgf/m.

Assume $\mu = \frac{1}{50}$ for tubs and $\mu = \frac{1}{10}$

Ans. :

Let the least gradient $= i = \frac{1}{n}$

i.e. 1 in n.

Now the force required to overcome frictional resistances is offered by the weight of coal coming down; considering number of loaded tubs = number of empty tubs i. e. the component along the incline due to gravity of the full tubs (= G) must be sufficient to overcome :

(a) The gradient resistance of the empty tubs (= g),

(b) The friction of the loaded tubs, empty tubs and the rope ($= F + f_1 + f$).

(c) Gradient resistance at the time of start of the rope drawing the empty train (g_1).

Where,

W = gross weight of full tubs (Newtons)

G = Gradient resistance of full tubs = wi (N)

w = weight of empty tubs (N)

g = gradient resistance of empty tubs = wi (N)

w_1 = weight of rope (N)

g_1 = gradient resistance of rope $= w_1 i$(N)

i = 1/n = gradient as 1 in n

F = Friction of full tubs $= \mu\ W$ (N)

μ = coefficient of friction of tubs

f = friction of empty tubs $= \mu w$ (N)

μ_1 = coefficient of friction of rope

f_1 = friction of rope $= \mu_1 w_1$ (N).

Then least gradient must be, when the following relation is satisfied.:

$$G = g + g_1 + F + f + f_1$$

or $$G - g - g_1 = F + f + f_1$$

or $$Wi - wi - w_1 i = F + f + f_1$$

or $$i = \frac{F + f + f_1}{W - w - w_1}$$

Now

$$W = 8 \times 1500 \times 9.81\ N = 117720\ N$$
$$w = 8 \times 500 \times 9.81\ N = 39240\ N$$
$$w_1 = 500 \times 1 \times 9.81\ N = 4905\ N$$

$$F = 117720 \times \frac{1}{50} = 2354.4\ N$$

$$f = 39240 \times \frac{1}{50} = 784.8\ N$$

$$f_1 = 4905 \times \frac{1}{10} = 490.5\ N$$

$$\therefore \quad i = \frac{2354.4 + 784.8 + 490.5}{117720 - 39240 - 4905}$$

$$= \frac{3629.7}{73575}$$

$$= 0.049$$

i.e. 49 m in 1 km

or 1 in 20.41.

A considerably steeper gradient would be required in practice to ensure efficient running.

Example :

An endless haulage operates on a roadway 800 m long dipping at 1 in 15 and draws 600 tonnes of coal a shift (1 shift = 7.5 hrs. effective time). The loads weigh 1500 kgf each and the empties 500 kgf. The rope weighs 3 kgf/m, tub friction is 1/50 and rope friction is 1/1. Speed of the rope is 4 km/hr. Estimate the power required at the surge wheel. The loads are pulled up the gradient.

Ans. :

In 7.5 hours, amount of coal drawn is 600 tonnes.

$\therefore$ In 1 hour amount of coal drawn is 80 tonnes.

80 tonnes of coal need 80 tubs.

Now, in 1 hour distance travelled is 4 km. i.e.

over 4000 m number of tubs distributed = 80

$\therefore$ over 800 m number of tubs distributed $= \frac{80 \times 800}{4000}$

= 16 tubs.

The forces which balance each other are :

i. The weight of tubs on either ropes.

ii. The weight of rope on loaded side and on empty side.

The forces to be overcome are as follows :

1. Component in the direction of incline due to force of gravity on coal content.

$$= \frac{16 \times 1000 \times 9.81}{15} = 10464 \text{ N.}$$

2. Friction of loads and empties :

$$\frac{16 \times 9810 \times 1.5 + 16 \times 9810 \times 0.5}{50}$$

$$= \frac{9810 \times 16 \times 2}{50}$$

$$= 6278.4 \text{ N.}$$

3. Friction of rope of both sides :

$$= \frac{3 \times 800 \times 9.81 \times 2}{10}$$

$$= 4708.8. \text{ N.}$$

Total force = (1) + (2) + (3) = 21451.2 N

work done/sec = Force × velocity

$$= 21451.2 \times \frac{10}{9} \frac{\text{Nm}}{\text{s}}$$

= 23834.67 Joules/sec

$\therefore$ Power = 23835 watts

or Power = 23.835 kW

Let us add 25% for the power required during acceleration.

$\therefore$ Power required at the surge wheel = 1.25 × 23.835

= 29.79 kW

say, 30 kW.

Example :

Calculate the motor power of an electric endless rope haulage to haul 350 tonnes in 7 hrs. effective hauling time, up an 1200 m long having a gradient of 1 in 12. An empty tub weighs 500 kgf and its carrying capacity is 1000 kgf of coal.

Ans. :

To find the motor power , we must know

(a) the tractive force to be exerted by the rope in N and

(b) the speed of the rope in n/s.

The product of the two will give the effective power exerted by the rope in N-m/s *i. e.* in watts. This is the power required at the drum.

Let speed of haulage = 4 km/hr $= \frac{10}{9}$ m /s

= 1.11 m /s.

Let tub friction = $\mu = \frac{1}{50}$

and rope friction $= \mu_1 = \frac{1}{10}$

Coal on rope $= \frac{350}{7} = 50$ te delivered in 1 hr.

This is delivered by 4 km of rope.

$\therefore$ Coal on rope of 1200 m $= \frac{50 \times 1200}{4000}$

= 15 te.

$\therefore$ Number of loaded tubs (15) = No. of empty tubs (15)

Size of rope :

Maximum tension in the rope is due to forces acting on loaded side, namely :

(a) Gradient resistance of the full tubs + frictional resistance of full tubs = G + F.

(b) Gradient resistance of the rope on load side + frictional resistance of ascending part of rope $= g_1 + f_1$.

Part (b) cannot be calculated readily until we know the size and weight of the rope, but part (a) forms by far the greater proportion of the total tension. We can, therefore, neglect (b) for the moment, merely adding a suitable allowance later; then;

Part (a) *i.e.* approximate tension in full rope

$$= \frac{15 \text{ tubs} \times 1.5 \times 9810}{12} + \frac{15 \times 1.5\ 9810}{50}$$

= 18393.75 + 4414.5

= 22808.25 N

= 22.81 kN.

Adding 20% extra for part (b).

maximum tension in rope = 27.372 kN

Let factor of safety be 7

$\therefore$ Breaking strength = 7 × 27.372 kN

= 191.604 kN

For a round stranded rope with fibre core, the formula for breaking strength is

B = sd² kN

Where B is the breaking strength in kN ;

s is a constant = 50 kN/cm²

and d = diameter in cm.

(Assuming the steel used having a breaking strength of 160 kgf/mm² *i.e.* 1570 MN/m², a typical value)

191.604 kN = 50 kN/cm² × d² cm²

or $d = \sqrt{\frac{191.604}{50}}$ cm

$= \sqrt{3.84}$ cm

= 1.96 cm, i.e. 2 cm (nearest size)

mass of the rope is given by the formula

mass in kg/m = kd²

where d = diameter in cm

k = 0.35 for round strand rope with fibre core.

$\therefore$ mass per meter length = 0.35 × 3,84 kg

= 1.344 kg

and wt. per meter length of rope = 1.344 × 9.81 N

Power of motor :

In endless rope haulage the gravity effect of the rope and tubs on the load side and empty side is balanced and thus the tractive force to be exerted by the rope comprises.

(a) Gradient resistance of coal alone = G_1 (coal)

(b) Friction of both full and empty tubs – F + f

(c) Friction of the whole of the rope = 2 f_1

Tractive force required = G_1 (coal) + F +f + $2f_1$

$$= \frac{15\times1.0\times9810}{12} + \frac{15\times1.5\times9810}{50} + \frac{15\times0.5\times9810}{50}$$

$$+ \frac{2\times1200\times1.344\times9.81}{10}$$

= 12262.5 + 4414.5 + 1471.5 + 3164.31 N

= 21312.8 N

= 21.313 kN.

Power required by the rope = power at the surge wheel

= Force × Velocity

$$= 21.313\times\frac{10}{9}\text{ kW}$$

= 23.68 kW

Assume 75 % transmission effciency

∴ Motor power = 31.573 kW.

QUESTIONS

1. An incline mine has to be equipped to produce 600 te/day, the average gradient of the seam is 1 in 20, the length of the haulage plane is 2000 meters. Given, rope friction is 1/20 and tub friction is 1/50, tub capacity is 1 tonne, and the tare is $\frac{1}{2}$ of the coal it carries Speed of rope is 4 km per hour. Net working time of the haulage is 6 hours per shift. Static factor of safety of the rope is 6. Ultimate tensile strength of rope is 160 kg/mm². Find out the H. P. of the haulage.

2. Calculate the size of rope and the drum diameter for a direct haulage to haul 300 te in 6 hrs. effective hauling time up an incline 1000 m long having a gradient of 1 in 8. Assume each tub to weigh 0.5 tonne and to carry 1.0 tonne of coal. Speed of the haulage may be taken as 12 km/h.

3. A direct haulage plane is 500 m long and dips at 1 in 4. Coal tubs are hauled up the road in sets of 8, using a rope of 18 mm diameter. The tub has a tare of 275 kgf and a capacity of 650 kgf of coal. What should be the weight of a rope, round stranded with fibre core for the duty? What should be the diameter of the haulage drum? Calculate the power of an electric motor to work the haulage. Maximum speed of rope is 10 km/h.

O O O

CHAPTER – 16

TRANSPORT : INTERNAL COMBUSTION ENGINES AND LOCOMOTIVES

The machinery used and most of the industries are driven by :

1. *Steam,*
2. *Electricity,*
3. *Petrol engines,*
4. *Diesel engines,*
5. *Compressed air,*
6. *Hydraulic power,*
7. *Gas engines.*

Engines powered by petrol, and gas come in the category of internal combustion engines. In an internal combustion engine, fuel is burnt directly inside cylinders. The combustion of fuel produces a large volume of gases at high pressure and temperature and the pressure is applied to drive the piston in the cylinder. The linear motion of the piston is used to rotate a crank shaft by means of connecting rod or other gear. A steam engine, on the other hand, is an external combustion engine for which the fuel (coal, wood or oil) is burnt in apparatus external to the engine. Combustion of fuel produces a large volume of steam under pressure which is utilised in a piston-type engine or turbine. The steam engine or turbine, therefore needs furnaces, boilers and large water supply. Moreover, transmission of steam is possible only for short distances. It has to be used hot and the efficiency of steam engine is low compared to that of an internal combustion engine.

Only the petrol and diesel engines will be described in this chapter.

The petrol and diesel engines are mostly straight-line engines with vertical cylinders. They have many features in common. Each of these burns a mixture of air and finely sprayed fuel (petrol or diesel) in a closed cylinder. The combustion produces instantaneously large volume of gases at high pressure. The resulting high temperature further increases the pressure. The result is a thrusting force which pushes the piston, as stated earlier, is converted into rotary motion of a crank-shaft. The engines convert the heat energy of fuel into mechanical energy.

Four-stroke engine :

In the large power units used on cars, trucks, locomotives and other heavy equipments, the engines are of four-stroke type i.e. out of the four strokes of the piston completed in two revolutions of the crankshaft, only one stroke is the power stroke or working stroke and the force of gases resulting from combustion of airfuel mixture is transmitted by the movement of piston to the crankshaft during that power stroke. Part of the energy of power stroke is transferred to a flywheel mounted on the crankshaft and energy stored in the flywheel as well as momentum of moving parts keep the crankshaft turning during the remaining three strokes.

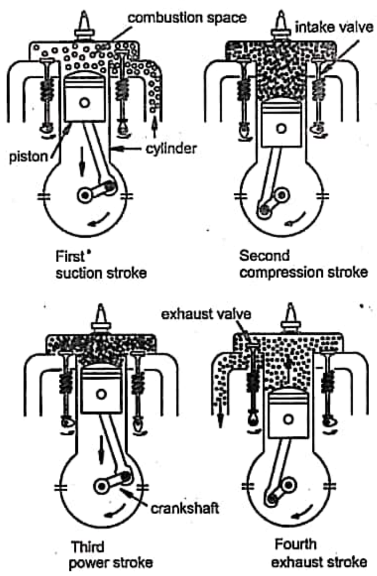


Fig. 16.1 A four-stroke petrol engine.
Spark plug is shown at the top of the cylinder.

In the **four-stroke petrol engine, the cycle** is as follows :

1. *Suction stroke or induction stroke* :

It is the downward stroke. An inlet valve opens just as the piston commences its downward movement from top dead centre (T.D.C.) and during the stroke, atmospheric air is sucked in throughout the stroke. The atmospheric air, as it passes towards the inlet valve, creates a suction effect which inducts petrol via the carburettor in the form of mist so that air mixed with petrol vapour enters the cylinder. Exhaust valve remains closed throughout the stroke. Petrol : air mixture in the ratio of 1 :15 is highly explosive.

2. *Compression stroke* :

It is upward stroke commencing from bottom dead centre (B.D.C.). As the piston commences its upward movement the inlet valve closes, the exahust valve continues to remain closed and the air-fuel mixture is compressed. In ordinary petrol engines the compression ratio is usually about 6 :1 but in petrol engines of sports cars, it is between 7:1 and 9 : 1 and the pressure at the end of compression is about 6.5 to 8.5 kgf/cm² with full throttle opening.

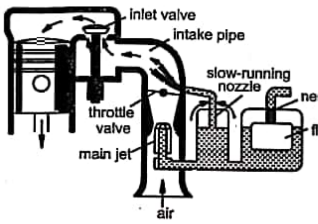


Fig. 16.2. Carburettor (schematic) and petrol flow.

. Compression ratio = Volume swept by the piston in full travel + clearance volume

When the piston is at the top of the stroke i.e. on the top dead centre (T.D.C.), an electric spark jumps across the electrodes of a spark plug. Very often the spark takes place very slightly before the piston reaches T. D. C. It is necessary to advance the moment of ignition to provide against the time lag in getting the mixture fully burnt by the time the piston is ready to start on downward stroke under the impact of pressure of gases formed by combustion The electric spark is timed through a separate electrical accessory such as a magneto or a distributor. The spark causes combustion of the air-fuel mixture resulting in large volume of gases under pressure which rises further due to high temperature.

3. *Power stroke* :

The gases of combustion expand and the sudden rise of pressure forces the piston on its downward travel. Both the suction and exhaust valves remain closed. The power stroke immediately follows the ignition caused by the spark.

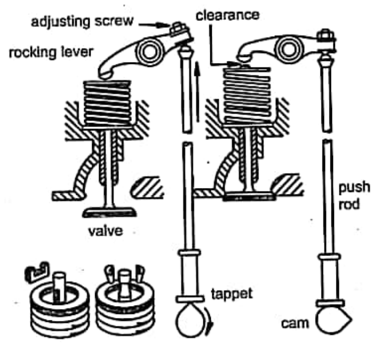


Fig. 16.3. Valve and tappet arrangement in a 4-stroke engine.

4. *Exhaust stroke* :

As the piston starts its upward travel, the exhaust valve is opened, the inlet valve remaining closed and the gases of combustion escape to the atmosphere through the exhaust pipe and the silencer.

These four strokes constitute a cycle which is completed in two revolutions of the crankshaft and during the two revolutions power is imparted to crankshaft only during $\frac{1}{2}$ revolution yielding some-what uneven rotation. To attain smoothness in motion and performance, a heavy flywheel is quite effective. Four, six or eight cylinders cast in one block are commonly used in an engine for increasing power output. Such arrangement results in a regular turning effect (torque) to the crankshaft and also obviates the use of a flywheel ; or if one is used, it need be small. The sequence in which electric spark occurs in different cylinders is called firing order which is usually as follows depending upon crank-shaft designs

4 – cyl. in-line engines : 1–3–4–2. or
1–2–4–3

6 – cyl. in-line engines : 1–5–3–6–2–4 or
1–2–4–6–5–3

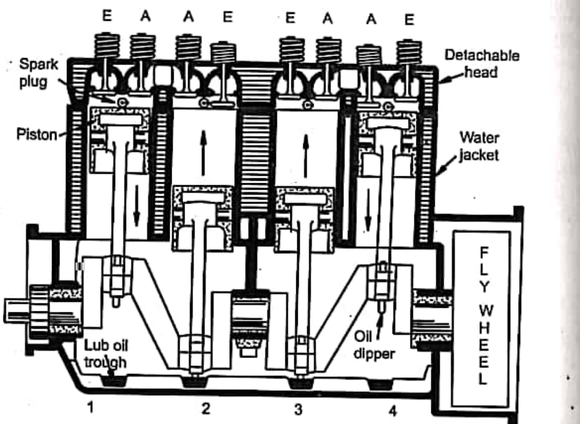


Fig 16.4. A-admission or inlet valve.

E - Exhaust valve.Operations in different cylinders of a 4-cylinder petrol engine having 4-stroke cycle. The figure shows beginning of :

1. *power stroke* 2. *exhaust stroke* 3. *compression stroke*
4. *suction stroke* *Firing order 1–3–4–2.*

A 4-stroke disel engine also completes its cycle of operations in four stroke viz.suction stroke, compression stroke power stroke and exhust stroke, but differs from the petrol engine in a few respects : it has no spark plug ; it sucks only atmospheric air during the suction stroke, and at the end of compression stroke, a metered quantity of diesel oil is injected at high pressure in the cylinder in the form of atomised spray. The high temperature of the compressed air ignites the fuel spray. The high temperature of the compressed air ignites the fuel spray producing gases and combustion results in further rise of temperature and heavy pressure which forces the piston on its downward stroke (power stroke). The compression ratio is generally 14 : 1 but may be 18 : 1 and even 20 : 1 in some engines. The pressure at the end of

compression is nearly 30 kgf/cm^2 and the heat of compression raises the air temperature to nearly 500°C. This high temperature is sufficient to ignite diesel or other fuel oil which has self ignition temperature between 350°C and 450°C in air at atmospheric pressure, but lower at higher pressures. As the ignition of fuel is caused by compressional heat, the diesel engines is also known as a compression-ignition engine. The pressure in cylinder after the combustion is nearly 45 kgf/cm^2 and the high temperatures and pressure resulting in diesel engines, demand robust construction from high quality materials. For this reason, a diesel engine is costlier than a petrol engine but it is more efficient. Moreover, diesel it cheaper than petrol compared to a petrol engine is more noisy and with slightly more vibrations.

Firing order, when referred, to a diesel engine, is the sequence in which deselunder pressure is injected to different cylinders at the end of compression stroke. The firing order is the same as stated for petrol engine On the Cummins engines, the order is as follows :

Cummins engine, st. line 6-cyl. (right hand rotation) : 1–5–3–6–2–4.

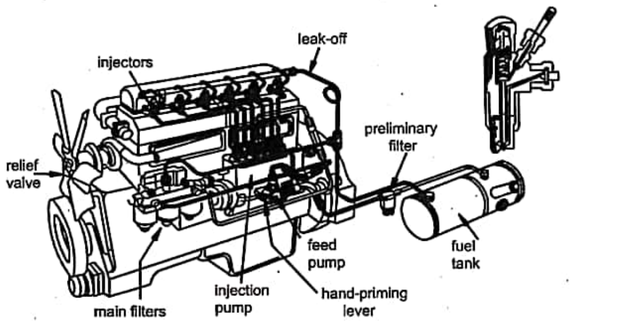


Fig. 16.5. Fuel supply in a disel engine (schematic).
Top right-injection nozzle.

V–type Cummins engines. (Right hand rotation) ;
V–8 engines : 1–5–4–8–6–3–7–2.
V–6 engines : 1–4–2–5–3–6.

On some 6-cyl, in-line engines of other manufacturers the firing orders is : 1–2–4–6–5–3.

Two-stroke engine :

In a 2-stroke engine, the operations of suction, compression, expansion and exhaust are completed in two strokes. The crank-shaft, therefore, receives a power thrust once in each revolution. One may expect that size for size, a two stroke engine developes twice the h.p. available from a 4-stroke engine, but this is not the case due to exigencies of design and the increase of power in a 2-stroke engine may be only from 10-30 %. A two-stroke engine has no valves, but has ports cut in the cylinder wall. Compared to a 4-stroke engine, it is mechanically simpler, easier to manufacture, gives a smoother turning effect to the crankshaft, and needs a less heavy flywheel.

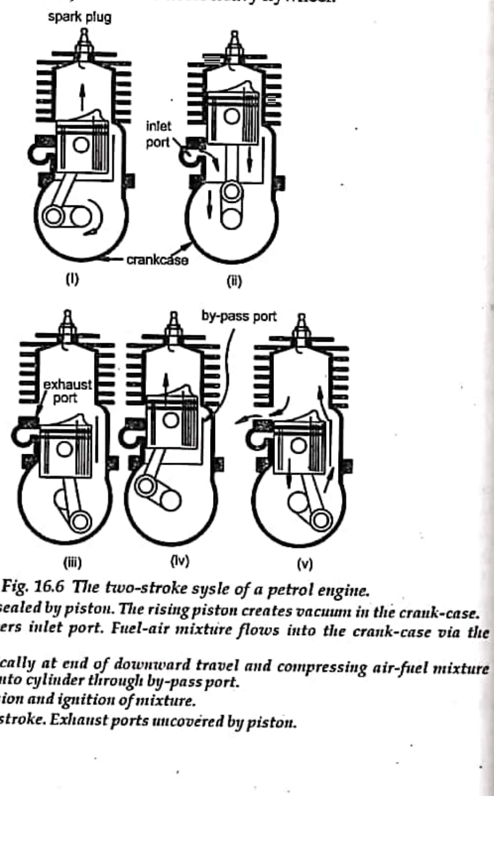


Fig. 16.6 The two-stroke sysle of a petrol engine.

i. All ports are sealed by piston. The rising piston creates vacuum in the crank-case.
ii. Piston uncovers inlet port. Fuel-air mixture flows into the crank-case via the carburettor,
iii. Piston practically at end of downward travel and compressing air-fuel mixture which flows into cylinder through by-pass port.
iv. Full compression and ignition of mixture.
v. End of power stroke. Exhaust ports uncovered by piston.

Fig.16.6 illustrates the working of a 2-stroke petrol engine. The piston top (crown), it may be noted, has a special shape. The inlet port remains open partly during the upward stroke and partly vapour (via the carburettor) and air, not into the cylinder, but into the crankcase. The descending piston, near the end of its downward stroke, closes the inlet port and the petrol-airmixture is pushed from the crankcase into the cylinder via a bypass port. The by-pass port is uncovered slightly later than the exhaust port. Now the upward stroke. The piston, during its upward stroke, first closes the by-pass port and then the exhaust port. Further upward movement of the piston compresses the fuel-air mixture and at the same time, since inlet port is open, air is sucked in. Ignition spark takes place when the piston is at top dead centre. The resultant combustion and expansion of gases thrusts the piston on its downward travel, imparting power to the crankshaft. The descending piston uncovers the exhaust port and the burnt gases exhaust to the atmosphere. The cycle is thus repeated.

The typical piston shape prevents the fuel-air mixture from passing straight out through the exhaust port since the mixture is deflected up into the cylinder. During the upward stroke, the piston shape assists in scavenging out any burnt out gases which might have remained in the cylinder . In the scavenging process it is unavoidable that part of the compressed fuel-air mixture enters the exhaust. This results in slightly increased fuel consumption as compared to that in a 4-stroke engine.

There is no oil in the crankcase. Engine lubrication is achieved by adding lubricating oil to the fuel (about 25 litres fuel and 1 litre oil). There are no valves and therefore no need for cam-shaft, tappets, etc. The piston rings are kept in their position by lugs ; otherwise, the ends (joints) of the piston rings may expand into the ports and cause serious damage. The big-end bearing in a 2-stroke engine, a two-stroke engine requires more frequent decarbonising of cylinder top, combustion chamber, exhaust port and mainfold, and by-pass port.

The two stroke disel engine works in the same way as a petrol engine with the difference that only air sucked in instead of air-fuel mixture and there is no spark of ignition but injection of a metered quantity of diesel under pressure at the end of compression stroke. In a diesel engine there is not as much loss of fuel during the scavenging action as in a petrol engine, because only air is sucked in and not fuel-air mixture.

The operations in a 4-stroke engine viz. suction, compression, power and exhaust, are well demarcated but in a 2-stroke engine, they partially overlap one another. The two stroke engine, petrol as well as the diesel one, is not as efficient as the four-stroke engine and its use is limited to engines of small horsepower, e.g. for motor cycles.

Fuel Supply :

In petrol engine a fuel pump either electrically operated, or mechanically operated by a cam in the engine, sucks petrol from the petrol tank and delivers it to a carburettor which feeds it to the cylinders. In a diesel engine a mechanically operated fuel pump sucks diesel from the diesel tank and feeds it to an injection pump supply by gravity feed is sometimes adopted for stationary engines but not on high h. p. engines of moving machines.

Starting an engine :

Small engines : On small petrol engines such as tyhose used on a motor cycle, scooter, etc., the ignition device is a magneto. The magneto has a small permanent magnet dynamo having a circuit breaker incorporated into its primary winding. A secondary coil induces high voltage impulses when the primary circuit is broken and a high voltage passes to a spark plug in the cylinder head. The spark is timed at the end of compression stroke.

To start the engine the crankshaft is rotated :

(a) through gears when a kick is given with the foot, as in a scooter, or with a handle as in a 3-wheeler auto-rickshaw, or

(b) by a cranking handle, or

(c) by smartly pulling a cord coiled 2-3 times on the circumference of a small pulley or fly-wheel attached to a crank shaft.

The medium size petrol engine such as the one used on a motor car or truck is usually provided with high tension coil, battery, distributor and a starter motor. The distributor distributes the high tension current from the H.T. induction coil to the spark plugs in a definite sequence of the firing order. The battery (6,12 or 24 volts) is kept charged by a belt-driven dynamo. The operator first switches on the ignition (indicated by a tell-tale lamp, usually red) and then pulls or pushes a spring-loaded switch that supplies current to the starter motor which is thereby set in motion. As the starter motor rotates, a movable pinion on its shaft engages with the flywheel of the engine and rotates it. When the engine starts running, the starter switch should be put off. This causes the pinion to move away from its meshing position as the starter motor slows down and comes to rest. The starter motor slows down and comes to rest. The starter motor should not be allowed to run a moment longer than is necessary to start the engine.

Small diesel engines are usually not equipped with a dynamo or a battery and can be started by a cranking handle. Medium as well as large sized diesel engines are provided with a storage battery, dynamo and a starter motor.

Due to the very high compression in diesel engines, a compression release lever forms part of the starting device on a medium and large sized diesel engines. Pulling this lever lifts the inlet or exhaust valve (depending on engine model) so that the piston does not have to work against high pressure in the cylinder. After initial rotation of the crankshaft, the compression release lever is released from the hand. The engine then starts without much difficulty. The lever has to be used only as an in cranking before starting and it should not be used to stop the engine.

To aid in starting a diesel engine, when the atmospheric temperature is low (10^0C or below) an intake-air pre-heater is used. The pre-heater equipment consists of *(a)* hand priming pump to pump fuel into intake manifold and *(b)* switch to turn on a glow plug which is electrically heated by current from a battery. Fuel burns in the intake manifold and heats the intake air.

Large H. P. stationary diesel engines are started by admitting compressed air at about 16 kgf/cm^2 from a special receiver into the cylinder of the diesel engine. By suitable air-starting valve arrangement, the intake manifold is utilised for such admission and the compressed air sets the pistons in motion. Compressed air is not admitted to all the cylinders but generally to only 3 cylinders of a six-cylinder engine. With the pistons in motion, the diesel injection is allowed to take place and the engine starts in the normal way. When the engine starts running, the air-starting valve lever is moved to the "run" position : this stops the admission of compressed air and permits atmospheric air (or air from super-charger/turbocharger, if provided) to enter the intake manifold. The compressed air, the receiver is filled, lasts for 6 to 7 starts.

Heavy duty diesel engines are sometimes started by an auxiliary petrol engine, To start it, the main diesel engine declutched, the auxiliary petrol engine mounted on the same chassis is started with the help of a battery and starter-motor, and when the petrol engine attains full speed, it is engaged with the flywheel of the main diesel engine is started, the petrol engine is declutched and stopped.

A simple method to start a diesel engine is by firing a powder cartridge but the method is getting obsolete.

In a 2-stroke diesel engine, the compression ratio is not as high as in a 4-stroke engine. The ignition temperature is therefore difficult to attain after initial few rotations of the crankshaft. The ignition chamber on the cylinder head is therefore heated by a blow lamp or by firing a powder cartridge into the chamber. When it is warmed up, a few rotations of the crankshaft by cranking handle starts the engine.

To shut down a diesel engine : It is important to idle the engine 3–5 minutes before shutting it down to allow lubricating oil and water to carry heat away from the combustion chamber, charged engines. The turbocharger contains bearings and seals that are subject to the high heat of combustion exhaust gases. While the engine is running, this heat is carried away by oil circulation, but if the engine is stopped suddenly, the turbo-charger temperature may rise as much as 56° C. The results of extreme heat may be seized bearings or loose oil seals.

For rotary internal combustion engines, see appendix.

LOCOMOTIVE TRANSPORT :

A locomotive haulage can be used in a mine :

1. Where the gradient of the road-way is mild. Nearly flat gradient is preferred. A gradient of 1 in 15 against the loads is considered to be limit though locos are generally employed on gradients milder than 1 in 25.
2. Where the loco track is in settled ground not subjected to movement by mining operations.
3. In the intake air-ways, where air velocity is adequate to keep fire-damp percentage apriciably low. If diesel locos are used the exhaust gases of the loco should be diluted by the air current sufficiently well so as to be unharmful to the workers.
4. Where roads are reasonably wide and high.
5. Where the transport of mine cars involves long haul distances. Small locos for shunting and marshalling in the pit bottom are not uncommon.

Locomotives used in mines range from light weight type (2 tef to $4\frac{1}{2}$ tef weight) to heavy duty types (8 to 13 tef weight). Units of 30 to 75-kW are considered as heavy duty locos and are used for main haul roads. A 75-kw diesel loco weighs nearly 15 tef. The designs of locos are such that the total weight supported by each axle is 5 te or less. Two 75-kW locomotives can be coupled in tandem to provide one 150-kW unit.

Every loco consists of :

1. a chassis which is a rigid frame work of rolled steel sections.
2. Driving wheels (traction wheels) on axles, springs, and brake blocks mounted underneath the chassis.

3. A power unit. This is a diesel engine, electric D. C.motor or compressed air motor, mounted on the chassis. Petrol engines are not permitted by alaw in underground mines as they produce a large amount of carbon monoxide.
4. Operator's cabin equipped with controls, brake operating system, sand boxes, horn.
5. On medium and large size locos, an air compressor for powered brakes.
6. Lights at both ends.
7. A hand screw brake for emergency, as required by law.

All locos are provided with brakes on each of the wheels and these are operated by the loco driver by a lever in his cabin. The wheels are of cast steel. The tyres of the wheels are of steel and are removeable to allow renewal. The brake blocks are of cast iron and act on the tread or the wheel. To improve braking effort, sanding (*i.e.*, spreading of sand) of rails of the leading wheels in either direction of travel is a standard provision on all locos. Sand boxes and feeding arrangements are provided for the purpose on a loco and the arrangement is controlled by the loco operator by a pedal from his cabin. Brakes are power hand operated brake is always required under the regulations as a parking brake.

Before describing different types of locos principle governing traction by locomotives need to be appreciated.

In the case of rope haulages and conveyor transport, the power to move the load is available from fixed motors external to the haulage or conveyor. The sizes of these motors can be varied to match the duty requirements. In the case of a locomotive haulage, however, the driving unit i.e. the locomotive provides the tractive effort and such loco moves along with the train of mine cars to which it is coupled.

Tractive force or tractive effort is the force required to cause movement, and the tractive effort depends on the weight of the loco and also on the frictional adhesion between the locomotive's driving wheels and the rail track. The hauling or tractive effort generated by the engine/motor of the loco is therefore limited and is used up partly in moving and accelerating the loco itself and only the remainder is available for pulling the train of mine cars through the medium of draw bar and accelerating them. The *co-efficient of adhesion* is the co-efficient of *static* friction, μ, between the wheels of a loco and the rails. If W is the total weight of the loco bearing on the driving wheels, μW is the total tractive effort exerted at the driving wheel treads.

The value of μ depends on the condition of the two surfaces in contact, *i.e.* the rails and the wheels. When the surfaces are dry the value is higher than the when wet, or covered with oil or slime. Sand or grit increases the value. Some average values of co-efficient of static friction are as follows

Surface Condition	Free of sand	Sanded
Dry	0.25	0.28–0.35
Wet	0.20	0.25–0.30
Slimy	0.15	0.22–0.25

Usually the value of co-efficient of adhesion (co-efficient of static friction) is taken as 0.2 to 0.25 but a lower value, about 0.16, is used when braking is considered as co-efficient of friction is less when the loco is running.

It will be seen that the theoretical maximum tractive effort is only $\frac{1}{4}$ or1/5th of the total weight of loco. Moreover, this is possible only if the weight of the loco is distributed equally over all the wheels and if the drive is transmitted through all those wheels. The tractive effort varies with the speed of the train ; it is more with low speeds and less with high speeds.

Example :

A locomotives weighs 15 tonnef and the adhesion to the tracks is 2246 N per tonnef. (a) What is the co-efficient of adhesion, and (b) what is the draw-bar pull which the locomotive is capable of exerting on (i) a level track, (ii) an adverse gradient of 1 in 100, and (iii) an adverse gradient of 1 in 5. Assume that the running resistance of the locomotive is 67 N per tonnefe.

Ans. :

(a) Coefficient of adhesion $= \dfrac{2246\text{ N}}{9810\text{ N}} = 0.229$

New tractive effort = 15 × 2246 N = 33690 N

Running resistance = 15 × 67 N =1005 N

Gradient resistance of loco *i.e.*, component of loco weight parallel to the track

i. on the level gradient = O

ii. on 1 in 100 $= 15 \times \dfrac{9810}{100}$ N

$= 1471.5$ N

iii. on 1 in 5 $= \dfrac{15 \times 9810}{5}$ N = 29430 N

Draw-bar pull :

In case *i.* 33690 – 1005 = 32685 N

ii. 33690 – 1005 – 1471.5= 31213.5 N

iii. 33690 – 1005 – 29430 = 3255 N.

Actually the tractive effort exerted on a gradient will be somewhat less than 33690 N, depending on the cosine of angle of inclination, but for the nearly flat inclination, this factor can be ignored. In case (iii) however, it will be seen that the locomotives can do little more than pull its own weight.

Example :

A locomotive and train have a total mass of 600 metric tonnes. The resistance opposing motion can be assumed to be constant, and amounts to 60 N/tonne. If the locomotive can exert a constant pull of 120 kN find how long it will take to accelerate the train from rest to 84 km/h on level track.

Ans. :

First change the units to a consistent set in terms of m, kg, S & N.

600 metric tonnes = 600 × 1000 kg = 600.000 kg.

120 kN =120 Kilonewtons = 120,000 N

$$84\text{ km/h} = 84 \times \frac{5}{18} = \frac{210}{9}\text{ m/s}$$

Frictional opposing force = 60 N/tonne × 600 tonnes

= 36000 N

Accelerating force = Pulling force - Resisting force

= 1,20,000 N - 36,000 N

= 84,000 N

$$P = M \times f \qquad \therefore\ f = \frac{P}{M}$$

(f will be in m /s² if P is in Newtons and M is in kg)

$$f = \frac{84,000}{600,000} = 0.14 \text{ m/s}^2$$

For a body starting from rest , $v = f\,t$ or $t = \frac{v}{f}$

$$t = \frac{210}{9 \times 0.14}\,\frac{m}{s} \times \frac{s^2}{m} = \frac{500}{3} = 166.7 \text{ s}$$

Ans. Time to accelerate = 167s.

The resistance of the locomotive itself (and of the mine cars, if any, attached to it) for running arises out of the friction caused at the wheel bearings (and friction against wind which is considered negligible). At starting the co-efficient of friction on this count is taken as 0.01 for the loco and the attached train ; when the locomotive is running, the value is taken as 0.0025. This running co-efficient of friction at the wheel bearings assists the loco to slow down and therefore its value its to be considered during braking. The resistance to motion of the loco itself when *in motion* is called its rolling resistance.

Optimum gradient for a locomotive haulage :

A realistic gradient to use for the full effectiveness of locomotives is the gradient at which the same size of train can be started and safety stopped under emergency braking conditions. The braking duty required is usually specified as a stopping distance at a particular speed, the full stopping distance being the distance ahead of the locomotive driver which can be seen without obstructions. Some allowance for "thinking time" and delay in applying brakes must also be made. The gradient is about 1 in 200 to 1 in 400 against the load trains and only the loaded train need be considered as braking and starting duties are not as severe with empty wagons/cars as those for the loaded train.

Diesel locomotives :

These are commonly used in a number of mines. Their weight ranges from 3 te to 15 te and the power from 15 kW to 75 kW. The power unit is a diesel engine with 2, 3 or 4 cylinders of 4-stroke cycle, compression-ignition type. Heavy duty locos are of six cylinders. Locos used in underground coal mines have the power unit in a flame proof enclosure as a safeguard against ignition of fire damp. The intake air going to the engine passes first through a

filter and then through a flame trap. Similar flame trap is fitted on the exhaust side of a diesel engine. A flame trap consists of a number of stainless steel plates contained within a stainless steel housing. The plates are 50 mm wide and welded into position with gaps of — mm between adjacent plates. The exhaust flame trap can be easily removed from its housing and it has to be throughly cleaned everyday. On the exhaust side the hot exhaust gases of the engine pass through an exhaust conditioner before entering the flame trap. These exhaust gases should have very low percentage of CO and other noxious and poisonous fumes before they enter the mine atmosphere of restricted airways. The diesel combustion has therefore to be satisfactory, and diesel oil should have a flash point of not less than 65 °C. The maximum permitted percentage of CO in the exhaust gases before they enter the mine atmosphere is 0.2 % but usually it is between 0.02 and 0.04 %. In coal mines diesel locomotives are not allowed to be used where the percentage of inflammable gases is more than 1.25 in the general body of the air. Their use is, therefore, confined to intake airways where large volume of air flows .The other gases contained in the exhaust include oxygen, nitrogen, carbon dioxide and small quantities of the oxides of sulphur and nitrogen mixed with certain organic compounds known as aldehydes which smell abominably and cause irritation of the nose, throat, and eyes. To remove these last-mentioned oxides and aldehydes, mine locomotives are fitted with an exhaust conditioner.

Exhaust conditioner : The principle of this is shown in Fig. 16.7 but the details of design vary with different makes and are subject from the engine development as time goes on. The exhaust gases from the engine, amounting in all to about 0.085 m³ per B. H. P. per minute, are conducted to the bottom of the conditioning chamber, A, and impinge on the surface of the water in the base. This traps hot particles and washes out the sulphur and nitrogen oxides and aldehydes.

The gases then rise through a flame-proof slag wool filtering medium kept moist by the evaportaion of the water, and thereafter pass into a second similar chamber, B, where they are further cooled and filtered before passing through the flame arrester. This consists, as at the inlet, of a grill of removable stainless steel plates, $\frac{1}{2}$ mm apart. Finally, the gases are mixed with about 30 to 40 times their volume of fresh air before being exhausted into the ventilating current.

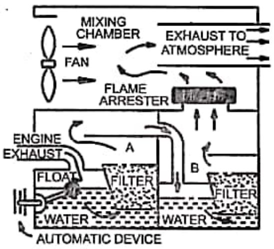


Fig. 16.7. Exhaust conditioner and flame trap used in u/g diesel locos

The filtering material and the flame grids are readily removable and must be replaced by a clean set every-24 hours. If the water is allowed to fall below a certain level, the fuel is automatically cut off from the engine and the brake applied.

The exhaust smell may mask the odour of spontaneous combustion and in mines where the coal is liable to spontaneous beating, the diesel locomotives should be avoided.

Electric battery locomotive :

The power unit of an electric battery locomotive is a D. C. electric motor receiving its current from a storage battery carried in a casing on the upper part of the chassis. Such locomotives are for light and medium duties as they are generally less powerful than diesel or trolley wire locomotives, though battery locos of even 13 tonnes weight are available in the country. Range of battery locomotives is from 4 to 70 kW continuous rating. The battery locomotive is relatively quiet in operation and produces no objectionable fumes. Compared with the diesel locomotive it generates much less heat. An important advantage of battery locomotive is that it can meet an appreciable overload of short duration. The battery constitutes a major portion (nearly 60 %) of the weight of the locomotive. Usually there are two batteries on a loco. The batteries are of lead acid type and each battery consists of a number of 2 - volt cells, their number varying from40-70. The battery cannot be made flamproof and its container has to be well ventilated. A fully charged battery gives service for nearly 8 hours i. e. one shift of regular traction duty. At the end of the shift, the battery has to be placed on a charging rack and it takes nearly 8 hours to fully

charge. It should be borne in mind that a fully charged battery can be discharged in a few hours only by overload or battery can be discharged in a few hours only by overload or mis-use. But to replanish the charge, it takes nearly 8 hours. The battery charging station layout is given in Fig.16.8. By a lifting tackle the nearly discharged battery of a loco is removed and placed on the charging bays at the end of the shift and a fully charged battery from the charging station replaces it. The direct current for charging at the station may be available from a motor generator set or by the use of mercury arc rectifier. The latter has the advantage that it has no moving or rubbing parts. The battery charging station should be close to an intake airway.

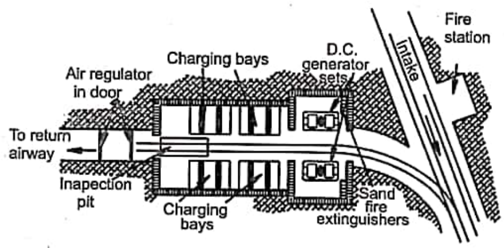


Fig. 16.8. Battery charging room layout.

*Travelling crane supported from roof gi *der changes the batteries.*

Overhead wire locomotive (or trolley wire locomotive) :

The trolley wire locomotive is equipped with electric motor fed with current from overhead electric wire through a pantagraph or through a long pole which is kept pressed against the overhead conductor by spring tension. Only direct current is supplied to the overhead wires though in some foreign countries A. C. is permitted. The main advantage of A. C. is that conversion equipment is not required between the supply mains and the overhead wires. The shock hazards are, however, much more serious with A. C. An important advantage of D. C. for traction is that the D. C. series motor is unrivalled for traction duty. The D. C. supply to overhead wires is at 250 volts. Trolley wire locomotives are used in a number of coal mines near Kurasia colliery and a few other coal mines of degree-I gassiness though the D. G. M. S. office is generally conservative in granting permission for their introduction in underground coal mine.

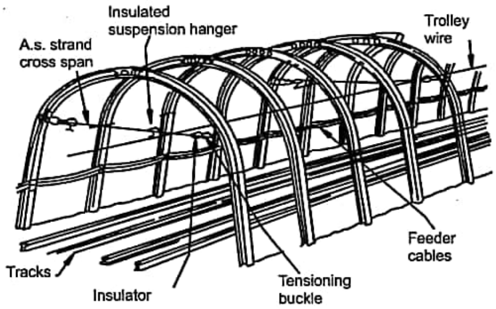


Fig. 16.9. Trolley wire for trolley wire loco.

The bare overhead conductors are of hard drawn copper wire suspended centrally over the track at a height of more than 2 m. The conductors are suspended through insulators from short cross wires of mild steel. An earth leakage wire is connected to each cross-wire. The rail track forms the return path for electric supply circuit and therefore the former must be suitably bridge at each rail joint by copper conductors. Section isolation switches for isolating parts of roadways have to be sited in easily accessible positons in the roadsides. The roadways for trolley wire locomotive should be sufficiently high and wide to provide safe clearances, and the ground free from any movement arising out of mining operations. The roadways have to be equipped with overhead wires and their support system. Branch roads cannot be negotiated unless they are also so equipped. These requirements, therfore, impose some restrictions on the flexibility trolley wire loco. Locos are taken to the face by feeding power through a cable reel from the terminus of the trolley wire line. The hazards of shock to workers through contact with bare wires and the possibility of explosion of fire damp in gassy coal mines due to sparking should not be ignored, though mining regulations are quite stringent in this respect. Such locomotives are used on a wide scale in Ruhr Coalfields (West Germany) in deep gassy mines and also in American underground coal mines.

The trolley wire loco system has the folllowing **advantages :**

1. High efficiency : Of all the different types of locomotives used in mines, trolley wire loco is the most efficient.
2. High overload capacity : For short periods, specially during peak loading activity, overloading of the motors do not pose any problem.

3. Simple maintenance : Most of the skilled work is to be done in the power house.
4. High power/weight ratio : The motor speed can be easily increased to give more tractive effort.
5. Reliability : It is robust in construction and not liable to breakdown.
6. Good control : It gives smooth acceleration and high torque.

Cable reels :

Cable reels are used (1) to enable a trolley loco to operate over a short distance beyond the terminal point of overhead conductors, (2) in the case of battery locomotives, for use at points where there is a lot of starting, stopping, shunting and collection of load. The cable reel is carried on the loco and the cable end is brought in contact with overhead wires by a long insulated hook, or alternatively, the cable end is plugged into a special socket of mains supply. As loco travels forward it uncoils the cable which then rests in the middle of the track; when returning, the reel is rotated by power and the cable is wound up on it.

Traction characteristics of D. C. motors :

It is a standard practice to provide two electric motors on storage battery locomotives as well as on trolley wire locomotives. As the current is D. C. ,series motors are preferred (field winding in series with armature or rotor winding) for the following advantages :

1. Series motor has a high starting torque. The armature current is large at the start and so also is the field current, being in series :
2. Current taken by the motor adjusts itself to the external load and the torque rises as the speed decreases.
3. Speed falls off as the torque for traction increases due to train load or adverse gradient.

A clutch or release mechanism is not provided on an electric locomotives as the load of mine cars should generally be kept attached to the locomotive, a typical condition of traction duty. If the load is not kept attached, *i.e.* if the torque decreases the rotor tends to race up and may develope dangerous centrifugal force.

The back e. m. f. of a motor depends on speed and field current. It is nil at the start and varies with the speed. At start the supply current can, therefore be kept within reasonable limits by inserting resistors in the circuit of the armature and the resistance is reduced in steps as the motor speeds up. At the normal working speed of the motor, full voltage is applied across the field and armature winding.

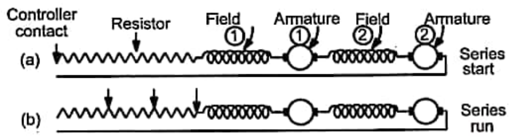


Fig. 16.10. Different electrical connections in a series wound D. C. motor of an electric trolley or battery loco having two motors.

In a series motor, however, reversal cannot be obtained by merely changing the direction of current in the armature since the relationship between the field and the armature remains the same and the direction of rotation is not affected. For purposes of reversal the armature is generally provided with two sets of windings. Changing the direction of current unchanged, causes reversal of the rotor. Fig. 16.10 shows the arrangements of speed control on twin motors. Use of resistors of control the full battery current is wasteful of electrical energy and wastage is more pronounced with frequent starting and stopping. The wastage appears as heat and it is, therefore, important to remember that with battery locomotives the motors should not be run with the resistors in the circuit except during speed changes which should not be sudden. The slowest economic speed is obtained with the battery paths in parallel and the motors in series.

Underground compressed air locomotives are not used in any of the coal mines in this country.

Diesel locos are not used at the face in this c ountry and their use on branch roadways is also rare. In inclined seams locomotives are used in the shaft levels or near the pit bottoms.

A loco can negotiate a right angle bend in 4.3 m wide gallery if track gauge is 0.60 m, the common gauge in most of our mines. If track gauge is wider, rhombus pillars with 120° angle are to be formed.

Example :

What is the maximum tractive effort that can be developed by a 15-tonne diesel locomotive of 75 kW assuming a coefficient of adhension of 0.25 ? At what speed will it haul a train when developing its fullpower and maximum tractive effort, assuming the mechanical efficiency to be 84 % ?

Ans. :

(a) Maximum tractive effort = Tm =0.25 × 15 × 9810 N

= 36787 N

(b) $\text{Power (kW)} = \dfrac{\textit{Tractive effort } (N) \times \textit{speed } (m/s)}{\textit{gear efficiency } (e)}$

$\therefore$ speed of train $= \dfrac{75 \times 10^3 \text{ Nm} \times 0.84}{36787 \text{ N}}$

= 1.71 m/s = 6.165 km/h.

If the locomotive was operating under conditions requiring a tractive effort of only $\dfrac{36787}{4}$ N, it would haul the train at 4 times the speed, namely 24.66 km/h when developing its maximum power. Alternatively if the speed were reduced at the lower tractive effort, the diesel engine would be required to develop a proportionately lower power.

Example :

A train of 200 tonnes is drawn up a slope of 1 in 200 against resistance of 60 N per tonne with uniform acceleration which in a distance of 2 km increases its speed from 36 to 54 km/h. Find the power at which the engine is working.

Ans. :

The average speed being 45 km/h, time for the journey

$= \dfrac{2}{45} \, hr = 160 \text{ sec}.$

The work done is in three parts :

(a) raising 200 tonnes weight through $\dfrac{2}{200}$ km.

$= 200 \times 10^3 \times 10 \times g = 19.62 \times 10^6$ J.

(b) overcoming resistance, 60 × 200× 2,000 J

$= 24 \times 10^6$ J.

(c) increasing kinetic energy ; v_1 =10 m/s. v_2 = 15 m/s

$$=\frac{1}{2}\times 200\times 10^3\ 15^2\text{-}10^2)$$

$$= 12.5 \times 10^6 \text{ J}$$

Total work done in 160 sec. = $10^6(19.62 + 24 + 12.5)$

$$= 56.12 \times 10^6 \text{ J}$$

$$\text{Average power} = \frac{\text{Work done}}{\text{time}} = \frac{56.12\times 10^3 \text{ kW}}{160}$$

$$= 351 \text{ kW.}$$

Example :

Find the maximum size of train (hauled load) which can be started up a gradient of 1 in 100 at an acceleration of 0.045 m/s² by a locomotive of weight 10 te if the coefficient of adhesion is 0.2 and the friction resistance coefficient is 0.01.

Ans. :

W_1 = wt. of loco = 10 tef = 98100 N

$\mu = 0.2$

$i = \frac{1}{100}$ (gradient of track)

f = 0.01 (frictional resistance coefficient)

Wt = ? (weight of train in Newtons).

Maximum tractive effort, Tm = 0.2 × 98100

This taractive effort has to provide 3 forces for the following purposes:

1. a force to accelerate the loco and the train. This is provided by the equation p=Mf.
2. a force to overcome the force due to gravity component of the loco and train acting parallel to the track.
3. a force to overcome the frictional resistance acting parallel and opposite to the direction of motion.

Forces resisting motion and parallel to the gradient

= Gravity component of loco and train + friction

$$= (Wt + 98100) \times 0.01 + (Wt + 98100) \times \frac{1}{100}$$

$$\text{Force required for accln. } P = Mf = \frac{(Wt + 98100)\times 0.045}{9.81}$$

or $0.2 \times 98100 = (Wt + 98100)\ (0.01 + 0.01 + 0.0046)$

$= (Wt + 98100)\ (0.0246)$

$$Wt + 98100 = \frac{0.2\times 98100 \text{ N}}{0.0246} = 797560 \text{ N}$$

or Wt = 699460 N = 71.3 tef.

Example :

A locomotive has a mass of 50 tonnes and the connected train has a mass of 250 tonnes. The train is to hauled up a slop of 1 in 120 at a 40 km/hr with an acceleration of 0.20 m/s². Resistance on account of friction and other causes is 70 N per tonne mass. Calculate the power that the locomotive has to develop.

Ans. :

(a) Component of the force along inclined plane due to wt. of loco and train.

$$= M \times g \times \sin\theta = 300 \times 1000 \times 9.8 \times \frac{1}{120} \quad \ldots (1)$$

$$= 24500 \text{ N}$$

Difference between sine and tan of the low angle of inclination is negligible.

(b) Frictional resistance = 70 × 300 = 21000 N ... (2)

(c) The accelerating force, P = Mf

$$= 300 \times 1000 \times 0.2 = 60000 \text{ N} \quad \ldots (3)$$

The tractive force exerted by the engine is

$$= 24500 + 21000 + 60000 = 105500 \text{ N}$$

Velocity of the train, $v = \frac{40 \times 1000}{3600} = 11.11 \text{ m/s}$

Power developed at this speed is = Total force × velocity

= 105500 × 11.11

= 1172105 W

= 1172.11 kW

Example :

A locomotive weighing 10 tef hauls a train of 50 tef down a gradient of 1 in 100 at a speed of 16 km/h. Brakes are applied on the locomotive to bring the train to rest. Calculate (a) the gross braking effort (b) the effective retarding force (c) the rate of retardation (d) the time taken to stop the train, and (e) the stopping distance. Assume a coefficient of dynamic friction of 0.16 and running resistance of 70 N per tonnefe.

Ans. :

(a) Gross braking effort = $Tb = \mu W_L = 0.16 \times 10 \times 9810$

$= 15696$ N

This includeS the running resistance of locomotive.

(b) Resistance to motion of loco and train

$$= R = F - G = (50 \times 70) - \left(\frac{60 \times 9810}{100}\right)$$

$= 3500 - 5886 = -2386$ N.

In this case the frictional resistance is less than the component of weight along the gradient *i.e.* the assistance to motion due to the gradient exceeds the frictional resistance and the net effect acts against the brakes. Thus,

Effective retarding force = Tr = 15696 - 2386

= 13310 N

(c) rate of retardation $= f = \frac{Tr \times g}{W} = \frac{13310 \times 9.81}{60 \times 9810}$

$= 0.222 \text{ m/s}^2$

W is the weight of the loco and train in Newtons.

(d) Full speed of train = 16 km/h = $\frac{16 \times 5}{18} = 4.44 \text{ m/s}$

$$\text{Time to stop train} = \frac{\text{initial velocity}}{\text{rate of retardation}}$$

$$= \frac{4.44}{0.222} = 20.02 \text{ sec.}$$

(e) Stopping distance

S = average speed × time

≈ 2.22 × 20.02 = 44

QUESTIONS

1. What are the limitations on the use of different types of locomotives in underground mines ?
2. Explain the following terms :
   i. Drawbar pull
   ii. Tractive effort
   iii. Braking distance
3. Describe the deviceS used on a diEsel locomotives for making it safe to work in gassy coal mines.
4. Sketch the layout of the battery charging room used for battery operated electric locomotives in underground mines.

O O O

CHAPTER – 17

CONVEYORS & OTHER TRANSPORT MEDIA

In coal mines and other mines of stratified deposits where the underground mineral is won by longwall method, the transport media which often consist of conveyors are termed as shown in Fig. 17.1, based on the location of the transport arrangement. The main haulage to pit-bottom may be belt conveyor, rope haulage or locomotives.

The principal types of conveyors used in the mines are :

i. Belt conveyor.
ii. Scraper chain conveyor.
iii. Shaker conveyor.

Other conveyors are variations of these and go by different names.

Belt conveyor :

The belt conveyor is basically an endless belt in a straight line stretched between two drums, one driving the system and the other acting as a return drum.

The system of transport by belt conveyors consists of the following :

1. A flat endless belt which continuously travels and carries on its top surface the material to be conveyed.
2. The idlers which support the belt.
3. The structure of angle iron or channel iron on which the idlers are mounted.
4. The tension arrangement for keeping the belt in proper tension, including the loop take-up arrangement.
5. The drums at the discharge end and opposite end (tail end) over which the belt passes.
6. The drive head comprising the electric motor, coupling, gearing and snub pulleys.

The belt :

The belt is an endless thick flat strip of woven cotton, rayon or nylon fabric laid up in plies, or layers and their surfaces and sides covered with rubber, plastic or P. V. C. The type of fabric, the number of plies and the reinforcement, if any, in the belt determines the strength of the latter.

Belts having nylon fabric are strong as nylon offers very high resistance to longitudinal tearing and damage due to edge turn up. Nylon also improves the resistance to impact damage and the belts are more flexible, light in weight, have better fastener-holding properties and are also designed for use on deep troughing idlers. Such belts are however costlier than the cotton-fabric belts. Cotton carcass is more susceptible to moisture and it rots on account of fungus formation in the presence of moisture.

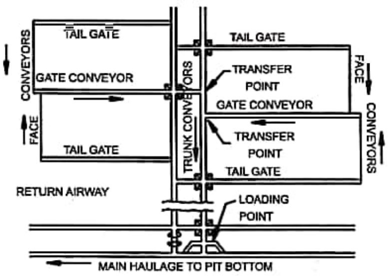


Fig. 17.1 Layout of face, gate and trunk conveyors in a coal mine.

Belts covered by a layer of P. V. C. instead of ruber are fireproof and only such P. V. C. coated belts have to be used in underground coal mines in accordance with the directions of the D. G. M. S. The P. V. C. belt is resistant to grease and oil. One disadvantage of P. V. C. coating is that it has a low co-efficient of friction rendering the belt unsuitable for steep gradients. *Normally* a belt conveyor with rubber coating is capable of working on a limiting gradient of 1 in 5 without any braking arrangement ; with thrustor brake the limiting gradient is 1 in3. The maximum angle of inclination for conveying coal on plied P. V. C. belting is 16°.

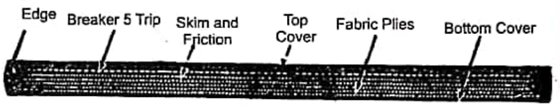


Fig. 17.2 Cross-section of belt for conveyor system.

Standard belt. Multiple layers of suitable duck of the same thickness and play across the entire belt.

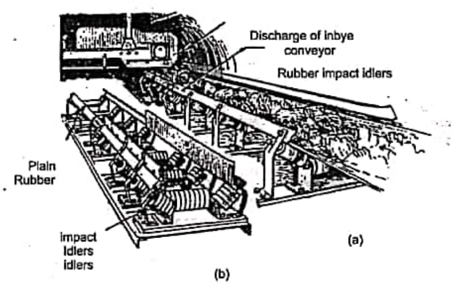


Fig. 17.3. (a) A belt conveyor in an underground mine.
(b) Left-Illustrating impact idlers and plain idlers.

The belt conveyor works on a *straight* roadway which may be level, inclined or partly level and partly inclined in patches. The conveyor must be erected in correct alignment. The belt speed varies from 45 m/min to 150 m/min but a speed of 45 m/min to 60 m/min is generally preferred.

The belt ends are joined by higned plate joints or wire hook joints, the latter being popular on account of their strength and are made more quickly than the hinged plate type. The belt ends can also be vulcanized but vulcanisation requires special electrically heated press and it also calls for special technique. PVC covered belts are joined by metal fasteners.

Belt widths vary from 0.60 m for face conveyors to 1 m for trunk, conveyors in underground coal mines but surface installations may have belts as wide as 1.2 m or even more as at Neyveli lignits mines. The maximum size is 1.5 m.

The carrying capacities of the belts are as follows for a troughed belt with a speed of 30 m/min with uniform feed of coal.

Belt width			Te/hr
650 mm ;	common as face or gate conveyor		50
750	do	do	70
1000 mm ;	common as trunk conveyor		105

If the belt speed is 60 m/min the capacity is obviously double. The capacity can thus be calculated. If the feed is not uniform, as very often happens, the capacity will be usually 30 to 60% of the calculated value. Compared to a troughed belt, a flat belt has nearly 50 % carrying capacity. The maximum lump size carried by a belt conveyor is limited to about half the belt width but large sizes like this cause spilling of the material.

The carrying capacity of troughed belt conveyor is given by $T = abv$

where T = the carrying capacity (in tonnes per sec)
a = the average cross-sectional area of material (in m^2)
b = the bulk density (te/m^3); this relates to density of broken material including air spaces.
v = speed of conveyor belt (in m/s)

For a belt of width w the value of the area a varies approximately between $\frac{w^2}{10}$ and $\frac{w^2}{12}$ depending on the nature of the material

The belts manufactured in this country are from 3-ply construction to 8-ply construction but PVC belts are available only of 3 and 4 ply construction.

Care of the belt :

1. Protect conveyor belt from direct sunlight during storage and keep it away from steam pipes or other places of heat.
2. Use drive drums and delivery as well as tail-end drums of adequate size so that sharp bending of the belt is avoided.
3. During handling do not subject the belt to many bendings or warpings.
4. Prevent the belt from rubbing against any prop, timber other stationary object and avoid wandering of the belt.
5. For a troughted belt an inclination of more than30° for the side rollers is not recommended.

6. When in use the cotton-fabric belt looses its strength gradually due to the action of moisture. Even small punctures can permit moisture to reach the feabric and this is spread through the cracass by wick action. Where the belt is joined with fasteners,moisture enters at the joint and after a time the belt fails at the fasteners so that it has to be shortened and the joint re-made.
7. The method of feeding on to a belt conveyor has a marked influence on the belt life and it should be made as smooth as possible, always depositing the load on to a stretch between supporting rollers so that the elasticity of the belt can cushion the shock and as far as can be arranged, in the direction of the belt travel and at the same speed. Use only impact absorbing rollers at the transfer point.

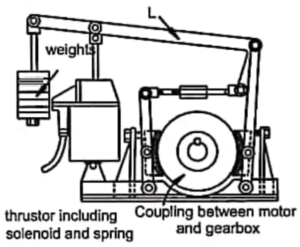


Fig. 17.4 A thrustor brake for a belt conveyor

When power is switched on the thrustor pushes up lever L thus releasing the brake. When power is cut off, the thrustor is released and the weight pulls down lever L to apply the brake.

Idlers and the supporting structure :

The belt travels on idlers placed at intervals of 1.5 m to 2 m. The idler is a long pulley moving on its own axle and ball bearings and filled with grease. For transport of broken and loose material, the flat belt should be given a troughed shape for that length which has to accommodate the mineral and such belt troughing is achieved by a set of three idlers at one place, the middle idler being horizontal and the two on either side, at 20° to 30° to the horizontal. All cotton carcass belts should be used for throughing not exceeding 30° to the horizontal. For some narrow belts two angled idlers may be used

and for very wide belts, a set of five separate idlers is sometimes employed so that the curve of the troughs is more regular. The idlers are often greased filled for life, *i.e.* they need not be lubricated of for installationand have to be discarded after their specified life is over. Some idlers are, however, provided with grease nipples which have to be greased once in 24 hours.

Idlers at the transfer points have to bear the impact of falling mineral and they are of special construction with thick corrugated rubber layer on their surface to take up the impact and provide cushoning.

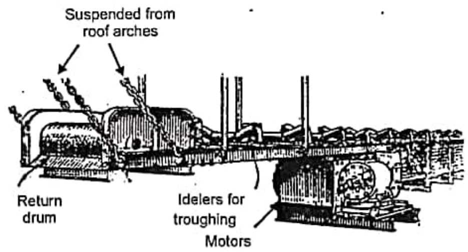


Fig. 17.5 Framework of belt conveyor supported from roof.

The idlers are supported on channel iron framework and the members of such framework are 3-4 m long, joined by bolts and nuts. The frame-work in an underground mine rests on the floor of the roadway but some stretch of it, particularly at the discharge end, may be suspended from the roof by eye-bolts and chains. Such suspension is necessary where a belt conveyor has to cross a haulage track with tubs or another conveyor, and at delivery ends in thin mineral beds. (Fig.17.5).

The direction of axles of the idlers, the delivery-end drum and the tail-end drum should be at right angles to the direction of belt travel, otherwise the belt wanders or shifts off the centre line for part of its length and its throughing is upset resulting in spillage of the contents. The belt edge may rub against stationary parts and get damaged. Prolonged rubbing can result in dangerous heating that may prove disastrous in underground coal mines with inflammable gas. Proper alignment of the pulleys, idlers and the belt is therefore very essential. The idlers can be swivelled at a small angle oblique to the normal setting. (Normal setting is at right angles to direction of the belt travel). Slight swivelling of the idlers sometime help keeping the belt central. When making adjustments to the idlers for keeping the belt run in its true centre line, it is better to make slight adjustments in several sets of idlers than to make a large adjustment in one. (Fig. 17.5). The general rule is the conveyor, the cause lies in the conveyor structure. If, however, one or more portions of the belt persistently run off-centre at every point in the conveyor, the belt or the joint is the cause. In case the belt has a tendancy to climb to one side, vertical idlers at intervals can check the climb. If the belt runs true when empty but off-centre with load, the obvious fault lies with the method of loading. The load should, as far as practicable, be fed on to the belt centrally, in the same direction as belt speed. Chutes with suitable angle be used at the transfer point with this object.

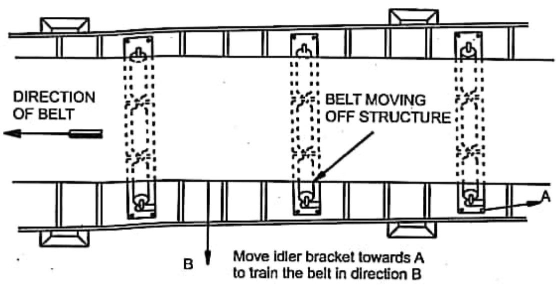


Fig. 17.6 Adjustment of idlers to course a wandering belt.

A few support structures near the end of a conveyor installation usually at the tail end, are of telescopic construction. They consist of two troughs, one sliding inside the other, and they are placed near the tension device so that only the drum and the conveyor structure adjacent to it need be moved when extending the conveyor, or when tightening the belt. Sometimes a telescopic trough is placed next to the drive-head.

Tension arrangements :

The belt should be in proper tension If it is loose or slack on the driving drum, the latter will not be able to transmit the power to the belt. The tension end of a belt conveyor of a small length is the return drum mounted in a steel framework to which hooks are fitted for tensioning by means of sylrester proper with drawer or by tensioning screws. Such arangement is provided on face conveyors of longwall faces in mines. On gate belt conveyors which have to extend as the roadway advances the extra belt required for extension is provided by the "loop take-up" arrangement, placed near the drive

head. (Fig. 17.6). Belt conveyors are extended generally after 90m advance of the roadway and the loop take-up arrangements accommodate extension upto 9m. On a belt conveyor system of more or less permanent nature the tension arrangement is provided near the drive head and it consists of weights placed on a movable pulley or drum over which the belt passes. Very tight loop tension will cause excessive stress in the loaded belt which is liable to break, or damage joints and shorten the belt life. It is therefore important that no belt conveyor should be run with the loop at a higher tension than is necessary to obtain starting and stopping without slip under the various conditions of load.

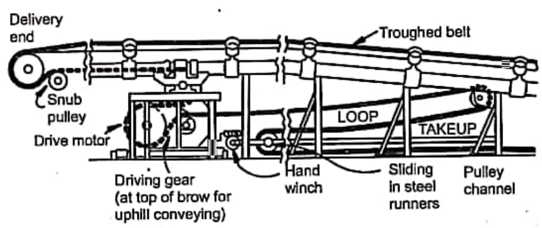


Fig. 17.7. Arrangement of driving gear and loop take-up for a belt conveyor on level or up-hill gradient.

The function of the tail end of a conveyor belt is :

1. To house a suitable drum for the return of the belt.
2. To guard the belt and the drum from being foulded by small coal, or other mineral being conveyed.
3. To cover the whole internal structure and thereby to avoid accidental contact with moving parts.
4. To provide means for anchoring and tensioning the belt.
5. To act as terminal member of the framework.

Drive head :

This consists of an electric motor or compressed air motor, fluid coupling, gear wheels and a drum which provides the necessary arc of contact to the belt by friction and snub pulleys are employed to increase the are of contact. The coefficient of friction between the driving drum and the belt is sometimes increased by lagging the drum with a suitable rubber- like material although this is not done if it can be avoided.

The position of drive head depends on :

i. On the level, uphill and undulating gradients the drive head is generally near the delivery end and the loop is tensioned immediately after the belt leaves the drive. (Fig.17.6)

ii. For down hill conveying the drive head is not very close to the delivery end.

Usually one motor drives a belt for a roadway of upto 200 m on level gradient. For longer length of the roadway there are two motors, one at each end of the roadway, and these are driven synchronously. On large capacity conveyors the two drive motors are sometimes placed close to each other. The motor is a squirrel cage type with high torque and is switched on direct on the line. Drive heads of intermediate conveyors and inbye conveyors, when the conveyors are in series, are sometimes provided with sequence control operation, described later.

Holdbacks :

An inclined conveyor tends to run backward when power is cut off. Because the weight of the load pulls the upper part of the belt down hill. The tendency can be overcome by means of a brake of sufficient size,but it is often more convenient to use a device which will automatically lock it against turning backward, without interfering with normal movement of the belt. Such device, called holdback, consists essentially of a ratchet wheel keyed to the drive pulley shaft and held between a pair of side plates which are hinged to a toggic and tooth pawl anchored on the conveyor frame. When the belt and drive pulleys are turning in the normal direction, friction between the ratchet and the side plates lifts the unit on the hinge, holding the pawl clear of the ratchet. If the drive drum starts to turn backwards, the hinge is pulled down and it meshes with the ratchet.

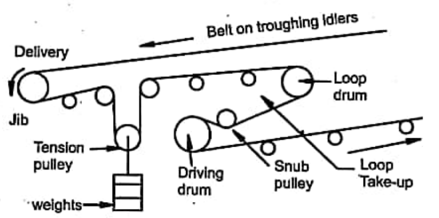


Fig. 17.8. Arrangement of drive motor, loop take-up and tensioning weights on a belt conveyor discharging down hill.

Sequence control of conveyors :

If an outbye conveyor stops for some reason but not the feeding inbye conveyor, there will be a pile-up at the transfer point i.e. discharge end of the inbye conveyor which will overload the outbye conveyor. This situation itself is not desirable, one of the results being that the belt will be damaged. The overloaded belt will slip on the drive drum and this will generate heat leading ultimately to the belt catching fire. This is disastorous. Such situation can be avoided by *sequence control* in the starting and stopping of a series of conveyors.

In a sequence control,

1. The inbye conveyor cannot be started unless the outby receiving conveyor is set in motion and has attained nearly 60% – 70% of its normal speed.
2. If the outbye conveyor stops for any reason, stalls or slips,the inbye conveyor automatically stops.

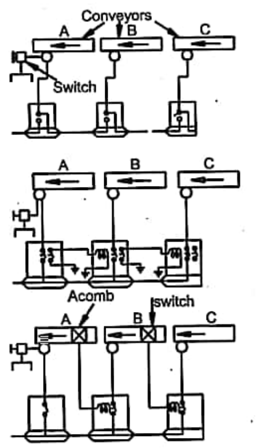


Fig. 17.9

These principles of sequence control are shown diagramatically in Fig. 17.9. A. B. C. are three conveyors in series. Normally if A stops the operator at delivery end of B should stop conveyor B. same applies to conveyor C also.

This brings in the human element and delay in stopping B can result in pile-up of mineral at its discharge end unless it is provided with a hopper of adequate capacity. Timely stoppage of B and also of C can be ensured by a sequence control of their "Stop" switches or contactors. There are three arrangements of sequence control.

1. *Automatic sequence control.*
2. *Power sequence control.*
3. *Pilot sequence control.*

The automatic sequence control and pilot sequence control are normally adopted.

Automatic sequence control : This is operated by and interlocked with the movement of outbye belt. Switch of belt B is closed by a governr functioning by the movement of the belt A (Fig. 17.8).

If A is not moving B cannot be started. As shown in the Fig. 17.9 the switch box is pivoted and its weight keeps the driving pulley presses against the under side of the belt. The shaft of the governr system of switch box can be driven, as an alternative arrangement, by the idler roller. To start with, the belt A is started by manually operating its motor switch. After it attains 75% speed, the governr of the switch box closes the contactor switch of motor of belt B which then starts. In a similar manner and sequence, belt C is started.

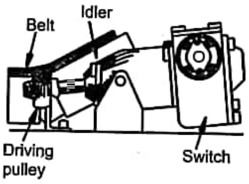


Fig. 17.10. Governor pulley driven by belt for sequence control.

The same governer-controlled switch of belt A, should the latter break, stop or stall. Thus protection of belt A is ensured. It will, therefore be seen that the governer-controlled microswitch shown in fig. 17.9 is connected to control switch of inbye belt for sequence control and also to the control and also to the control switch of its own driving belt, electrically, for belt protection.

Pilot sequence control : There is a pilot cable throughout the system for all the conveyors and the pilot circuit only is controlled. No contactor passes more current than is necessary for the conveyor to which it is connected. The pilot cable operates on the low-voltage auxiliary supply from one control switch to another. The supply to inbye conveyor is controlled by means of auxiliary contacts on each contactor. The disadvantage of the system is the necessity to use pilot cables which may be of considerable length, imposing extra expanditure. (Fig. 17.10)

All the sequence control arrangements have suitable provision to nullify or defeat the sequence control operation for routine checking, maintenance and repairs so that any single conveyor can be stopped at any time. One of the results of sequence control of conveyors is that only one operator at the outbye conveyor is required for starting or stopping the conveyors. Attendants at each conveyor is are not necessary and this saves manpower.

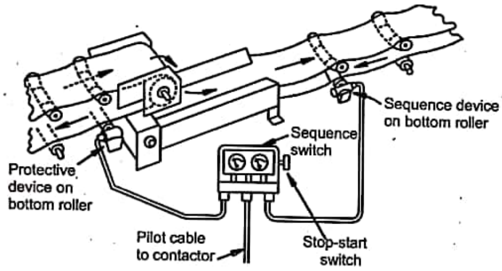


Fig. 17.11. Pilot sequence control of conveyors.

Remote control of conveyors :

On conveyor systems, the remote control switch at the delivery point can be connected in parallel to a cable running the length of the conveyor with press buttons inserted in the cable for local control. Thus a converyor can be stopped from any point along the road, or a pull wire is stretched all along the length of a conveyor system. It is connected to the switches of the conveyors and a pull at the wire from any point along its length stops the conveyor.

There are, in addition, start/stop buttons at drive end and delivery ends of conveyors and these buttons are controlled by pilot cable or pilot circuit. Such start/stop buttons and the "stop" arrangement operated by the pull wire are interlocked so that the conveyor cannot be started if the "stop" button is pressed at any point, and the "stop" button needs to be re-set before starting the conveyor.

Belt sway monitoring switch :

A belt of a conveyor system may wander or sway off its alignment for various reasons. When the sway exceeds a permissible limit, the belt sway switch gives an audio-visual alarm and can also be arranged to switch off power supply to the driving motor. The sway monitoring switch consists of a contact roller, a cam, a camshaft and a limit switch. Except the contact roller all the components are housed in a C I. housing which is bolted to the conveyor frame work in such a way that the edge of the belt is always in contact with the contact roller. The deflection of the roller causes the main shaft carrying the cam to move through an arc and the cam then actuates the limit switch whose contacts are wired in the motor control circuit for switching off power in case of sway of the belt beyond a permissible limit. The roller revolves on pre-lubricated ball bearings during the motion of the belt and a heavy duty spring inside the housing cushions the impact from the belt and resets the roller as soon as the former returns to normal position.

Scraper chain conveyor :

A scraper chain consists essentially of stationary steel troughs, each usually 2 m long, connected together end to end, and an endless chain with flights moving in the troughs, which are nearly 450 mm wide at top and 300 mm at bottom. The troughs are supported on angle iron frame work. Each trough is slightly dished at one end so that the next one fits in to form a flush joint. Adjacent troughs are secured together and to the underframe by bolts. This gives rigidity to the assembled coveyor and facilities dismantling and re-assembly. The chain consists of links and after every 3-4 links a flight is provided so that the flights are 2-2.5 m apart. Links are provided with shear pins at intervals and in some models, the driving sprocket is not keyed to its shaft but transmits the drive through a renewable shear pin and this provides a safety measure against overload.

The return or tail end of the conveyor, with its totally enclosed sprocket-drum, is provided with a telescopic trough by which the tension of the chain can be adjusted through sylvester chain. The sylvester draws back the sliding portion to the required position where it can be fixed by means of bolts.

The capacity of a commonly used scraper chain conveyor is 30 to 40 tph on a level roadway, nearly, 50 m long and the drive motor is 12-15 kW. It is a single chain conveyor, the chain moving with a speed of 35 m/min.

The advantages of scraper chain conveyor are :

1. Unlike the shaker conveyor, , it can convey mineral uphill against relatively steep gradients (upto about **1** in 3, or even steeper) as well as on level or downhill gradient.
2. It is much stronger and can withstand more rough handling than the belt conveyor.
3. It can be readily dismantled, moved forward, extended or shortended.

The **disadvantages** : It is fairly high in first cost and in power consumption; it has too many moving parts ; it is somewhat noisy in operation and tends to increase the percentage of small coal. The convenient length to handle is 100m.

Armoured chain conveyors (also called snaking or python conveyors) :

These are principally for use on a prop-free front of a longwall coal face. They can be advanced without dismantling, with the help of hydraulic, or pneumatic rams, or even hand operated jacks. They can work with lateral or vertical undulations and coal cutting machines and shearers may be mounted on them. These are used in India in only a few coal mines. The total power of motors used varies from 30 to 185 kW ; the pan width at top varies from 750 to 850 mm and pan length from 1.3 to 1.8 m ; the vertical flexibility of pans is 3-4° and the horizontal flexibility is 2-3°; limiting gradient without flights is 1 in 3 and with flights, 1 in 1.5; usual length with one drive is 90 m; with multidrives the length may be increased upto 360 m. Capacity is upto 100 te/hour. These are fitted with single chain or double chain.

Cable belt conveyors : In this type, the driving tension is taken by two separate steel wire ropes, one on each side of the belting and not by the belting inself. Belt only carries the material and is provided on either side with shoes, which rest on carrier ropes. Straight course is essential for operation. A 1 m belt at 75 m/min. has a rated output of 300 te per hour. Rope diameter is generally 25 mm. Sizes of belts are available between 0.75 and 1m. Such conveyors are suitable for long lengths, particulary on inclinations. Cable belt conveyors are at present not used in Indian mines. (Fig.17.12)

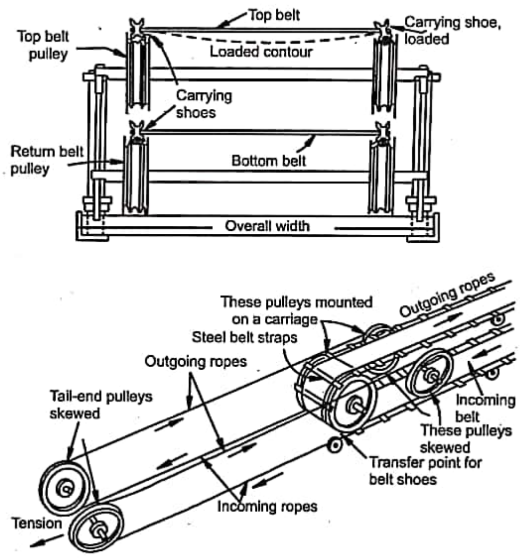


Fig. 17.12. Cable belt conveyor

Steel band conveyor : This is,in essence, the usual type of belt conveyor but the rubber belt is reinforced with steel bands and the belt does not stretch appreciately in operation.

Negotiating curves on belt conveyor installations :

In our country belt conveyor installations follow a straight-line path and small undulations in a vertical plane are covered up by the single belt. Curves in a horizontal plane are not covered up. If the belt has to deviate off the straight-line path by even a small angle the practice is to have transfer points, with belts in series in two straight-lline paths. Each beltrequires its driving motor, return drum, etc. In foreign countries some belt conveyor installations have single belts which cover up the angular deviations in the horizontal plane along the path. The construction Company, R.E.I. of France, had constructed in 1970 a belt conveyor installation at Nepoui (New Caledonia in the Pacific) for transport of nickel ore from the mine to the harbour installation of Society Le Nickel. Its capacity was 1,000 tph and the singal flight curved belt conveyor was 13 km long with overall descent of 27m. Such curved installations of belt conveyor are known by the trade name Stereoduc. The actual conveyor belt measured 26400 m & 800 mm width. The single flight of 13 km distance had only two driving heads using six motors; 4 units each of 370 H. P. at the tail end and two units each of 370 H. P. at the head end. Belt speed was 4 m/s.

Another curved path belt conveyor installation by the same firm is by the trade name Curvoduc at MEA in the Pacific for transport of Nickel ore, across a hilly terrain. The conveyor distance is 11, 120 m in a single flight and the belt measures over 22 km in one piece.

Fig. 17.14 Map showing the route Curvoduc. certain stretches vertical and horizontal curves are combined Main features of the installation :

Main features of the installation :

Distance between drum centres	–	11,103 m
Total vertical descent of the product along the route	–	567 m
Belt width and speed	–	800mm; 3.6 m/s
Drive unit (single drive unit)	–	1100 H. P. synchronous variable speed motor
Structure for supports	–	Tubular structure welded in 100 m lengths
Throughput	–	600 tph

REI has commissioned for National Coal Board, U. K., a Steroduc 15km long to convey upto 3200 tph (10 million tonnes/ year) on a 7100 N/ mm steel cord belt weighing 2500 tonnes running at 8.4 m/s. Its through lift is 996 m. its drive unit includes only 3 components, 2 motors (14,000 H. P.) and a singal drum-shaft weighing 90 tonnes.

Disc conveyors : These are designed for steep gradients, especially for retarding conditions. Curved troughing is used. (Fig. 17.14)

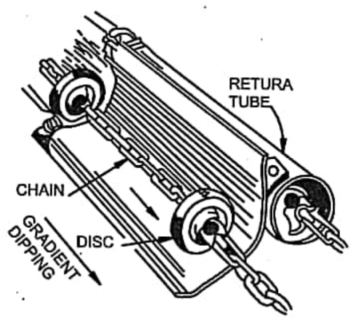


Fig. 17.15. Disc conveyor

Gate end stage loader (also called gate end Loader) : These are generally short, low powered chain conveyors, which whilst being low in construction at the face end, and easily extensible, can elevate the coal and feed it uniformly on to the gate belt without causing spillage and damage. It is essentially a chain conveyor. Gate end loader can be profitably used for tub loading but is not suitable for large outputs. The unit is wheel mounted, nearly 9 m long having 4.5 to 8 kW motor and a loading capacity of 1 to 1.5 te/min.

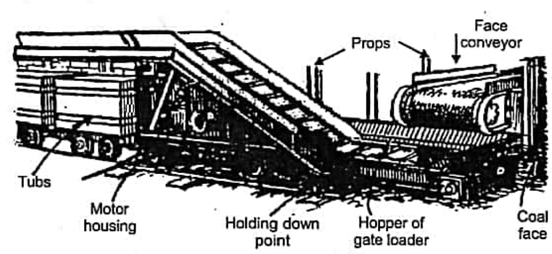


Fig. 17.16 Gate end stage loader.

Shuttle car :

A shuttle car (Fig. 17.16) is a pneumatic tyre mounted electrically driven, low-height transport vehicle of 5-7 te capacity with an open-topped and open-ended body, used for transport of mineral from face to a central loading point. The floor of the shuttle car is fitted with a scraper chain conveyor which is run at intervals during loading of the shuttle car for its even loading. When discharging the contents at the central loading point the built-in scraper chain conveyor is operated to empty the car within 45-60 seconds. The trailing cable used is 90 m, the maximum length permitted by DGMS, and with such length the operating range of the shuttle car is 180 m though this range can be increased by placing gate end boxes at intervals,of 180 m. Time is however lost in connecting and disconnecting the cable at the gate end boxes. The average load of the shuttle car comes to nearly 75 % of the struck capacity. Rhombus pillars of 120° and wide galleries (4.2 m to 4.8 m) are to be provided in coal mines for negotiating curves. One shuttle car (7-te struck capacity), operating within 90 m range and fed by a 4-te/min mechanical loader is capable of transporting nearly 150 te of coal per shift of 8 hours and 2 shuttle cars used in the same district may deal with 250 te/shift. Travelling speed of the shuttle car is 5-6 km/hr with load and 7-8 km per hr.

Battery operated shuttle cars are also available. Disel operated shuttle cars are not used in our country. Electric shuttle cars provided with cable tapping D . C. power from underground overhead trolley wires (where trolley wire locos are used) are available for use in deg. 1 gassy mines.

A shuttle car works on nearly level gradient and for proper working the floor should be strong. In any case gradient should not exceed 6° though short hauls at a gradient upto 10° are feasible in special circumstances.

In coal mines if the floor is of coal or softer rock, the shuttle car operation results in mucky and slushy floor as water usually percolates from the side and roof strata. The minimum height of the roadway for shuttle car working is 1.2 m.

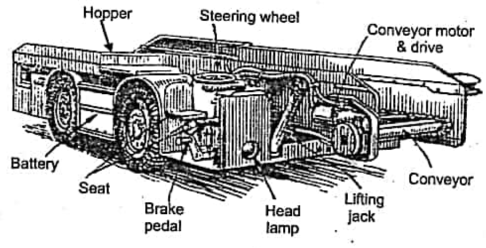


Fig. 17.17. Battery operated shuttle car.

QUESTIONS

1. Describe the construction and working of a belt conveyor. State the normal sizes of belt conveyors used in the mines and their capacity for a speed of 60 m/min. (troughed belt).
2. Describe an arrangement of sequence control of belt conveyors.
3. Write short notes on :
   - i. *Scraper chain conveyor*
   - ii. *Shaker conveyor*
   - iii. *Cable belt conveyor*
   - iv. *Gate end stage loader.*
4. A belt of 1 m width conveys material of bulk density 1.35 te/m³ at a speed of 90 m/min. What is its carrying capacity?

   (Ans. 666 Te/hr.)
5. In a belt conveyor installation the rubber belt frequently wanders off the straight path. Describe the various methods for rectifying the defect.

O O O

CHAPTER – 18

PRINCIPLES OF HYDRAULICS & MINE PUMPS

A fluid is a substance which may be a liquid or a gas. It is capable of flowing and offers negligible resistance to change of shape. Whereas a gas completely fills the space in which it is placed, a liquid has limited volume and if the container is big enough, it will have a free surface.

The subject of hydraulics refers to the branch of engineering science dealing with the physical properties and motion of liquids (specially water), and the transmission of power by means water pressure.

The working of puumps and such other devices as a hydraulic press, siphon, *etc.* depends on the following well familiar properties of liquids.

1. Liquid find their own level.
2. A liquid is almost incompressible. In this respect it differs from gas which is easily compressible.
3. A liquid exerts pressure in all directions in the container, and at any particular depth below the free surface, pressure at a given point in a liquid acts eqally in all directions. Alternatively we may say that pressures at all points on the same level in a liquid at rest are equal.
4. Pressure per unit area varies directly as the head *i.e.* the vertical depth below the free surface. If it be desired to calculate the total pressure on any immersed area, either vertical or inclined. *e.g.* on the sides of a tank, a dam, or a sluice gate, it is obvious that the calculation must be based on the average pressure per m². For this purpose the average head may be taken as the height measured from the centre of gravity of the area concerned to the surface of the liquid.

   The centre of pressure of a plain immersed surface, either vertical or inclined, is the point on the surface through which the resultant of all the pressures on the surface acts. This point, of course, is not at the centre of the surface, but below it, because the pressure is greatest over the lower portion of the surface due to increasing depth.

5. When a body is immeresed, either partially or wholly, in a liquid,the liquid presses vertically upwards with a force equal to weight of the liquid displaced. This is known as Archimedes principle. The upward pressure by the liquid on the body immeresed is called the buoyancy of the liquid.
6. **Bernouilli's theorem :**

It states : when a fluid flows through a passage of varying cross section, the total energy of the moving stream remains constant, assuming no friction losses.

The total energy is the sum of the kinetic and pressure energies, and so it follows that a reduction in the velocity energy is accompanied by a corresponding increase in the pressure energy.

If U and V are the velocities of water at two points in a pipe line, the difference in pressure energy of water is

$$P = \frac{V^2}{2g} - \frac{U^2}{2g}$$

This theoretical gain, however, is never obtained in practice, for it assumes perfect conversion of energy, and it ignores all frictional and shock losses within diverging passages.

A simple machine where the above principles find an application as far as ability of liquid to transmit pressure is concerned is the hydraulic lift. Its principle of working is explained in Fig. 18.1. We know that density is equal to $\frac{\text{Mass}}{\text{Volume}}$. If a liquid has a density of d kg/m³ the pressure caused by a liquid of column of h meters high = hdg N/m². This is, of course, only the pressure providedby the liquid column. If there were also some pressure on the top surface of the liquid the pressure at the bottom of the column would be greater.

Suppose a force F_1 is applied to a piston of area A_1 resting on top of the liquid in the narrow limb,

then pressure exerted on the top of the liquid F

The pressure at $X = \frac{F_1}{A_1} + h_{1\,dg}$

This is also the pressure at Y.

The pressure at Z = Pressure at Y - h_2 dg.

∴ if the force at Z is acting on an area A_2 the force at Z = pressure at Z × $A_2 = F_2$.

$$\frac{F_2}{A_2} = \text{pressure at } Z = \frac{F_1}{A_1} + h_1 dg - h_2 dg.$$

In most cases $h_1 dg - h_2 dg$ is insignificant in comparison with F_1/A_1 so that

$$\frac{F_1}{A_1} = \frac{F_2}{A_2}$$

The hydraulic lift is a machine which can be used for converting a small force F_1, into a larger force F_2.

Mechanical advantage

$$= \frac{\text{Load}}{\text{Effort}} = \frac{F_2}{F_1} = \frac{A_2}{A_1}$$

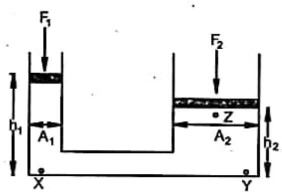


Fig. 18.1 Illustrating principle of hydraulic lift.

Calculation of V. R. :

Let the load rise a distance s.

$\therefore$ Volume of liquid pushed into right-hand limb = $s \times A_2$

This quantity must have been pushed out of the left-hand limb by the effort. If the distance moved by the effort is S, the volume of liquid pushed out = $S \times A_1$

$\therefore$ $S \times A_1 = s \times A_2$

$$\text{V. R.} = \frac{S}{s} = \frac{A_2}{A_1}$$

Hydraulics finds an extensive application in the working of pumps in mines. Reciprocating pumps mainly because reciprocating pumps are bulky, slow-speed, make more noise, have more moving parts and demand better standard of maintenance. Rotary pumps, like the centrifugal and turbine pumps, are direct-coupled to the electric motors, eleminating use of bulky gearing arrangement. In the following pages only the centrifugal and turbine pumps (and one pump of special construction, Roto pump) will be described.

TURBINE & CENTRIFUGAL PUMPS :

A centrifugal pump consists essentially of :

1. an impeller keyed to a shaft.
2. a stationary spiral or volute casing within which the impeller rotates rapidly (usually 1450 or 3000 rpm).
3. suction pipe connecting flange.
4. delivery pipe connecting flange.

The impeller looks somewhat like a wheel formed of two discs between which a number of curved blades or vanes are fixed. These blades are usually curved backwards, compared with direction of rotation. There is an opening at the centre, called the eye of the impeller, for entry of water sucked into the pump. In a single inlet pump there is only one eye on one side of the impeller and in the double inlet pump,there are two eyes or entries, one on either side of the impeller. The diameter of the impeller ranges between $1\frac{1}{2}$ and 3 times the diameter of the eye.

As the impeller revolves the water is carried round by the blades and thrown off from the impeller periphery at an increased radial velocity and pressure. The water enters the volute casing which is of spiral construction with gradually increasing cross section. In the volute casing the water velocity gradually decreases but the pressure energy of water correspondingly increases in accordance with Bernoulli's theorem and the principle of conversion of energy. When the water leaves the volute casing it possesses high pressure energy but only a little kinetic energy. In practice more than half the total pressure is created within the impeller itself and the balance in the volute casing. Such pump is suitable for heads (upto 20 m) and large quantities of water(even upto 4000,000 *l*/min). In small quarries, in the coal washeries and for irrigation purposes a centrifugal pump has proved quite popular.

The words *centrifugal pump and turbine pump* are used very often loosely to describe a pump in which water develops pressure mainely by rapid rotation of impellers within a stationary casing. The two terms are however different and a turbine pump differs from the centrifugal pump in that the former consists of a number of impellers mounted on one shaft and the water of each impeller enters stationary diffusing channels of the diffuser surrounding the impeller. The direction of water is shown by arrows in Fig. 18.3. It will be observed that water enters the impeller nearest to the suction pipe, is carried by the rotating impeller to the periphery at a high speed and somewhat increased pressure, and then discharge pressure energy and only a little kinetic energy. Leaving the diffuser, the water enters the next impeller at a high pressure and low velocity to undergo similar process whereby its velocity is again increased and the pressure further boosted. The process continues till water enters the delivery pipe with a high pressure but only a little velocity. Fig. 18.6 & 7 show the build up of pressure in the impeller and the diffuser. Each impeller with the diffuser surrounding it constitutes one stage and the head developed per stage varies from 15 m to 50 m depending upon (1) diameter and speed of the impeller (2) the curvature of the impeller, whether forward or back-ward, (3) the design of the diffuser. (Sec Fig 18.6 and 18.7 and pump calculations).

A turbine pump carries a balancing disc which is not provided on the centrifugal pump. It is described in more details later in this chapter.

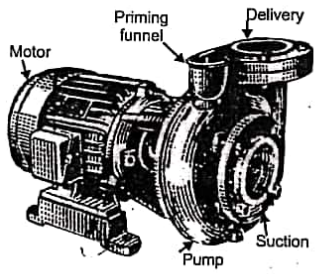


Fig. 18.2 Centrifugal pump; monoblock constr.iction.

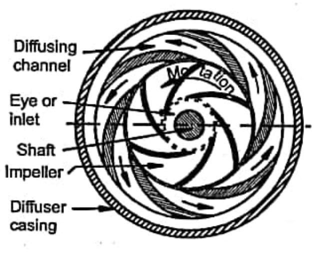
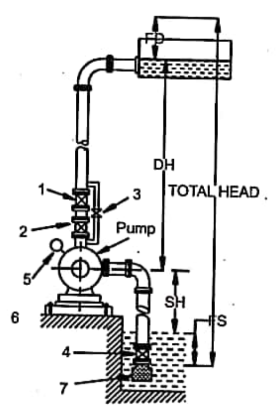


Fig. 18.3 Impeller and diffusing channels in a centrifugal pump (Cross-section)



1. *Retaining valve*
2. *Sluice valve*
3. *Bypass valve*
4. *Footvalve*
5. *Pressure gauge*
6. *H. D. bolt*
7. *Straine*

SH-Suction head
DH-Delivery head
FD-Head due to friction in delivery pipe
FS-Head due to friction in suction pipe

Fig. 18.4 Fittings on a centrifugal or turbine pump

The number of impellers on the pump shaft normally does not exceed 10 in order to prevent bending and to reduce the length. The diffusers, when placed side by side complete the outer casing of the pump and the diffusers are held together by 4 or 5 long bolts passing through the flanges of the two end-covers which form the suction and delivery chambers respectively.

In centrifugal pumps of monobloc construction (Fig. 18.2) impeller is mounted on the motor shaft. To avoid long shaft length, such pump is only single stage.

Pump fittings :

The valves required with a centrifugal or a turbine pump are :

1. a foot-valve in the suction pipe to prevent water returning to the sump.
2. A main valve (also called sluice valve or gate valve) in the delivery column.
3. A retaining valve to hold the water in the delivery column if the pump stops while the main valve is open.

4. Bye-pass valve to enable the pump to be primed with water from the delivery column before starting up. On small pumps this is generally not provided.

5. Air cocks (one on each stage to release the air when priming the pump.

All these valves are external to the pump and remain steady while the pump is working. Other fittings include a pressure gauge on the delivery branch, a vacuum gauge on the suction branch as an optional fitting, and a hydraulic balancing disc to counteract the thrust.

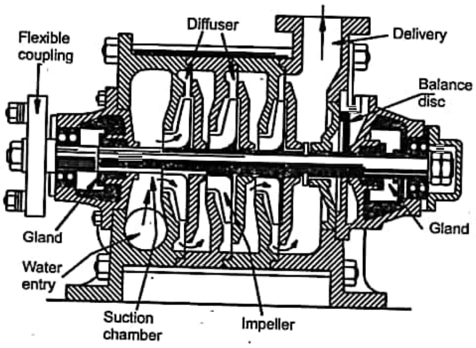


Fig. 18.5 Turbine pump in section.

Arrangement of pipes and valves :

The requirements in the suction pipe of a turbine pump are :

1. The total suction lift, including vertical lift, pipe friction, and the friction of the foot-valve and strainer, should not exceed 5 m upto the centre line of the pump.

2. The suction pipe should be as short as possible, of large diameter and have minimum number of bends or elbows.

3. The pipe line should rise all the way to the pump so as to avoid air pockets.

4. An efficient strainer should be fitted well below the lowest water level.

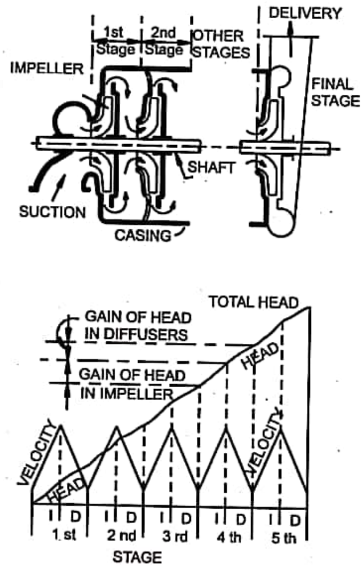


Fig. 18.6 & 18.7 Pressure build-up in a turbine pump

It should be borne in mind that when the impeller of a centrifugal or turbine pump rotates, it causes a suction effect in the pump and water enters the suction pipe as the atmospheric pressure forces the water in the suction chamber. The atmospheric pressure can however hold a water column, theoretically, which is 10 m high at sea level, if the water is at atmospheric temperature. The atmospheric pressure has to balance not only the vertical height of water column in the suction pipe, but also to overcome the friction in the suction pipe, at bends and elbows of the suction pipe and further, it has to impart velocity to the water which enters the pipe. No matter how efficient the pump is, it can suck water only upto a maximum of 9 m at sea level, as the atmospheric pressure can push it up only upto that vertical height. In practice, however, a vertical lift of 4-5 m should be considered to be a convenient maximum height for a suction pipe.

Starting a centrifugal or turbine pump :

A centrifugal or a turbine pump must never be started without priming with water as it does not create a vacuum of more than a few centimetres when working on air. The procedure to be adopted to start a centrifugal or turbine pump is as follows :

1. Before priming keep the air cock of each stage open.When the particular stage is full of water, the air cock will over-flow with water. Close the air cock when the water overflows from it.
2. Close the main valve on the delivery column.
3. Check up for any leakage of air or water on the suction pipe and upto the air cocks.
4. Put the motor switch "on". Let the motor and pump run for ½ - 1 minute with the valve closed. Open the air cock of one or two stages ; if the water forces out with pressure, the pump is working satisfactorily. Check this up from the pressure gauge on the delivery side of the pump. The gauge should record full pressure.
5. Now close the air cocks and open the main valve on the delivery column slowly if the latter is not full with water, but if it is full, open the main valve fairly rapidly. If this precaution is not observed the motor may get overloaded. it is a good practice to watch the ammeter while the main valve is being opened so that the load on the motor can be properly controlled.

To stop the pump, first close the main valve and then open the motor switch.

When starting the pump, if it refuses to deliver the water the reasons and remedies are as follows :

1. See if the direction of rotation is correct. It is always marked on the casing by an arrow.
2. See that the strainer and the footvalve are below water level in the pump and also check that the footvalve is not kept open by some obstruction of wooden piece or coal lump. The water in the suction pipe will flow away, if the footvalve kept open by such obstruction.
3. Check for air leakage on the suction side. The suction hose may have small punctured holes due to rough usage. Check at all pipe joints on the suction side, Covering the joints with moist clay, wherever practicable, helps plug the air leakage.

4. Air may leak at the gland of the stuffing box. If possible cover the stuffing box with an improvised water seal. Cotton waste, fully drenched with water, may be placed at the entry of shaft into the gland. Very often this helps.
5. Foreign substance may have obstructed water passage into the suction pipe. Tapping the steel suction range with hammer may dislodge the obstruction from its position.
6. Delivery range might have developed a large leak at a place not easily noticeable. The motor will be overloaded in such case and ammeter will indicate this.

Balancing axial thrust :

In all centrifugal and multi-stage turbine pumps having single inlet impellers, a considerable end thrust is developed which acts towards the suction end of the pump, and this must be counteracted in some way in order to ensure that the impellers revolve truly in their designed positions within each cell or stage.

The axial end-thrust occurs because water under pressure leaks into the clearance spaces on both sides of each impeller, between the impeller and its enclosing diaphragms. Now the area exposed to this pressure on the delivery side of the impeller is greater than the area on the suction side (by an amount equal to the area of the impeller inlet) with the result that an out of balance pressure sets up an end-thrust towards the suction end.

Total thrust = difference of two areas × pressure per unit area in the clearance space× no. of impellers.

Axial thrust can be countered in one of the following ways.

1. In the case of single stage centrifugal pump this can be eliminated by using a double entry impeller or by "drilling" the eye of the single impeller which must then have false "neck-rings" behind it.
2. In the case of 2-stage centrifugal pump the thrust can be eliminated by placing the impellers back to back e. g. Kirlosker DSM pumps.
3. By use of a thrust bearing.
4. By use of a balancer disc.

The common method of counteracting end-thrust in turbine pumps is by means of a balance disc which is keyed near the delivery end of the shaft and revolves in close contact with a seat fixed to the delivery chamber.

It consists of (a) a castiron balancing disc fixed on and revolving with the shaft or spindle and (b) a bronze seat fixed to the delivery cover (sometimes the disc is of stainless steel, and seat are fitted with renewable bronze rings which can readily be replaced when worn).

The periphery of the disc moves in close contact with the seat but does not touch it, there being a clearance C between them. the chamber A between the disc and its seat, is supplied with water under pressure from the last impeller through the clearance B between the shaft and the central bush in the delivery cover.

The axial thrust towards the suction end causes the clearance C to close up, so that less water leaves the chamberA than is trying to enter it, and the pressure in this chamber, therefore, increases until the axial thrust of the water on the balancing disc towards the delivery end balances the axial thrust towards the suction end. The balancing disc is, in effect, a water-cooled floating thrust bearing working without metallic contact and therefore not subject to wear except when pumping gritty water. There is a certain small loss of energy due to friction and due to escape of water from the balance chamber, but this loss does not amount to more than 1% or 2%.

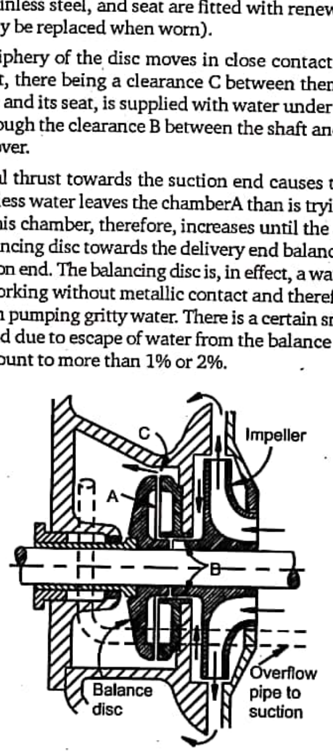


Fig. 18.8

If the axial thrust from the impellers decreases the clearance C opens and the pressure in chamber A decreases until equilibrium is again restored. The total axial movement of the disc and spindle is so small that it can hardly be measured.

The water from the balance chamber may be led away externally to waste or (as shown) it may be led back to the suction inlet through the overflow pipe.

Thrust bearing :

The ordinary bush or ball bearing cannot be used when a shaft carries a load in a direction parallel to its axis, e. g. in a turbine pump where the force on the suction side. Due to such difference in force the shaft experiences a thrust from the delivery towards suction. For moderate end thrust ball bearings with the races arranged side by side instead of consentrically are sometimes used. Tapered roller bearing is also sometimes employed. For heavy duty the most suitable bearing is the Michell thrust bearing named after its inventor, A. G. M. Michell. It consists of a number of segmental brasses or pads mounted in a ring attached to the bearing housing. Each pad is ridged on its back face, the ridge forming a pivot line on which the pad rocks bearing loads should be carried completely by the lubricating oil instead of the oil acting merly to reduce friction between metallic surfaces.

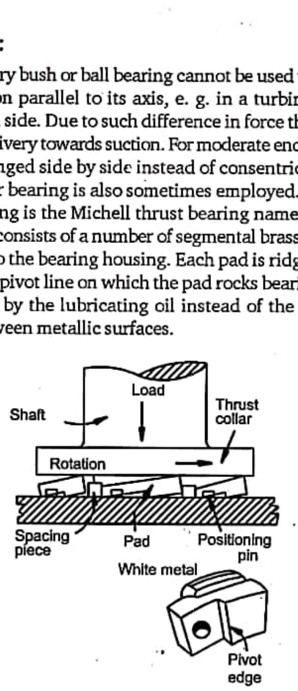


Fig. 18.9. Michell thrust bearing

The shaft having an integral forged collar (called thrust collar) bears on stationary pads that are pivoted. The moving shaft collar and the stationary pads do not come in contact as they are forcibly separated and kept apart by automatically generated tapered film of oil drawn from the normal oil supply. The tapered oil film exists when the shaft is running and is not dependent on any extraneous pressure from an oil pump.The pads are so designed that they tilt and cause the shaft to float on the oil film when a suitable speed is attained. They have white metal faces as white metal is less liable to damage from foreign matter which may be present in the oil. The pads are square with rounded corners and every pad generates a pressure oil film of a thickness appropriate to the load, the speed and viscosity of the lubricating oil. The thrust shaft may be disposed at any angle and standardised designs are available.

The working load depends on several factors, mainly diameter, length, peripheral speed, and viscosity of the oil.For bearings which do not start under full load upto 35 kgf/cm^2 of the bearing surface may be used but 25 kgf/cm^2 may be taken as the limit when starting under full load. The bearings will, however, work satisfactorily with considerable overload. The coefficient of running friction in a Michell thrust bearing is about 0.002; on the other hand the coefficient of friction of a good ordinary bearing is 0/036, about 18 times as much.

Laws govering centrifugal or turbine pumps :

These are similar to those governing centrifugal fans :

1. The quantity of water delivered by a given pump varies directly as the peripheral speed or r. p. m. of the impeller.
2. The pressure developed by each impeller varies as the square of the speed,
3. The power required varies as the product of pressure and quantity, *i.e.* as the cube of the speed. Thus, if the speed of the pump is increased to (say) 1.5 times the original speed, it will pass $1\frac{1}{2}$ times as much water; it will overcome $\left(1\frac{1}{2}\right)^2 = 2\frac{1}{4}$ times the head and with this increase intend and quantity it will require $(1\frac{1}{2})^3 = 3\frac{3}{8}$ times the power. These rules are only approximately true.

A centrifugal or turbine pump only works at its best efficiency when dealing with the exact quantity of water and the exact head for which it is designed. If the head is much reduced the quantity of water will increase appreciably and this will overload the motor. If a large pump designed for a particular head has to work for a small head temporarily, a good arrangement is to take out one or two impellers and replace them by dummy impellers.

A dummy impeller is one which has no vanes (except for joining the two "discs' constituting the impeller) and it therefore does not impart any pressure head to the water though the impeller itself rotates along with the shaft.

Characteristic curves for turbine pumps :

A "characteristic" or a "characteristic curve" is a curve which shows how the magnitude of one quantity varies with the changes in some other related quantity and it is by means of a set of such characteristic curves that the performance of a turbine pump can be most readily appreciated.

In the case of a pump the curves show the quantity delivered at various heads and the mechanical efficiency and power of the pump when running at a constant speed.

The efficiency curve : the efficiency of any machine is the ratio of power output to power input, and in the case of a *direct-driven* centrifugal or turbine pump

$$\text{Mechanical efficiency of pump} = \frac{\text{H. P. in water}}{\text{H. P. input to pump shaft}}$$

$$= \frac{\textit{water H. P.}}{\textit{Brake H. P. of driver motor}}$$

It will be seen from the characteristic curve (Fig. 18.10) that the curve rises from zero with a closed sluice valve to a maximum at normal duty and thereafter falls as the quantity increases. A pump should be run for a quantity which gives nearly maximum efficiency for a small variation in discharge. In other words the operating point of the pump should be on the flat portion of the curve depicting efficiency Vs. quantity. The maximum value of the efficiency varies with the size and make of the pump and it may range from 70 % for small pumps. say,20 l/s to nearly 80 % or so for large pumps of 80l/s or more.

The head-volume curve :- The head-volume curve is considered to be the true characteristic of the pump as it depends only on the impeller design and its speed. The other curves condition of internal surfaces, etc. The points to notice about the head-volume curve are :

i. the static head is somewhat less than the total head shown in the graph

ii. The curve is nearly flat for small discharge quantities but falls as the quantity is increased.

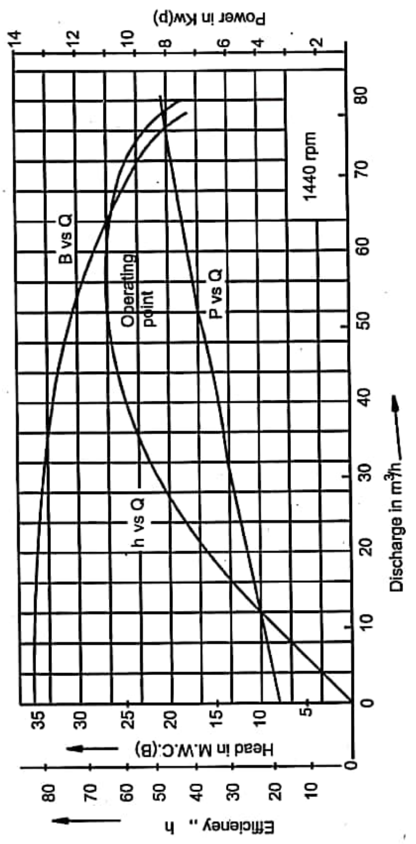


Fig. 18.10. Characteristics of a centrifual pump. Courtesy of M. A. MC.,Durgapur

iii. The maximum head developes when the sluice valve is closed and discharge is zero. Some pumps, however, have a curve which shows that the maximum head is nearly 10 % above the " sluice valve closed" valve. At the maximum value of head the pump passes some quantity but the head developed falls off gradually as the quantity but the head developed falls off gradually as the quantity increases. Such curve is said to have a "humped-back" profile. The falling head with increased quantity is attributed mainly to friction and shock losses within the pump.

The maximum pressure is fixed by the impeller diameter and its speed and we cannot obtain greater pressure head without increasing one or the other. It is, therefore, futile to attempt to use a turbine pump on a total head greater than that for which it is designed.

The brake-H. P. curve : It will be seen that the B. H. P. increases more or less uniformly with increasing quantities and it is possible to overload the motor if the head against which the pump is working is reduced. It can be further noted that the amount of overload is limited and does not become excessive.

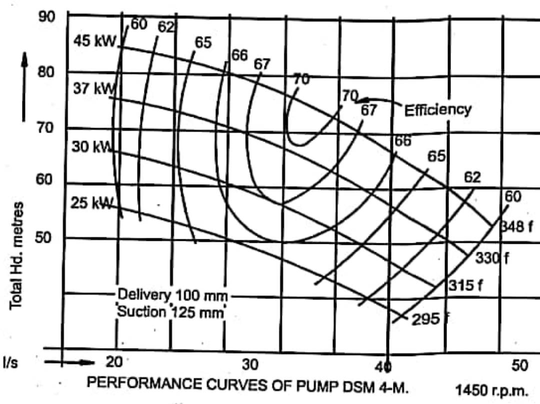


Fig. 18.11 Performance curves of pump, Kirloskar DSm 4-M.

Fig. 18.11 depicts the performance curves of DSM-4M pump, manufactured by Kirloskar Bros. Ltd. A close study of the curves will indicate that in a pump using impellers of 348 mm diameter, when the head is, say, 53 m, the discharge is nearly 47 l/s and the pump consumes 45 kW (pump alone), at an efficiency of 60 %. The same head is developed by a pump using 330 mm diameter, impellers but the discharge reduces to 43 l/s and under those conditions the pump alone requires 37 kW at pump efficiency of nearly 63 %. The actual power consumption by motors in each case will depend upon motor efficiency and also on efficiency of gears Screw pump. The Roto Pump is an example of the screw pump.

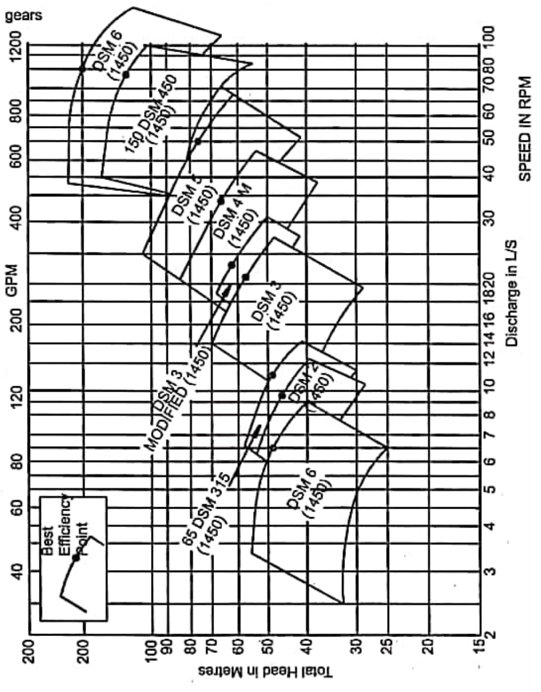


Fig. 18.12 Performance characteristics of DSM type Kirloskar pumps

This type of pump differs from the reciprocating and turbine pumps in its construction and working principle. It is a special type of electrically driven valveless, rotative pump which is inherently self-priming with a lift (suction head) of upto 8 m of water. It consists of essentially :

1. A rubber stator which has the form of a double internal helix and is a push fit in the machined cast iron barrel. The stator may be of natural or synthetic rubber or of hypalon, viton or other plastic material.
2. A single helical rotor of special abrasion-resisting or non-corroding steel(monel metal or stainless steel) :
3. Suction and delivery branches, ranging from 19mm to 75mm diameter.
4. Hollow driving shaft, running in ball bearings and transmitting an eccentric motion to the rotor by a coupling rod of high tensile steel.

The pump requires no foundation and will work on any gradient, and even when placed vertical.

Action of the pump :

It is an eccentric screw pump. The radial cross-section of the rotor is circular and is at all points eccentric to the axis, the centres of the sections lying along a helix whose axis forms the axis of the rotor. The pitch of the stator is twice that of the rotor and the two engage in such a fashion that the rotor section travels back and forth across the stator passage. The rotor maintains a constant seal across the stator. Whilst the rotor rotates in the stator, cavity formed between the two progresses from suction to delivery side resulting in uniform metered flow of water. The rotary motion creates an exceptionally high suction which exhaust all air from intake line resulting in immediate lift of water without need for priming.

Water which enters the suction branch is thus caught up in the space between the rotor and stator and is forced through the pump as the rotor revolves. A positive pressure is developed on the delivery side and there must be a free passage for the water before the pump is started up.

The roto pump is normally direct-driven by a three phase A. C, quirrel-cage induction motor running at 580, 720, 960 or 1,450 revs.per minute. The motor is switched direct on to the line. The pumps are available as single stage pumps (0.33 to H. P. of motor) or double stage pumps (10 to 20 H. P. of motor).

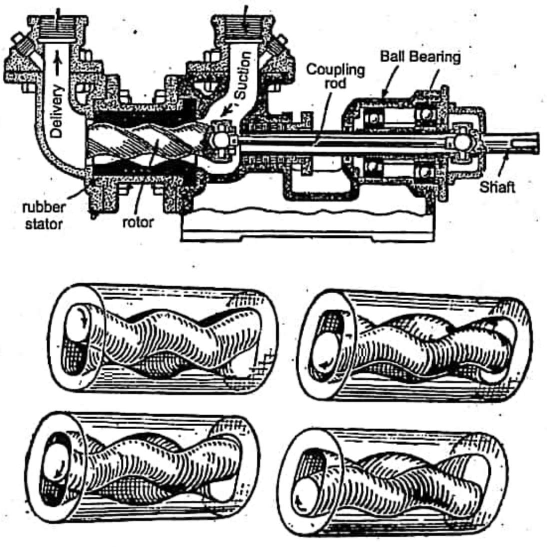


Fig. 18.13 Principle of the roto pumps and (bottom) a Roto pump in section

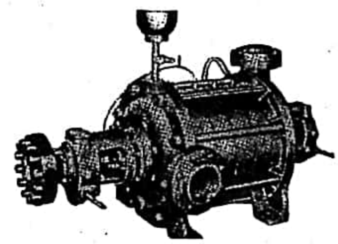


Fig. 18.14

Operating the pump :

i. The pump must never be run in a dry condition, or the stator will be immediately damaged. The pump must first be filled with water for lubrication purposes before the pipes are connected. Therefore, when pump is stopped, sufficient liquid is normally trapped in the pump to provide lubrication on starting again.

ii. When the delivery head exceeds about 30 m a hand-controlled valve, with a pipe leading back to the sump, should be provided below the non-return valve in the delivery pipe in order to relive the pressure developed when the pump starts up against a full delivery column.

The pump is inherently non-clogging and can deal with slurry or gritty water. It is capable of working on snore, *i.e.* it can handle appreciable amount of air alongwith water. In a pump this feature is of particular importance for face dewatering operations where it is necessary to pump out water from uneven surfaces and the suction pipe is partially uncovered.Use of Roto pump avoids construction of deep water collecting pits are necessary for centrifugal pumps which require the footvalve to be always submerged in water. In coal mines, it is ideal as a face pump and is extensively used at the advancing faces where the water contains coal particles of various sizes in large quantity. The pump is skid mounted, and can be easily shifted and installed as it needs no foundation. Repaires and replacement are, therefore, easy, with the help of even semiskilled workers in underground mines and pump need not be brought to surface. It has only one gland which can be arranged either at suction side or delivery side. Leakage of water through gland minimal.The pump is reversible, *i.e.* suction and delivery of the pump can be interchanged by merely changing the direction of rotation. The maximum head from all causes may be upto 90 m for a suitably selected pump.

As the pump is inherently non-clogging and self-priming, a regular pump attendant is not required. This saves manpower.

The internal velocity of fluid in Roto pump is negligible as compared to that in a centrifugal pump. This feature combined with lower pump speeds, minimises wear on housing and rotating parts due to erosion considerably, resulting in longer service life. The high efficiency of Roto pump is maintained over a wide range of delivery heads unlike in centrifugal pumps. This aspect makes it highly adaptable for face dewatering duties where fluctuations in delivery head are encountered.

The earlier models in single eccentric pumps had free fitting rubber stators but due to latest design development stators bonded to metal-sleeve have been introduced in the market. The cross-sectional drawing of a Roto metal bonded stator pump is shown in Fig. 18.13. The metal bonded torsion free stator has longer service life and this also results in higher efficiency of the pump and higher per stage pressure of 60 m.

Drill operated portable pump :

One of the centrifugal pumps which has no motor coupled to it but is operated by the electric coal drill in coal mines has proved quite popular at the advancing coal faces to deal with small accumulations of water which are normally bailed out by bailing majdoors. The pump, therefore, serves more as a substitute for water bailers rather than as a face pump. One make available in the market was Rana Drill Pump manufactured by Rana Sales & Service (Pvt.) Ltd. Chandigarh and it was on the pattern of Blagdon Durham portable drill pump which was imported until a few years ago.

This centrifugal pump is not coupled to any separate electric motor but the drive shaft of the pump has arrangement which engages with the drill chuck. The drill has to be held above the water level by hand, otherwise water may enter the motor. When power is switched on to the drill a firm grip of the latter is sufficient to overcome the starting torque reaction. The pump can work at a time for about 20 minutes, the normal rating of most of the coal drills. Longer operation makes the drill motor hot and cooling takes 30 to 40 minutes. It has no suction pipe, no external strainer or foot valve and it is self priming, capable of dealing with gritty water or slurry at the face. The delivery pipe is 50 mm bore and the suction is equivalent of 50 mm bore. It has a capacity of nearly 180 l/min at a total head of 12.2 m when operated by a coal drill of about 450 r.p.m. with 1.25 H. P. input. The head and capacity increase slightly with higher r.p.m. drills.

Pipes for conveyance of water :

Pipes for the conveyance of water may be made either of mild steel or cast iron. Of these materials, mild steel is generally to be preferred. It has a much higher tensile strength than cast iron and can therefore be much thinner and lighter in weight for a given strength. It is therefore much more convenient to handle, both in shafts and underground. It is also a more ductile material and less liable to fracture from shock loads, and it can be bent, when necessary, it can be threaded and where necessary flanges or small pipe lengths can be welded on to it.

On the other hand, cast iron offers greater resistance to corrosion, both because of its nature and because of the greater thickness of metal as compared with mild steel pipes of similar strength. In cases, therefore, where the water contains corrosive acids that would rapidly eat through mild steel pipes, cast iron is used in spite of its greater weight, lower tensile strength, brittleness, rigidly and difficulty in welding.

In recent years alkathene pipes are being used on an increasing scale mainly due to their lightness and low coefficient of friction.

The diameter of the pipe depends on the volume of water to be conveyed (the velocity generally ranging somewhere between 1 m and 2.4 m per second) and on the permissible head due to friction. The thickness of the pipes depends on the material used, the diameter of the pipe, and the head of water to be overcome.

Friction of water in pipes :

When water flows through pipes, a certain amount of pressure is invitably expended on overcoming friction, and the friction head (in metres) must be added to the vertical lift (in m) in order to find the total head against which a pump is working. This is especially true if the pipes are of small diameter and the velocity of flow is high. The frictional resistance is

1. Independent of the pressure.
2. Proportional to the area of surface in contact.
3. Proportional to the square of the velocity.
4. Depends on the nature of the surface in contact
5. Proportional to the density of the fluid.

D' arcy's formula for friction of water flow in pipes

$$Hf = \frac{4\ flv^2}{2\ gd}$$

Where, Hf is the head of water lost by friction, (m)

f is the coefficient of friction, (generally 0.01 for clean water and clean pipes).

l is the pipe length, (m)

v is the velocity of uniform flow, (m/s)

d is the pipe diameter, (m).

Support of rising main in mine shafts :

A common method of supporting the delivery column (or rising main) in a shaft is shown in Fig. 18.15. In this case, every third pipe has its top flanges resting upon a bunton across the shaft, and the pipe is secured by a mild steel strap or clamp, which is bent around the pipe and is bolted through the bunton at each side. The clamp has two screwed ends over which a steel plate is fixed, and the whole is firmly tightened by the nuts. It is advisable, where possible, to put in special buntons for the pipes, separate from those used for cage guides, so as avoid transmitting to the guides any vibration or shock due to water hammer.

Alternatively, with large heavy pipe column of cast iron every third pipe may be a special"stand-pipe' having supporting brackets cast on half-way in its length. These brackets rest upon short cross-joists or timbers, which are let into the shaft wall at one end and are bolted to a supporting girder or bunton at the other end. In this way, the pipes are supported independently of the flanges. In many cases, however, the stand pipes are omitted and the upper flange of every second or third pipe is then arranged to rest on the cross-pieces.

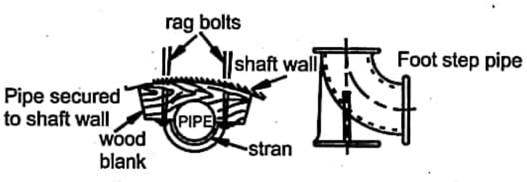
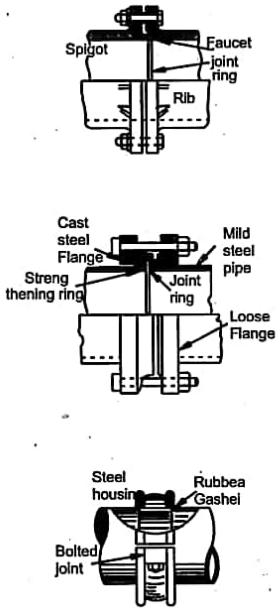
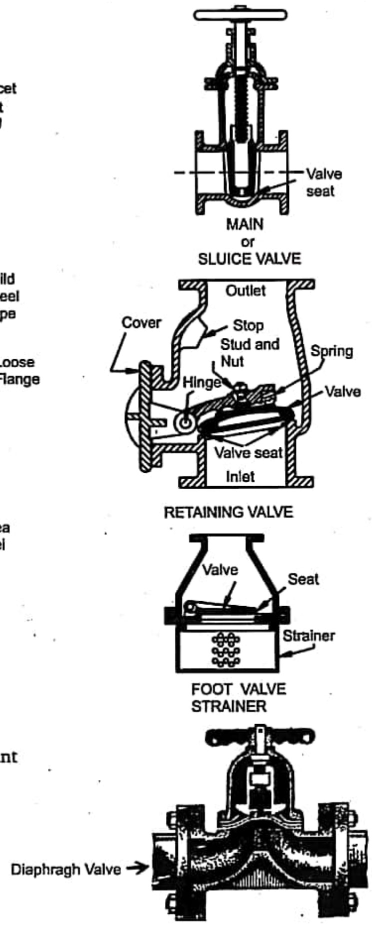


Fig. 18.15

Fig. 18.17 shows the different types of valves used in a pump installation.





i. a spigot-and faucet joint

ii. a loose-flange joint

iii. a Unicone joint

Fig. 18.16 & 8.17

Calculation of quantity flowing :

Venturimeter : A venturimeter is an instrument for measuring flow of water in pipes of 75-100 mm diameter and above. The meter consists essentially of a pipe which gradually narrows down to a throat and then gradually increases in size to its normal diameter. The diameter at throat is $\frac{1}{2}$ – 1/3rd of the normal diameter. There is no obstruction to the flow of the water and no moving parts.

According to Bernouilli's theroem

$$P_1 + \frac{U^2}{2g} = P_2 + \frac{V^2}{2g}$$

and the quantity $Q = A_1U = A_2V$

Where P_1 = Pressure head at one point up-stream side, of area A_1
U = Velocity at that point.
P_2 = Pressure head at the throat, area A_2
V = Velocity at the throat, down-stream side.
g = Acceleration due to gravity.

$$\text{quantity, } Q = C\frac{A_1A_2}{\sqrt{A_1^2 - A_2^2}} \times \sqrt{2g\left(P_1 - P_2\right)}$$

Where C is a co-efficient to allow for friction losses (usually value of C is taken as 0.99). A manometer is generally used to measure $P_1 - P_2$.

The quantity of water flowing can be calculated by arranging the flow over a triangular notch or a rectangular notch.

Rectangular notch formula

$$L = 1.11\ b\sqrt{h^3}$$

Trianguar notch formula (notch with 90° angle)

$$L = 0.884\ \sqrt{h^5}$$

Where L is quantity of water discharge, litres/min.

b is breadth of notch, (cm).

h is head of water over the notch, (cm).

A triangular notch is to be preferred as the wetted edge is due to the sides only and varies directly with head.

TURBINE PUMP CALCULATIONS :

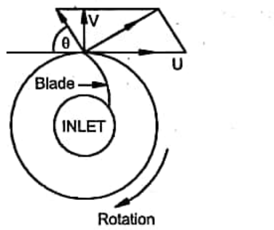
Head developed (allowing for blade curvature) :

Centrifugal pumps are normally fitted with impellers having *backward curved blades,* This having the effect of reducing the absolute velocity of the water at the impeller outlet for a given energy remaining to be converted for a given energy input, and the losses in the diffuser passages are reduced. A somewhat higher peripheral speed is needed for a given head, but the all-round efficiency of the pump is improved.

In Fig. 18.18 which depicts an impeller with backward curved blades,

$$\text{theoretical head per stage} = \frac{U(U - V\cot\theta)}{g} \text{ meters}$$

where θ = the angle made by the blade tips with the tangential direction
V = radial velocity of discharge, m/s
U = tangential speed of blade tips, m/s



Of this total theoretical head, more than half (in some cases 3/4ths) is generated directly as pressure head within the impeller and only the balance by conversion of kinetic energy to pressure energy in the diffuser channels. The exact proportions vary with curvature of blades.

Fig. 18.18 impeller with backwards curved blades

Example :

A turbine pump is required to work under the following conditions.

R. P. M. 1440; capacity 2700 litres per min ; total head from all causes 300 m ; angle of curvature of the impeller blades θ =30° (backward curved); manometic efficiency 0.7 ; radial velocity of water 2 m/s.

Find the number of impellers, their diameter and width of each impeller.

Ans. :

Assume an inlet velocity of 1.7 m/s which is normal. Area of suction inlet $\frac{2700}{1000}\times\frac{1}{1.7}\times\frac{1}{60}=0.026\text{ m}^2$

Diameter of suction inlet, D, gives $\frac{\pi D^2}{4}=0.026$

Therefore D = $\sqrt{0.033}$ m = 0.182 m = 18.2 cm

Diameter of delivery outlet is usually 2.5 cm less.

$\therefore$ Diameter of delivery outlet = 18.2 - 2.5 = 15.7 cm,

Before making further calculations we must, at this stage, assume either *(i) the impeller diameter*, or *(ii) the head per stage*. Let the impeller diameter be assumed to be twice the bore of the suction or near size.

Suppose the impeller diameter is 35 cm. so that its circumference is 110 cm.

Peripheral speed of the impeller = circumference × r. p. s.

$$=\frac{1440}{60}\times\frac{110}{100}=26.4\text{ m/s}$$

$$\text{Head per stage}=\frac{U(U-V\cot\theta)}{g}\times 0.7$$

$$=\frac{26.4(26.4-2\times1.73)}{9.81}\times0.7$$

$$=43.21\text{ m}$$

$$\text{Number of impellers required}=\frac{300}{43.21}=7\text{ (in complete number)}$$

$$\text{Area of impeller outlet}=\frac{\text{volume of water}}{\text{radial velocity}}$$

$$=\frac{2.7}{60}\times\frac{1}{2}=0.225\text{ m}^2$$

$$\text{Width of impeller outlet}=\frac{\text{area}}{\text{circumference}}$$

If b is the width in meters, b × 1.10 = 0.0225

$$\text{b (in meters)}=\frac{0.0225}{1.10}=.02045\text{ m}=2.04\text{ cm.}$$

Example :

A turbine pump has six impellers, 30 cm diameter running at 1440 r. p. m. The delivery branch is 15 cm bore and the suction branch is 18 cm bore. For what rate of delivery of water and head should the pump be suitable ? What should be the width of impeller outlet around its periphery, assuming a radial velocity of 2.4 m/s.

Ans. :

The capacity of this pump may be based on the area of the suction branch and the velocity, or on the area of the delivery branch and the delivery velocity.

The average figure for suction velocity is 1.68 or 1.70 metres /sec. Assuming 1.68 m/s.

Area of suction branch $\frac{\pi}{4}\times(0.18)^2\text{ m}^2=0.025\text{ m}^2$

Rate of delivery = 0.025 × 1.68 × 60 m^3/min.

= 2.52 m^3/min.

Area of delivery branch = $\frac{\pi}{4}\times(0.15)^2\text{ m}^2=0.0177\text{ m}^2$

$$\therefore\ \text{Discharge velocity}=\frac{2.52}{0.0177}\text{ m/min}=142.37\text{ m/min}$$

$$=2.37\text{ m/s}$$

Actual head per stage, $H_s=\frac{V^2}{g}\times K$

where K is manometic efficiency.

Now V = π D × rev/s

$$=\pi\times\frac{30}{100}\times\frac{1440}{60}=22.62\text{ m/s}$$

Let us assume K = 0.6, the normal value for manometric efficiency.

$$\therefore \quad H_a = \frac{V^2}{g} \times K = \frac{(22.62)^2 \times 0.6}{9.81} = 31.3 \text{ m}$$

$\therefore$ Total head = 6 × H_a = 6 ×31.3 = 187.8 m

$$\text{Area of impeller outlet} = \frac{\text{Volume of water}}{\text{radial velocity}} = \frac{2.52 \text{ m}^3/\text{min}}{2.4 \text{ m/s} \times 60}$$

$$= \text{Area of impeller outlet} = \frac{\text{Volume of water}}{\text{radial velocity}} = \frac{2.52 \text{ m}^3/\text{min}}{2.4 \text{ m/s} \times 60}$$

$$= 0.0175 \text{ m}^2$$

But area of outlet around periphery = π D b

$$= \pi \times 0.30 \times b \text{ m}^2$$

$$b = \frac{0.0175 \text{ m}}{\pi \times 0.30}$$

$$= \frac{0.0175 \times 100}{\pi \times 0.30} \text{ cm} = 1.86 \text{ cm}$$

Example :

(a) Calculate the dimensions of a turbine pump, and of suction and delivery branches, suitable for raising 2 m³ per minute of water against a total head of 240 m.

(b) What should be the power of a 4-pole, 3 ph., 50 cycles. a. c. induction motor for driving the pump coupled direct ? ignore blade curvature.

Ans. :

Assume an inlet velocity of 1.7 m/s = 102 m/min

$$\text{Area of suction inlet} = \frac{\text{Volume of water}}{\text{Velocity}}$$

$$= \frac{2 \text{ m}^3/\text{min}}{102 \text{ m/min}}$$

$$= .0196 \text{ m}^2 = 196 \text{ cm}^2$$

$$\therefore \quad \text{Diameter of suction inlet} = \sqrt{\frac{196 \times 4}{\pi}} = 15.8 \text{ cm}$$

Assume pump speed=1440 r. p. m. & manometric coefficient, K= 0.6

Assume the head per stage = 30 m

(assume 8 stage pump *i.e.* head/stage = $\frac{240}{8}$ = 30 m)

To find impeller diameter

$$\text{Head per stage} = \frac{V^2}{g} \times 0.6 \text{ where V is velocity in m/s}$$

$$= 30$$

$$\text{or} \quad V^2 = \frac{30 \times 9.81}{0.6} = 490.5 \text{ or } V = 22.15 \text{ m/s.}$$

Now $\frac{\pi DN}{60} = V = 22.15\, m/s$ when D is impeller diameter.

$$\text{or} \quad D = \frac{22.15 \times 60 \times 100 \text{ cm}}{\pi \times 1440}$$

$$= 29.40 \text{ cm}$$

1 m³ water weighs 1 tonnef = 1000 kgf

= 9810 N

$$\therefore \quad \text{Power} = \frac{240 \times 2 \times 9810}{60} \frac{\text{Nm}}{\text{s}}$$

$$= 78480 \text{ watts} = 78.5 \text{ kw}$$

Brake Power of motor (assuming 70 % pump efficiency)

$$= \frac{78.5}{0.70}$$

$$= 112.14 \text{ kW.}$$

QUESTIONS

1. Explain the working of a hydraulic lift and a siphon stating the principles underlying their working.
2. What are the advantages of centrifugal pumps over reciprocating pumps?

   Explain how head is developed in a turbine pump, and give the formula for the head developed in a turbine pump with backward curved blades.
3. What is an axial thrust ? What are the different methods of countering axial thrust? Describe the working of a balancing disc used in turbine pumps.
4. Show with sketches and only brief description how water pipe range is supported from the pump upto the surface in a pit mine

   Draw sketches to show a joint (a) for M. S. pipes b) for C.I. pipes.
5. Explain with sketch the woeking of a Roto pump. State the advantages of such pump.
6. In a mine the waterlogged workings have to be dewatered. The distance from the pit bottom sump to the dip-most face upto which dewatering will have to be done is 1200 metres. The vertical level difference between the bottom most point and the pit bottom sump is 150 metres. The total quantity of water in the lodgement to be pumped out is 12000 kilo litres and the percolation from strata in 24 hours is 2000 kilo litres. It has been decided to pump out the water in 30 days. Net hours of run of the pump, is 20 hours a day. Available pipe range is of 150 mm diameter. Find out the capacity, head and h. p. of the pumpset you need to install. Co-efficient of friction of the pipe range is 0.015. The overall efficiency of the pumping set is 75 %.
7. How many impellers and what diameter and width of impeller would you use for turbine pump to run at 1440 RPM and to raise 2000 l/min. against a total head from all causes of 308 m ? Given radial velocity V= 1.8 m /sec, manometric co-efficient = 0.7 and curvature of impeller vanes 30° (Backward curved).

O O O

CHAPTER – 19

FACE MECHANISATION

Deployment of machines at the working place from where mineral has to be extracted is known as face mechanisation.

Some machines drill holes for blasting which may be at the face of the mineral or in the rock of roadways. After blasting other machines load the broken mineral into minecars / tubs or transport it by loader shovels to a nearby chute. In some cases roof is also supported by machines but this chapter willbe developted to onlky the machines deployed for drilling blast holes and loading of broken mineral rock as roof supporting machines have been described in sufficient detail in Vol. I. In softer rock like coal, and load it on to conveyors.

During the last 40 years significant strides have been made in coal face machanisation but the metalliferous mines have not recorded any major steps in mechanisation of the faces, except in drilling, mainly because of the hardness of the ore and associated rock in the hangwall and footwall. This chapter deals primarily with machines used underground in coal mining and only a brief description is given of some machines used in metalliferous mining.

The two principal methods of extracting coal are :

(a) Longwall methods.

(b) Board and pillar methods.

Other methods which go by different names are derivatives of the above principal methods. Longwall is adopted only in a few mines in India whereas board and pillar is widely practised and is an established method.

The machines used in coal mines on the face for board and pillar mining and longwall mining methods include the following :

Coal cutting machines
Drills
Shearers
Coal ploughs
Continuous miners
Roadheaders
Mechanical loaders
Roof supporting machines
Water infusion pumps
Face pumps
Transport machines

Of these the conventional coal cutting machines (Fig. 19.1) have gone practically out of use.

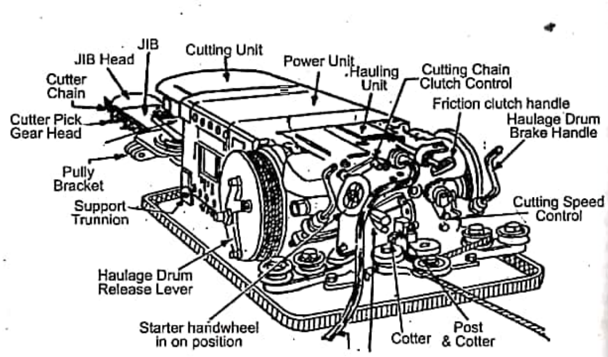


Fig. 19.1. A typical coal cutting machine, Shortwall type.

DRILLS :

Holes are drilled mechanically in rock and coal by drilling machines driven by compressed air or electricity. The drilling machine is known simply as a *drill*. Where compressed air is used a drill commonly employed is the jack hammer drill. For drilling in stone, drills operated by compressed air give better results, last longer than electrically operated drills and are in the

long run economic in operation. For drilling in coal, however, electrically operated rotary drills are universally employed as they give better performance in soft rocks and are more efficient with faster rate of drilling. Moreover they are lighter than jackhammers reducing fatigue of the operetor. Hydraulic drills are nowadays coming into vogue for hard rocks and they appear to be the drills of the future. The compressed air operated drills for underground use are : Jack hammers, (also called sinkets)

Drifters.

Stopers.

A Jack hammer, so familiar to mine workers, has been described in Ch. 10. It is a hand-held (and normally unmounted) drill used for vertically downward drilling upto a depth of 3 m ; hole diameter is generally 25 to 37 mm and rarely 50 mm. In a few cases a jack hammer may be mounted on an air leg. Though ordinarily used for dry drilling it can be adopted for wet drilling as well by replacing some parts.

A drifter is a mounted drill, generally designed for horizontal drilling. It is heavier than the jack hammer and is used extensively in metal mining and for tunnel driving. The widely used mounting is the column and arm and the drill may be used for wet drilling.

A stoper is a drill for drilling upward and derives its name from its widespread use in mine stopes. It is used normally for wet drilling.

A drifter and stoper work on the same principle as a jack hammer.

Air Leg :

Where compressed air is the motive power for drills, air legs may be advantageously used to mount the compressed air drills, (Fig. 10.9, p. 233). An air leg essentially a long cylinder in which a piston is actuated by compressed-air controlled valve which is also used to release the air pressure to lower the piston. The valve controls the feeding pressure on the drill. An air leg relieves the operator of the fatigue involved in holding the drill and keeping it pressed forward as the leg exerts an upward lift and a forward feeding pressure on the drill. The air leg does not increase the rate of penetration or feed and it is used for drifts upto 2 m in height.

In underground mines drilling rigs or jumbos have to be used for high speed drivage of large size drifts. The terms "Jumbo" and "rig" are often used synonymously, but jumbo is a portable carriage for underground use which has arms for mounting of 2 or more drills. The arms can be raised, lowered and slewed at any angle in position by hydraúlic or air pressure and all the drill steels are placed in the carriage. A Jumbo has a crew of 3-4 operators who perform various operations of setting a drill, drilling, dismantling, etc. at the face simultaneously.

Simba drill :

Manufactured by Atlas Copco Ltd, under trade name Simba, the drilling rigs for underground working in metalliferous mines are available as Simba Junior, Simba H-221 and Simba 300 series drill rigs. These are pneumatic tyre mounted or skid mounted rigs. The Simba Junior is intended for sub-level stopping where long vertical holes have to be drilled on a fan pattern or ring pattern. The holes may be up to a maximum length of 24m but are usually much shorter. Simba Junior rig consists of two skid mounted drill units and each skid carries a screw feed and a rock drill permitting a full 360° ring drilling pattern. The ring of the rig is nearly 1.5 m in diameter and lines with red lead are marked on it to indicate the angle (off vertical) for drilling. The rock drills and feed are operated from a control panel. The operating control panel together with lubricators and a drill steel rack is mounted on a third skid placed between the drill rigs.

The Simba Junior Special is based on the same principles as Simba Junior rig, but it consists of one skid mounted drill unit and a second skid equipped with the controls, lubricators and the drill steel rack.

The Simba H221 is a mechanised hydraulic drilling rig level stopping and is also designed for ring drilling (360°) in sub level stoping. The chassis for Simba H 221 is ydraulically powered and is fitted with a hydraulic system for rock drill (type COP 1038 HL) for long hole drilling. All feed movements are performed by hydraulic cylinders which are operated from a control panel on the feed holder. The feed is a operated from a control panel on the feed holder. The feed is a screw feed powered by a hydraulic motor. The crew for the rig consists of an operator and two helpers. Simba H 221 can be equipped with an electric remote control system for all rock drill operations.

Atlas Copco's rock drill COP 131 EL and BBC 120 F mountd on a srew feed are suitable for long hold drilling with 50 mm drilling bit and these are used on Simba drill rigs.

Simba 300 series drill rigs are designed for heavier duties where the conventional fan pattern or parallel hole pattern is to be adopted. The chassis is mounted on pneumatic tyres which have a four wheel drive, powered by a separate air motor. The screw feeds are air motor powered.

Cavo Drills :

These are light drill rigs for underground drilling. Cavo drill 550-89 is working at Mosabani copper and other mines. It incorporates the pneumatic traction unit from Cavo 500 loader and two booms (type BUT-6) with COP 89 D or COP 115 ED rock drills. The specifications of the drill are : Gross weight 6950 kgf, total length 8.83 m, total width 1.80 m, travelling speed 3.6 km/h, climbing ability 1 : 6 gradient. Cavo drills are equipped with a standard swing table of ± 20°. The drills in 550 series with diesel engine for traction (32 kW) or hydraulic traction motors are also availble.

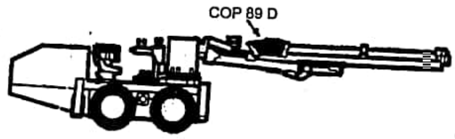


Fig. 19.2. Cavodrill model 550-89

Fig. 19.3 shows a light rotating boom model BUT 6 E of Atlas Copco with telescopic extension and can be used for Cavo drill 550. It is suitable for tunnel driving (small or medium size tunnel, about 7 to 16 m² cross section) and miscellaneous drilling jobs for underground mining application. The boom is capable of rotation through 360° and covers a circular radius of 2.8 m.

Another heavy duty drilling rig used in underground metalliferous mines is the ROC 306 drill rig of Atlas Copcodesigned for down- the-hole. drilling in sublevel stoping with large holes, cut holes in drifting, holes for raise driving and holes for drainage and ventilation (165 mm diameter). The drilling unit is mounted on a compact crawler carrier to facilitate operation aperation and travelling in confined areas. The minimum dimensions of the drift for its operation are 2 m width × 3.4 m height. The feed can be swung to drill holes at any angle, by a simple repining of a hydraulic cylinder. The rig accommodates one drill and the unit can drill holes of upto 100 m length with high precisio. Its climbing ability is 30° excluding compressor. Its gross wt. is 3800 kgf maximum height in transport 2.1 m, length 3.4 m and width 1.4 m. It is used in drilling hoies required for blasting, the recent trend in metalliferous mining.

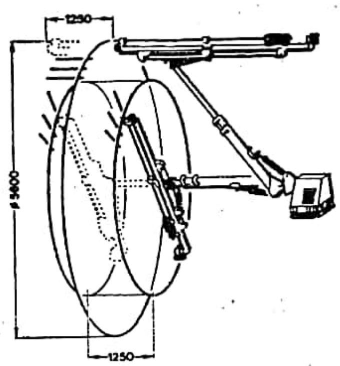


Fig. 19.3. Rotating boom model BUT 6 E.

The feed mechanism can be titled through a total of 135° without reconnecting. The boom can be raised 55° above the horizontal and lowered 20°beneath it. In other words it can describe a total are of 75°. Feed extension A= 460 mm (Fig. 19.4).

Vishwa Industrial Co (P) Ltd. is marketing heavy duty drills with the trade name Bison Drills. Bison crawler drill is illustrated in the photograph. On the heavy duty crawler based chassis can be mounted a choice of rugged drill booms and rotary percussive or rotary drills below centre of gravity makes it suitable for operating in cross gradients upto 1 in 6 and inclines upto 1 in 4. High torque hydraulic motors are fitted to each track.

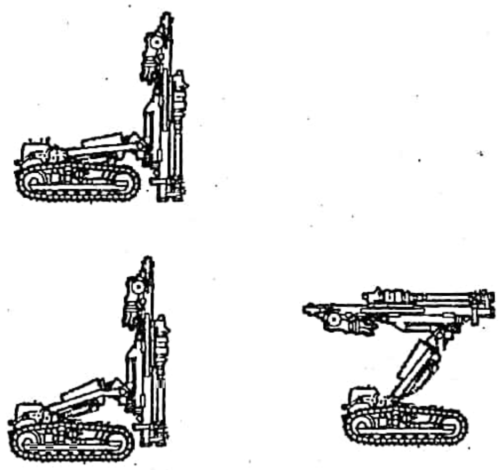


Fig. 19.4.

ROC 306 drill rig of Atlas Copco for down-the =hole drilling of 165 mm diameter holes in underground metal mines.

Hydraulics: The machines power pack consists of a motor driven variable displacement constant pressure axial piston pump with a maximum working pressure of 172 bars. The pump is suitable for use with 60/40 oil in water emulsion or mineral oil. Interlocking arrangments ensure that the machine cannot move when drilling and it cannot drill when being moved to a new location.

Specifications of the drill are :

Overall tramming length 7.5 m; overall width 1.2 m : crawler track width 300 mm ; track centres 900 mm; overall height (with operator seated) 2 m: overall weight 9.15 te; tramming speed (level ground) 1 km/hr maximum floor gradient 1 in 4 ; maximum cross graddient 1 in 6.

The in *built safety features* include : Fail safe braking; "Dead man's" type foot pedal provides an immediate "stop" to the crawler power when released ; oil temperature switch; oil level switch; 4 strategically placed emergency stop switches ; operator stop/start switch ; earth leakage protection ; pilot circuit including self start protection.

The electric motor is rated 49 kw at 1500 rpm.

The Company also markets an **Augerborer** for use in soft rock.

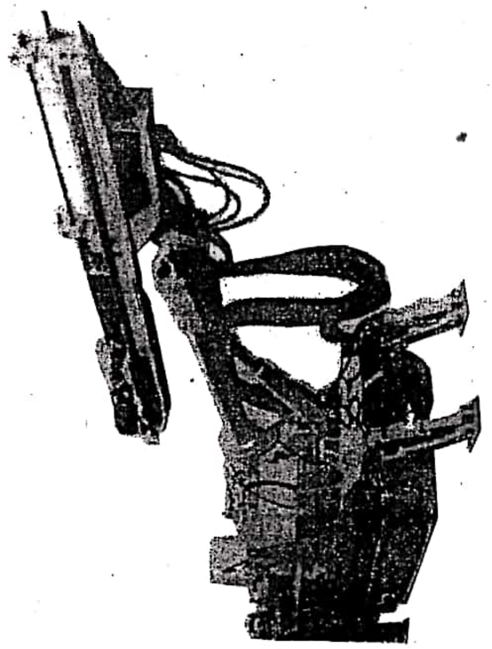


Fig. 19.5. Bison Crawler Drill (with single boom) Courtesy of Vishwa Industrial Co. (P) Ltd.

Coal drill :

The drill used for drilling holes in coal and similar soft rock is electrically operated and is of rotary type. Such drills are manufactured by a few companies like MAMC, Voltas (under brand name VOLKOL S-50), Chanda & Co. and others. Coal drills manufactured by MAMC are of two types : Type CD-1 with steel body and CD-2 with aluminium body. All the spare parts of these two drills are inter-changeable and only the casing differ from each other. The steel body drill weighs 21.5 kgf and the aluminium body drill weighs 17.5 kgf. The coal drill is used not only for coal but other rocks in coal mines except very hard grades of stone. The drill essentially contains a squirrel cage induction motor in a flame proof casing with two handgrips symmetrically placed on two sides of the machine. The switching device is placed under the right hand grip of the motor casing while the cable entry is belowthe left hand grip through the plug and socket arrangement. For remote control the drill is furnished with a single pole pilot switch. The output power of the motor which has two poles is 1 kW, half hourly rated and is wound for 125 volts, 3 phase, 50 cycles, AC supply. A double stage reduction gear box helps to utilise the machine to drill in coal or stone of various hardnesses. With changeable gear pairs the drill can be used to give speeds of RPM of 600, 500 or 430 depending upon the gear box used. The power is supplied through the 6.5 mm², 5-core trailing cable. 100 m long from a drill panel which receives power at 550 V by armoured cable and steps it down to 110 V. The drill is operated by remote control and a trigger switch on the drill which is pressed by the operator when operating the drill closes a pilot circuit having a voltage of 25 V. This closure of circuit operates electromagnetically the main switch at 110 V in the drill panel.

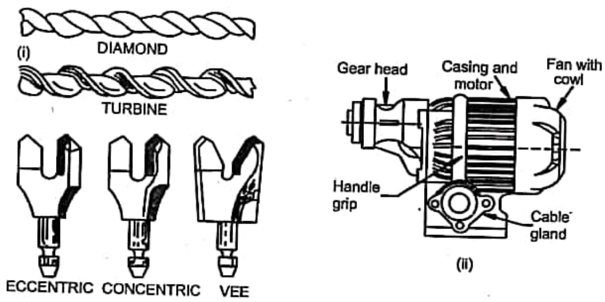


Fig. 19.6. (i) Drill rods for rotary drill (ii) An electric coal drill (iii) common types of drill bits for rotary drills.

A cowled fan mountedon the drill together with the fins on the cover helps to bring down the temperature of the drill.

The drill rod is of diamond section for drilling in coal and it fits in the drill chuck by a bayonet joint but the bit is attached to the rod by a wirenail. Tungsten carbide tipped drill bits are used and of these the eccentric type bit is employed for coal. The rate of penetration of the bit in coal is generally 1.5 m/min. The drill is capable of drilling nearly 80 holes. each 1.5 m deep in a shift of 8 hours. For drilling in stone. a coal drill is sometimes used by changing the gear box to reduce the speed of the bit to nearly 250 rpm and turbine section drill rod replaces the one of diamond section. For large scale drifting, mechanical power feed is the common practice as the arrangement not only supports the drill but it also relieves the drill operator of the exertion to keep the machine pressed against the rock.

The Company Eimco Elecon (India) Ltd. is manufacturing auger drills. One such drill marketed under the trade name Model 625 SDL base auger drill has proved useful in coal and is working in some coal mines of coal India Ltd. Its **main specifications** are :

Electric motor – 65 H. P., 550 or 440 v, 3 phase, FLP motor

Hydraulics – Pump geared type

traction motor High torque, low speed, five cylinder (radial motor)

Wt. of machine – 9500 kg approx.

Speed of machine – 1.5 km/hr

Drill unit – rotary hydraulic suitable for 45 mm diameter drill holes upto 2 m depth at 1.25 m height

Gradient :

upto 1 in 4 to rise or dip

upto 1 in 8 cross gradient

The hydraulic fluid is cooled by a radiator with a power driven fan for low operating temperature. The hydraulic system is protected by fluid level and temperature switches.

The operator sits in a comfortable central position on the machine where he is safe and has an excellent view of his working area with all controls readily at hand.

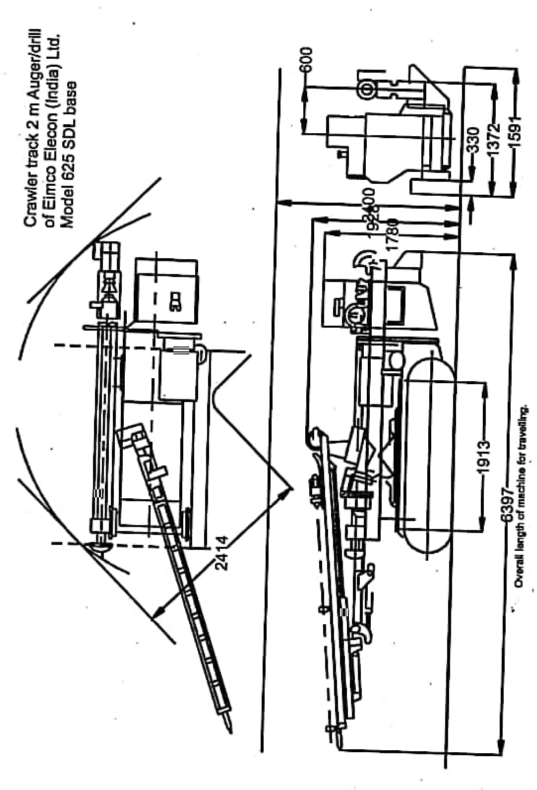


Fig. 19.7

The standard fitments include : Grade retardation on traction motors ; ...w oil level trip switch ; high oil temperature trip switch; remote start push ...utton station.

In a coal mine of Western Coalfields Ltd. the drill was used in galleries 4.5 in wide and 2.6 m high; the coal yield per blast was nearly 25 te.

Mechanical loaders :

Mechanical loaders which are commonly used are of the following types :

1. *Slusher loaders*
2. *Duck bill loaders*
3. *Gathering arm loaders*
4. *Front end loading shovels and side load dumpers*
5. *Flight loaders.*

The slusher loader has been described in Vol. I. Duck bill loaders are not used in our mines. Gathering arm loaders are commonly used where face operations are mechanised in board and pillar method of working, because of the ease in flitting, compactness, and suitability to work on gradients upto 1 in 5. The front end loading shovel in uncommon for coal for drifting and is common in metalliferous mines. Flight loading has not been practised so far, but is likely to be introduced in our mines in the near future.

Gathering Arm Loader :

Fig. 19.8 shows a typical gathering arm loader. It consists of three principle units *(i) a gathering head (ii) a central crawler mounted chassis, and (iii) a rear boom or jib.* A chain conveyor extending from the gathering head upto the boom end is the transporting medium, conveying coal gathered from the face to the receiving mine car, tub, conveyor or shuttle car.

The gathering arms are operated by twin crank discs. These discs are flushed of with the working surface of the head. There is a separate driling motor for each of the arms. The rate of loading varies with models, and commonly used models are for 1 te/min to 4 te/min capacities. The rate of loading depends upon the number of strokes/min. and the conveyor speed is also related to the rate of loading of the gathering arms. Three variable speeds are available on the loader for gathering speeds of the arms.

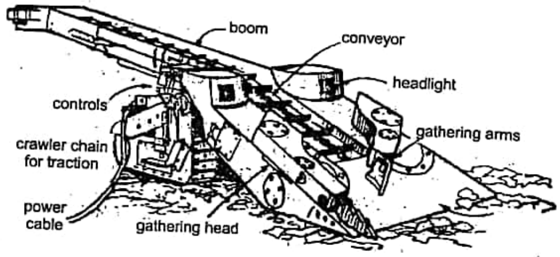


Fig. 19.8 Gathering arm loader.

The ramp of gathering head can be raised or lowered usually through 0.5 m. During flitting the head is kept elevated and during normal loading operation the head rests on the floor. Hydraulic jacks are used for elevation of the gathering, head and they are controlled by the operator from his position at the controls.

Flitting of the machine is possible by crawler chains and each chain is operated by a separate motor through gears. The motors are housed in the central body of the loader.

The conveyor chain is supported by the delivery jib or boom. This jib end, and along with it the conveyor, can be raised or lowered through a vertical range of 0.6 to 0.9 m by hydraulic jacks. The jib can also be slewed through a horizontal plane so as to make an angle of 45° to the axis of the machine on either side. This enables the conveyor to discharge coal on either side of the centre line of the loader. The slewing is possible with the help of either-tension ropes or jacks hydraulically operated by oil pressure.

At the face, as the loader loads coal it has to move forward to be close to coal heap and in a gallery width of 4 to 4.8 m to and fro movements of the machine are frequent for clearing up the gallery. Clearing up the last remains of the face coal usually takes a long time.

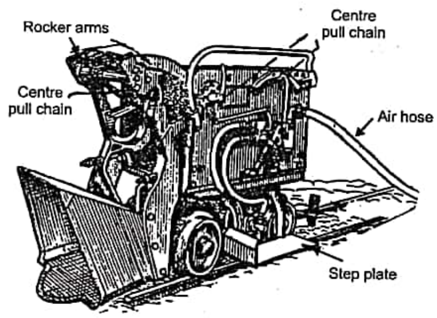
A loader is normally used to discharge into conveyors or shuttle cars as the discharge into a tub requires high elevation of boom of the loader and it is not practicable in most of the cases. The loader has sometimes to load the coal by double handling and this reduces its output considerably.

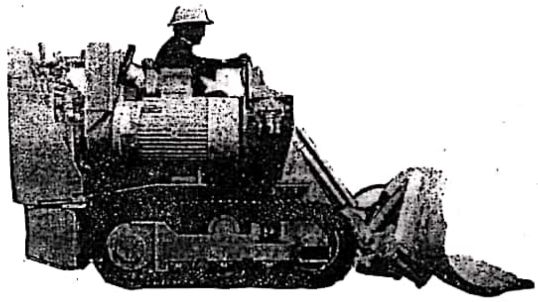
Rocker Shovel :

A rocker shovel consists of *(i) a chassis fitted with crawler chain, pneumatic tyres or wheels for traction (ii) a loader bucket shovel at one end, (iii) motor for power, and (iv) necessary controls for fitting and for operation of the shovel* (Fig. 19.9). The crawler chain equipped shovel is best for a dipping roadway (max. gradient one in four) as the rocker shovel,or conveyor. In a rising drift or roadway having a gradient of 1 in 6 or more, tyred rocker shovel, when loaded, raises problems of braking and control and a crawler chain shovel is to be prefered.

A rocker shovel has a bucket or shovel at its front end which may be used for dumping the contents to one side (known as side dum loader) or behind it. Fig. 19.9 shows a rocker shovel model Eimco 21 manufactured by Eimco-Elecon. The operator pushes the bucket into the pile of the blasted rock. The bucket scoops the material with an upward-and over motion for discharge into a tub or conveyor behind the loader. Such front end loader requires sufficient head-room height for its upward movements. Some of the specifications of Eimco-Elecon front end loader shovel, model 21, are as follows. The model is used in many metal mines :

A Overall operating width	1067 mm
B Overall Length (caging)	1397 mm
C Overall length bucket down	2210-2286 mm
D Overall height bucket down	1422-1549 mm
E Headroom height	2235-2515 mm
F Discharge height of bucket	1321-1727 mm
G Discharge distance behind loader	457-711 mm
H Cleanup range	2286-2489 mm
Range of track gauge	460-1220 mm
Air pressure range	4.2-7.0 kgf/cm^2
Air hose size	25-32 mm
Bucket capacity	0.21-0.28 m^3
Motors (air)	2 Nos
Weight, completely assembled	3266 kgf
Air consumption	8.5 m^3/min





Side discharge loader

Fig. 19.9

With well fragmented rock in a convenient pile and the tub placed close behind the rocker shovel the average loading sysle takes about a minute.

The model 21 rocker shovel provides cleanup for tunnels or drifts from to 3.7 m in diameter. For drifts of less height, smaller capacity shovels are also available, but for such drifts of less height, a side dump loader with crawler chains or pneumatic tyres permitsuse of large capacity bucket.

Cavo loader 310 :

Loaders of cavo series are extensively used in matalliferous underground mines such as jaduguda, Rakha, Surla, khetri, Zawar and other mines. Fig19.10 shows principal dimensions of Cavo drill model 310. The machine can be equipped for remote control operation also (e.g. at Khetri copper mines). Cavo loader 310 has a bucket capacity of 0.13 m^3 and two traction air motors of 6.4 kW each. Air motor for bucket is 7.2 kW. The dumper is steered by running the two traction motors at different speeds. If the motors are running at the same speed, but in opposite direction of rotation, the loader turns on the spot. Travelling speed is 85 m/min. Air consumption 133 /s. The air motors are of vane type for traction as well as for bucket. The loading capacity is 40 m^3/h in well piled fragmented rock; total weight 3150 kgf (with empty bucket). The loader is provided with tyred wheels.

Over-head front end loaders of Atlas Copco are available in LM series also and they are equipped with steel wheels for movement on rail tracks and with vane type air motors. For small machines, the bucket capacities of LM type loaders are (m^3) 0.14, 0.26, 0.40, 0.60 and a loading capacity in m^3/h of loose material is 35, 60, 85 and 120 respectively.

Machines used for loading at the working face and equipped with remote control operate normally 4-6 m from the chute.

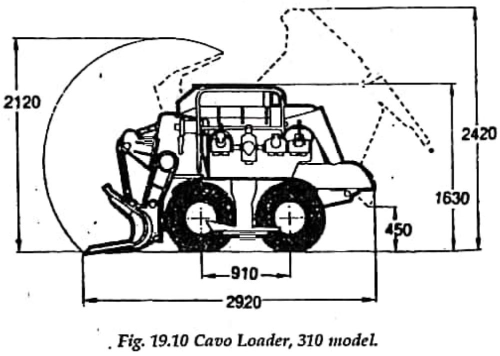


Fig. 19.10 Cavo Loader, 310 model.

Fig. 19.11 shows 2000 DL drill-cum-loader of Atlas Copco for underground drifting and tunneling. It can be adopted for drilling as well as for loading. The machine is electro-hydraulic and compressed air is not used.

The drilling portion can be dotached after drilling operations are over and only the drilling portion can be replaced by a loader bucket for loading the blasted material. Specifications :

Wt. 12.65 te, length 6.7 m, Width 1.6 m, Ht. to seated operator's helmet 2.2 m, Bucket capacity 0.5 m^3, Loading capacity 0.7 m^3/min. Hydraulic pump mptor 48.5 kW, Ground speed 36 m/min. Max. gradient 1 : 4. Max. roadway size 4.3 m width × 3.6 m ht.

Side dump loading is the standard practice. Drill rod of 2000 DL are 2700 mm long and the advance (feed travel) is 2.4 m. The Rock drill suitable is COP 420 R or COP 1033 HD.

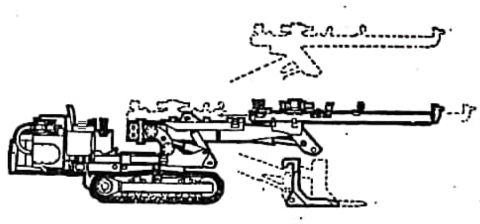


Fig. 19.11 Drill-cum loader model 2000 DL of Atlas Copco (India) ltd.

Continuous miner :

Continuous miners are heading machines which combine the operations of cutting coal or soft rock and loading it simultaneously into mine cars, shuttle cars or conveyors without the usual unproductive breaks that are inherent in the conventional mining which follows a definite cycle of operations. Basically these machines include a cutting/ripping unit, a coal gathering unit and a delivery unit. These cutter loader machines work on gradients upto 15°, in seams of thickness upto3 m (rarely upto 4.5 m) and give a progress of 20 to 40 m in a day or nearly 900 m per month, in a coal heading. The ventilation system has to be efficient to keep pace with the rapid advance of the face.

A continuous miner essentially consists of 5 main parts hinged to the chassis viz.

1. a bottom carrying the cutting head,
2. a gathering head sited below the boom,
3. a frame which supports the delivery jib,
4. a jib which can be slewed,
5. propelling unit which consists of crawler chains.

The two main types in use are *(1) the borer type, and (2) the ripper type.* The borer type gives very high rates of advance and depending upon hardness of coal, can penetrate at the rate of 0.3 to 0.6 m/min.

Description of one continuous miner, borer type,is given here to instruct its construction, manner of working and performance.

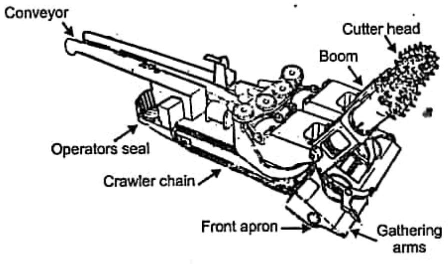
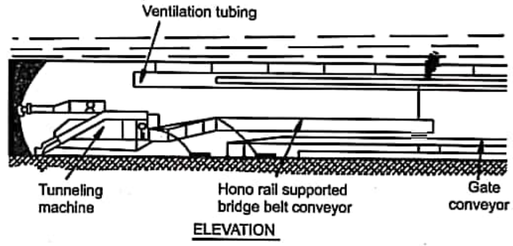


Fig. 19.12. A typical machine; borer type continuous miner.

Alphine Miner model AM-50 (of Voest Alphine, Austria) :

This continuous miner of borer type is mounted on crawler tracks driven by electric motors through worm gearing and negotiates inclination up to = 16°. It is a selective sutting machine designed for operation in medium hard rock up to an unconfined compressive strength of about 800 kgf/cm^2 (Hardness upto 4 on Protodyakonov scale). In selsctive heading the fully articulated cutter boom can cat any required circular or trapezoidal cross-section within the limits of 3.75 m in height and 4.8 m in width, if the machine is located on the centre-line of a heading. The cutting unit of the miner consists of the cutter head laced with carbide tipped picks and driven by a water cooled 120 kW motor through spur-gear. Selection of the cutter head, picks and speeds is governed by the breaking characteristics of the rock. The cutter boom consisting of cutter head, gear and motor is mounted on the turret which provides horizontal and vertical positioning of the cutter boom.



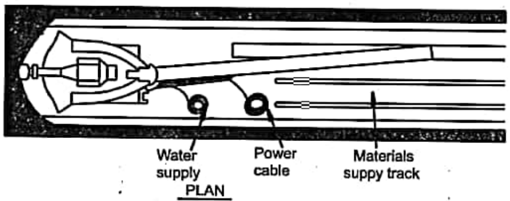


Fig. 19.13. A general layout at the face with continuous miner.

The loading system consists of a hydraulically adjustable apron with two gathering arms. These move the ripped coal/rock into the single-strand chain conveyor located in the centre of the machine. The gathering arms are through bevel gearing by the chain conveyor reversing shaft. The chain conveyor is actuated by two 13.24 kW motors.

Hydraulic pressure for operation of the turret jacks.,Loader head jacks, rear clamping unit and the slewable conveyor belt and other optional equipment is provided by a hydraulic reservoir with built-in axial piston pump and motor on the machine. The hydraulic system works at a pressure of 200 bars.

Water sprays on the cutter picks suppress the rock dust produced during operation. If conical picks are used, these are rotated by the sprayed water effect.

Normally the AM-50 is operated with short stage conveyor following the single-strand chain conveyor. A discharge chute or belt suspension fitted on the discharge end of the chain conveyor serves as the link between belts. A universal joint design enables the machine to swing 90°each way from the stage belt line. To overcome uneven floor, vertical play of 12° is possible.

Some of the technical specifications of AM-50 are as follows :

Overall height, 1645 mm ; overall length, 7500 mm ; overall width without loader head, 1910 mm ; total weight, 24 tef ; ground pressure under track, 1.4 kgf/cm².

Cutting range (without cutter boom extension)	
Max. height above floor level	3758 mm
Max. depth below floor level	100 mm
Max. width of excavated section	4800 mm
Length of cutter boom	3200 mm
Chain speed (60 c/s motors)	6 m/min.
Max. passable gradient	± 16°
Chain speed of conveyor	1.1 m/sec.
Total power of all installed motors	187 kW, 550 V.

Coal ploughs :

The coal plough is employed on a non-cyclic longwall face with a prop-free front. A coal plough is a machine which is mounted on an armoured chain conveyor and cuts a slice of coal, 100 mm to 200 mm from the entire working height of the seam during its travel along the face. The cut coal is loaded on the conveyor by a ramp which is a built-in part of the plough and which follows the cutting teeth. Separate arrangementfor loading is therefore not necessary. Coal ploughs are used on a large scale in Germany, holland and other European countries. The seam thickness suitable for its operation is from 0.6 m to 2 m. The plough consists of 4 or more teeth in two nearly vertical planes fixed to a base plate which is mounted on the latter through a chain. The motor is usually situated only at the haul gate end of the conveyor. The hauling chain pulls the plough up or down the coal face and it is threaded through a 115 mm diameter tube attached to the conveyor all along the face. The two ends of the chain pass over the two sprockets, one at each end of the conveyor, and rate finally attached to the plough. The plough cuts the thin web of coal during travel in either direction. The conveyor and the plough are held up to the face by hydraulic jacks placed at intervals along the goaf side of the face conveyor.

Two types of ploughs are in use. The slow speed plough with a speed of 3.0 to 18 m/min. and the high speed plough with a speed of 18 m to 30 m/min. The normal pull required for the plough during cutting may be 20 to 30 tef in hard coal seams and 5 to 20 tef in comparatively softer coal.

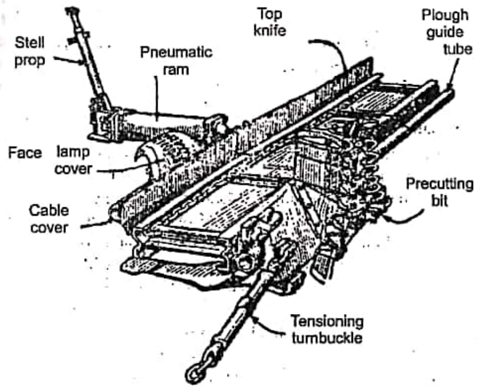


Fig. 19.14 Coal plough.

Where the coal is hard, infusion of the coal face precedes the ploughing operation. Water infusion is a process of drilling 1.2 to 1.5 m deep holes spaced 3 to 6 m apart. along the face at nearly 45° to it and forcing water at high pressure (6 to 10 kgf/cm²) through them. A water infuse face not only renders ploughing easier but it also helps in effective coal dust supression. Some coal seams are not amenable to water infusion even at pressures of 350 kgf/cm² due to hardness of coal or lack of cleavage, usually seams which can be ploughed will accept water infusion.

The ploughability of a coal seam is determined in Britain by the impact strength index (I.S.I.) which is a modification of a test devised in Russia by Protodyakonov. The penetrometer developed in Britain,however, offers a better assessment of the ploughability of coal. The penetrometer consists essentially of a solid rod 13 mm diameter (usually called an indenter) and a hydraulic ram which forces the indenter horizontally into the coal, The whole being supported by a special portable staking system. The resistance to penetration of the coal is recorded on a gauge and readings are taken at every 6 mm of penetration up to a depth, if possible, of 150 mm.

The Dintheader and Roadheader MK2A are described in Vol.I, Chapter7, and will therefore not be duplicated here.

Shearer loaders :

A natural and logical development of the coal cutting machine is the cutter loader, the latest example of which is the shearer loader, commonly referred to as simply "shearer' in mining terminology. It is used only on longwall coal faces. Most of the newly opened coal mines of Coal India Ltd. adopting longwall methods of mining, have shearers as the standard face equipement. M. A. M. C., Durgapur manufactures shearers in collaboration with Anderson Strathclyde of U. K.

The shearer cuts coal and loads it onto an armoured face conveyor on which it (shearer) is mounted. The shearer is mounted on a skid plate provided with bearing pads which rest on the AFC. The machine consists of 3 units.

(a) *The cutting unit or gearhead*

(b) *The haulage and control unit,* and

(c) *The motor unit.*

The following description rlates to the model AM 500 manufactured by Anderson Strathclyde.

The cutting unit : It houses a special gearbox which drives a horizontal staft projecting towards the face. Shearing drums of shearing discs, laced with cutter picks are mounted on the extension of the shaft. The shearing drum can be raised or lowered in a vertical plane with the help of a boom or ranging arm for cutting at various heights and is therefore known as ranging drum. There may be just one ranging drum at one end on the machine which is then known as single ended ranging drum shearer, SERD shearer, or there may be two ranging drums , one at each end of the machine, which is then called double ended ranging drum shearer, DERD shearer. This arrangement of one drum at each end in a DERD shearer pemits coal cutting as well as loading on the conveyor when the machine is travelling in either direction. Such bi-directional machine is specially advantageous where the nature of roof is such that speedy erection of roof supports is essential. The DERD shearer has its gearhead of the same type as the SERD shearer, permitting interchangeability.

The minimum drum diameters in different models of Am 500 series vary from 1370 mm to 1650 mm and the corresponding reach (in mm,trunnion to shaft centres) varies from 998 mm to 2776 mm. The cutter drums are fitted with helical vanes to assist the loading of cut material and dust suppression water is fed through internal drillings in the drum shaft and the drum to jets on the vane. A hollow shaft venturi system ventilates the face side of the

cutting drum to dilute any accumulation of inflammable gas. The thickness of the seam suitable for a DERD shearer in our country is shown in Fig. 19.16. For a short single ended shearer the maximum seam thickness is 1.37 m.

During travel of the machine the leading drum cuts at the roof level and the trailing drum, at the floor level. The operation of shearing starts from one end of the longwall face. The shearer drum placed nearer the floor cuts nearly half of the seam thickness and the remaining overhanging half is cut by the other drum ranged near the roof, when travelling in the same direction.

The shearer cuts from the coal face a slice of thickness 450 to 650 mm. A plough is attached to the front end of the shearer through an articulated joint and the cut coal is deflected by the plough on to the AFC. The shearer cuts the coal, loads it on the AFC and simultaneously travels on theAFC.

Haulage and control unit :

The shearer may be fitted with i. Roll Rack or other chainless haulage systems, or with ii. chain haulage. Whether equipped with chain haulage, or chainless haulage, the AM 500 incorporates hydraulic system with a pump box and gearbox. The internal arrangements are such that the pump box is devoid of all gearing. An automatic speed control system is included which allows the maximum haulage speed to be pre-set manually, after which the auto control system will automatically regulate the speed according to the load on the electric motor. The haulage gearbox (unit 3) houses the main pump drive. The chain haulage is fitted with sprockets to suit either 22 mm or 26 mm haulage chain depending on the pull required.

In the case of chainless haulage *i.e.* Roll Rack shown in the figure the power pack (unit 7) supplies hydraulic powert to motors in the traction units which may be underframe mounted (unit 8) or end mounted (unit 9), or located in an intermediate box (unit 10). The traction units for the chainless system are essentially gearbox containing high torque, low speed hydraulic motors driving final drive shafts. For roll rack, each of these is fitted with a roller wheel designed to engage with the teeth on the rack bars.

Radio Control : An integral radio control system is available providing remote control of most machine functions for one man or two men operation.

Gearhead gearboxes : This (unit 4) houses the train of gears which takes the drive from the electric motor to the ranging arm. It enables control of the drum speed. Cutting drum speeds down to 19 rpm are available.

Ranging arm : The ranging arm (unit 5) contains the gearing which transmits power from the gearhead gearbox to the cutting drum and is trunnion mounted in an outboard position. The arm is ranged in a vertical plane by means of a hydraulic cylinder. The total available vertical movement of the arm is 90°. Five lengeths of ranging arm are available.

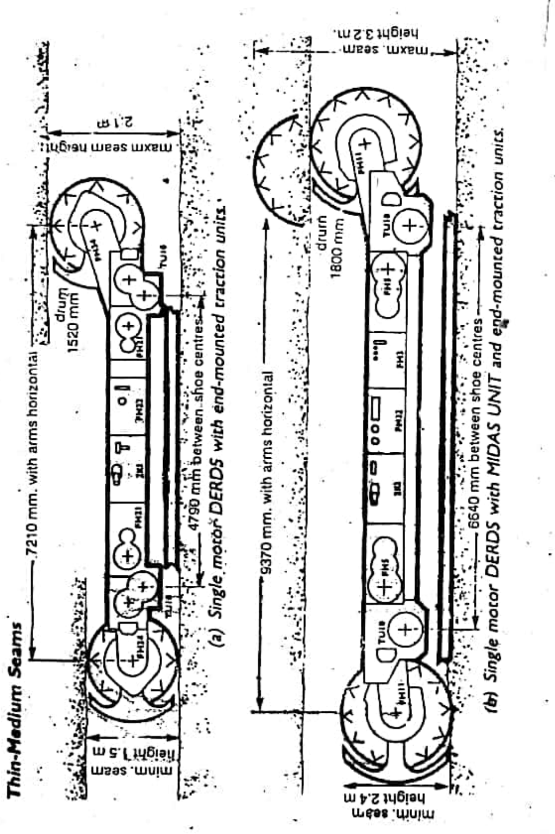


Fig. 19.15

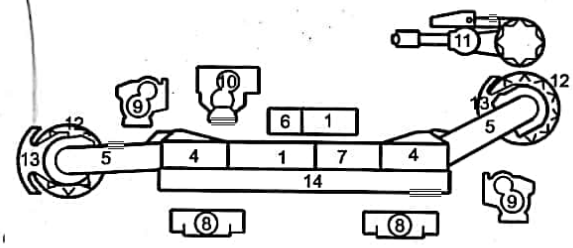


Table of Machine Units

1. Electric motor (1 or 2)
2. Hydraulic haulage pump box (chain hauled machine).
3. Haulage gearbox (chain hauled machine)
4. Gearhead gearbox (all machines)
5. Ranging arm (outboard)
6. Adaptor box (with 2 motors) or radio control box.
7. Power pack (chainless haulage machine only)
8. Underframe mounted chainless haulage traction unit.
9. End mounted chainless haulage traction unit.
10. Intermediate box for chainless haulage traction unit.
11. Lumpbreaker (optional) on chain haulage and chainless machines
12. Cutting drums
13. Powered cowl
14. Underframe– roll steering or fixed – for chain haulage and chainless machines.

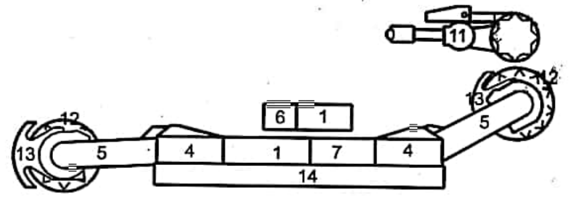


Fig. 19.16

QUESTIONS

1. What are the advantages of using a coal cutting machine in a mine? What is meant by a seven-line cutter chain? State the difference between a long-wall and short-wall coal cutting machine.

2. What are the machines used on longwall faces for cutting and loading coal simultaneously ? Describe one of them, stating its limitations.

3. Give the general specifications of a coal drill? What changes have to be made in it and the rod for large scale drilling in stone drifts ?

   What is the performance of a coal drill when drilling in coal ?

4. What arangements are possible for transport of coal cut by a continuous miner ? State the limitations for the working of a continuous miner.

5. Write short notes on : Air leg, flight loader, water infusion at a coal face.

O O O

APPENDIX – I

Electronic Automatic Contrivance-cum-speed Indicator / Recorder

(B. G. M.L.)

The whole equipment consists of 3 parts

1. Pick-up wheel with slot limit switch :

Pick-up wheel is made out of Aluminium disc with a number of holes, and is rotated in between gaps of slotted optical limit switch which transmits light from a Ga As infrared emitting diode to a silicon photo transistor. Both semiconductor chips of the slot limit switch face each other across a 2.5 mm. air gap. It senses an object in the air gap by the effect of light transmission. Whenever aluminium disc rotates infrared light cuts and photo transistor produces high and low voltage levels which will be counted and displayed in digital cage speed indicator. The slotted optical limit switch is single piece construction of the emitter and detector components, which provides excellent moisture resistance, immunity from thermal shocks, high and low temperature stability and protection from thermal shocks and vibration. Also its recessed detector provides a high signal to noise ratio in ambient light.

2. The Counter / Display Unit :

The counter consists of the following main parts. (a) Display, (b) Counter board, (c) Step down - transformer, (d) Relays.

(a) *Display :* The display is a 2 – digit 7 – segments light emitting diode (LED). It directly indicates speed for every second. (In m/s).

(b) *The counter Board :*This is the heart of the whole system. It has a capacity to count upto 2 digits and also display the count. It operates the minimum and maximum set speed relays whenever the count exceeds the predetermined values.

(c) *Step down transformer :*It is double wound isolated transformer giving 15-05-15 volts output at 1 amp. The primary voltage is 220 V., 50 Hz. AC single Ph.

(d) *Relays :*It is a 3 change over 12 V. DC relays, rated at 5 amps, at 230 V. resistive. One change-over contact is used for operation of solenoid valves/brake thrustor. One relay operates for slow easily be located where it can be most convenient to winder operator (Driver). Its