TVET CERTIFICATE V IN PRODUCTION TECHNOLOGY

MACHINE ELEMENTS

PRTME501

Describe machine elements

Competence

REQF Level: 5

Learning hours: 80

Credits: 8

Sub-sector: Mining and Manufacturing

Sub-sector: Production Technology

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Purpose statement

This module describes the skills, knowledge and attitudes required to describe machine elements. At the end of this module, participants will be able to analyze machine elements, analyze stress-strain and assess machine parts failure in order to ensure that the functionality of the machine meet the specifications and working principles

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Learning Unit 1 -Identify machine parts

LO 1.1 - Identify machine parts assembly

- Content/Topic 1: Classification of machine

Introduction

All the machines are made up of elements or parts and units. Each element is a separate part of the machine and it may have to be designed separately and in assembly. Each element in turn can be a complete part or made up of several small pieces which are joined together by riveting, welding etc. Several machine elements are assembled together which are perform various applications. A machine elements are based on skills of designer, technical and information. These elements include fastening devices, chain, belt and gear drives, bearings, oil seals and gaskets, springs, shafts, keys, couplings, and so on. The final outcome of the design process consists of the description of the machine. The description is in the form of drawings of assembly and individual components. It is created to satisfy a recognized need of customer.

1. **Machines generating mechanical energy**

What is machine and mechanical energy?

Machine:is a device or mechanical structure that uses power to apply forces and control movement to perform an intended action.

A machine is a combination of resistant bodies (links or elements) with successfully constrained relative motions, which is used for transmitting other forms of energy into mechanical energy or transmitting and modifying available energy to do some particular kind of work.

Example machine

Heat engine: Receives heat energy and transformers it into mechanical energy.

Electric motor: Changes electric energy into mechanical energy.

A pump: Input electric power and output hydraulic power.

Note: It should be noted that machine must be capable of doing useful work. the majority of machines receives mechanical energy, and modify it so that the energy can be used for doing some specific task, for which it is designed, common examples of such machines being hoist, lathe, screw jack.

2. **Machines transforming mechanical energy**

These machines are called converting machines because they convert mechanical energy into other form of energy like electricity, hydraulic energy etc.

Examples

A device which converts electrical energy into mechanical energy is called an electrical motor.

The working principle of an electric motor depends on the magnetic and electric field interaction.

There are two types of electric motor and they are:

- ✓ AC motor: Converts alternating current into mechanical power. Linear motor, synchronous motor, and induction motor are the three types of AC motor.

- ✓ DC motor: Converts direct current into mechanical power. Self-excited motor and separately excited are the two classifications of DC motor.

3. **Machines utilizing mechanical energy**

Machines utilizing mechanical energy: These machines receive mechanical energy and utilize it for various applications.

Examples of these machines are turbine that utilizes the mechanical energy to cut metals and washing machine that utilizes the rotation of the rotor for washing the clothes.

- **Content/Topic2: Machine structural components**

Machine components and structures are frequently subjected to variable amplitude loading in which significant portions of the fatigue cycles have amplitudes less than fatigue limit observed under constant amplitude loading.

1. **Frame members**

is structure that consist several members connected via the pins. They are very similar to trusses, except that some member is multi –force member. Basically it's fancy way of saying there are not two force member (trusses are) because of additional force acting on it. Here is example

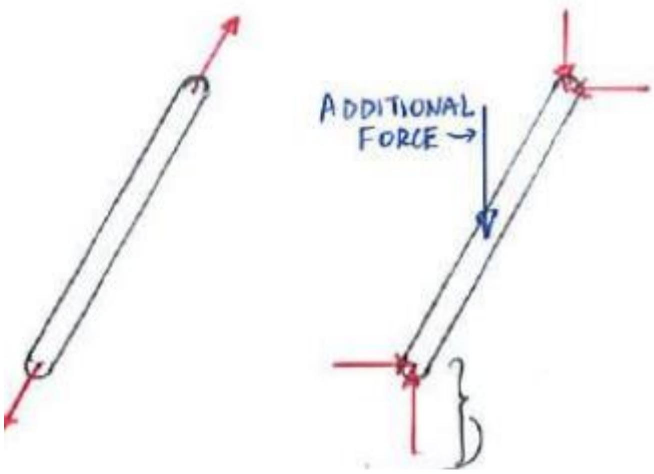


Figure 1Frame members

Machine are basically devices used to transmit forces. A good example is the plier,notice that the plier consist of two member connected by a pin in the center.

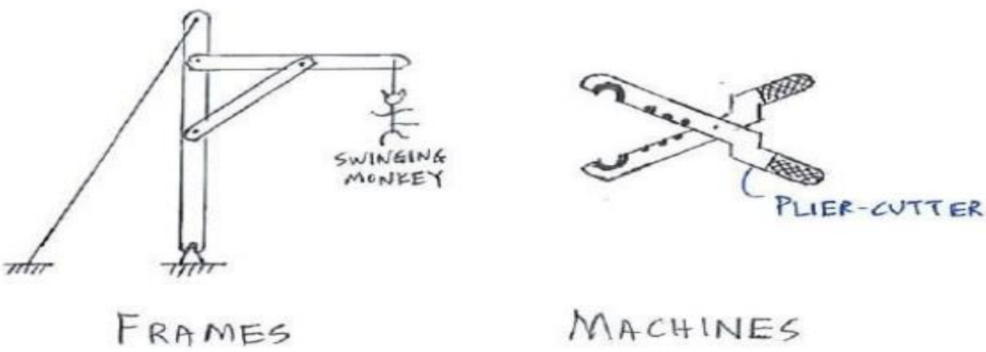


Figure 2.Frame members

In this section, we seek to analyses the force acting in frame and machine. the main strategy of tackling problems in this section is to:

- ✓ Separate the frame /machine into individual members.
- ✓ Apply the x- and y- action reaction pairs at the pin joint, and solve for equation of equilibrium.

A quick example of these steps performed on a plier is shown below

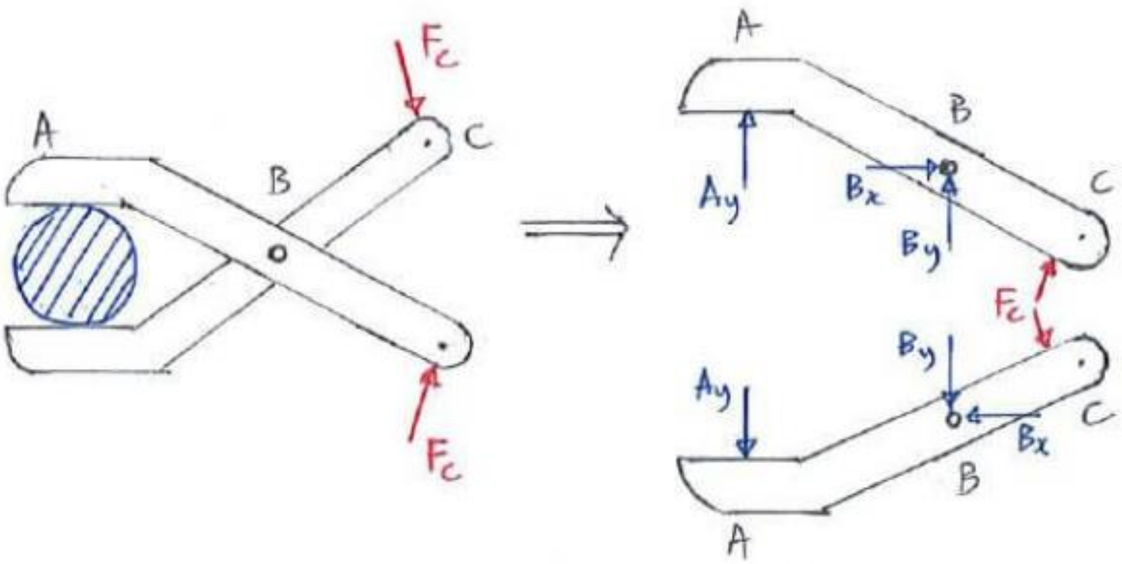


Figure 3. Analysis of Frames and Machines

The process used to analyze frames and machines, involves breaking the structure down into individual components in order to solve for the forces acting on each component. Sometimes the structure as a whole can be analyzed as a rigid body, and each component can always be analyzed as a rigid body.

The process used to analyze frames and machines is outlined below:

- ❖ In the beginning it is usually useful to label the members in your structure. This will help you keep everything organized and consistent in later analysis. In this book, we will label everything by assigning letters to each of the joints.

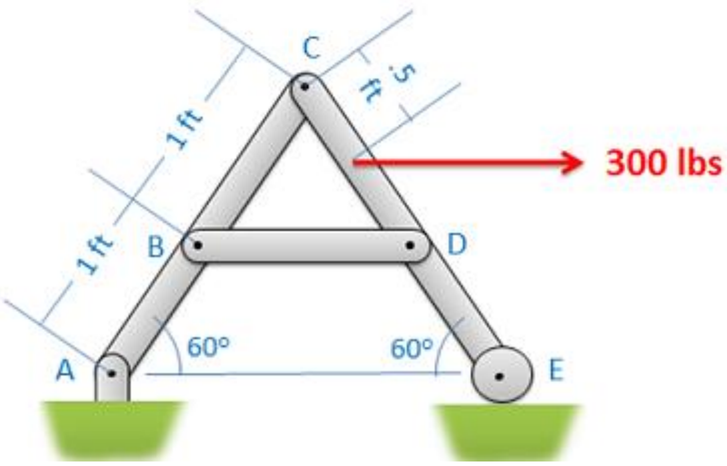


Figure 4.frame

- ❖ Next you will need to determine if we can analyze the entire structure as a rigid body. In order to do this, the structure needs to be independently rigid. This means that it would be rigid even if we separated it from its supports. If the structure is independently rigid (no machines, and only some frames will be independently rigid), then analyze the structure as a single rigid body to determine the reaction forces acting on the structure. If the structure is not independently rigid then skip this step.

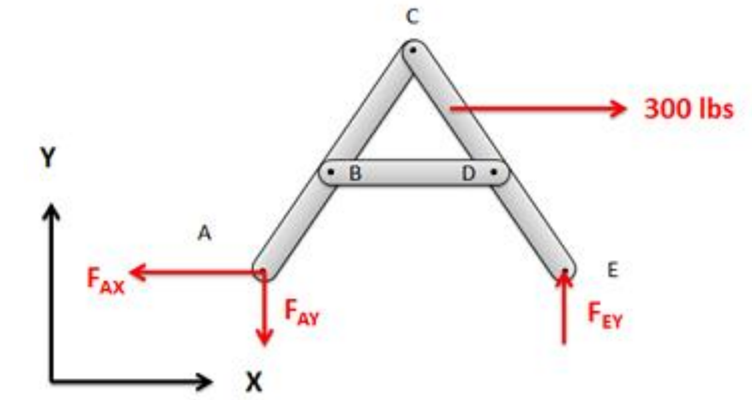


Figure 5.Frames and Machines

- Next you will draw a free body diagram for each of the components in the structure. You will need to include all forces acting on each member, including:
  - First add any external reaction or load forces that may be acting at the components.
  - Second identify any two force members in the structure. At their connection points they will cause a force with an unknown magnitude but a known direction (the forces will act along the line between the two connection points on the member).
  - Next add in the reaction forces (and possibly moments) at the connection points between non-two force members. For forces with an unknown magnitude and direction (such as in pin joints) the forces are often drawn in as having unknown x and y components (x, y and z for 3D truss problems).
  - Remember that the forces at each of the connection points will be a Newton's Third Law pair. This means that if one member exerts some force on some other member, then the second member will exert an equal and opposite force back on the first. When we draw out our unknown forces at the connection points, we must make sure that the forces acting on each member are opposite in direction.

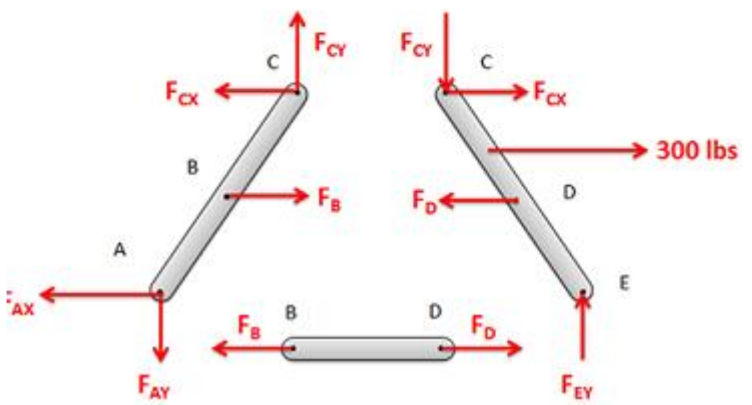


Figure 6.frame analyses

Write out the equilibrium equations for each component you drew a free body diagram of. These will be extended bodies, so you will need to write out the force and the moment equations.

For 2D problems you will have three possible equations for each component, two force equations and one moment equation.

∑→F=0∑F→=0 ∑→M=0∑M→=0

∑Fx=0∑Fx=0 ∑Fy=0∑Fy=0 ∑Mz=0∑Mz=0

For 3D problems you will have six possible equations for each component, three force equations and three moment equations.

∑→F=0∑F→=0

∑Fx=0∑Fx=0 ∑Fy=0∑Fy=0 ∑Fz=0∑Fz=0

∑→M=0∑M→=0

∑Mx=0∑Mx=0 ∑My=0∑My=0 ∑Mz=0

Example question one

Find all the forces acting on each of the members in the structure below.

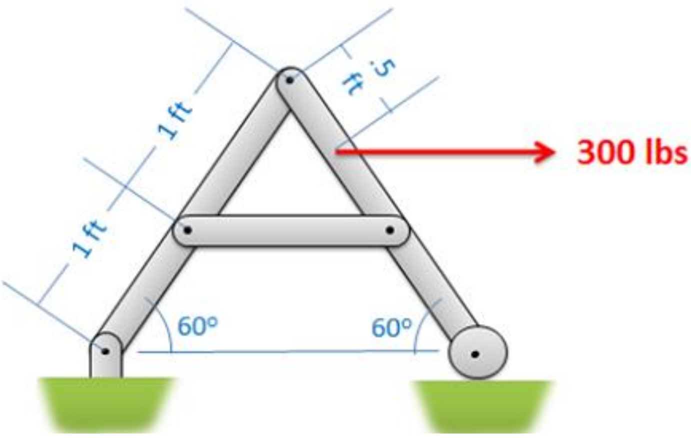
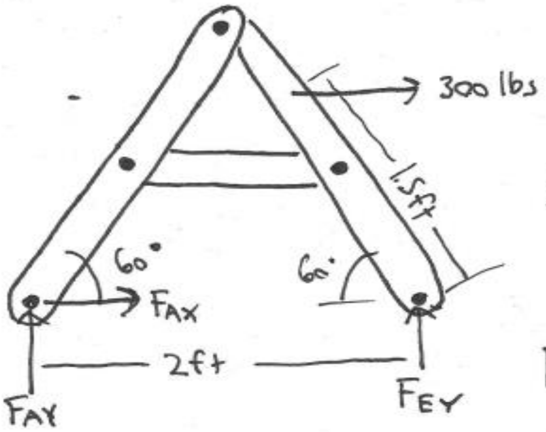


Figure 7.frame

Solution



$$\sum F_x = F_{AX} + 300 = 0$$

$$\sum F_y = F_{AY} + F_{EY} = 0$$

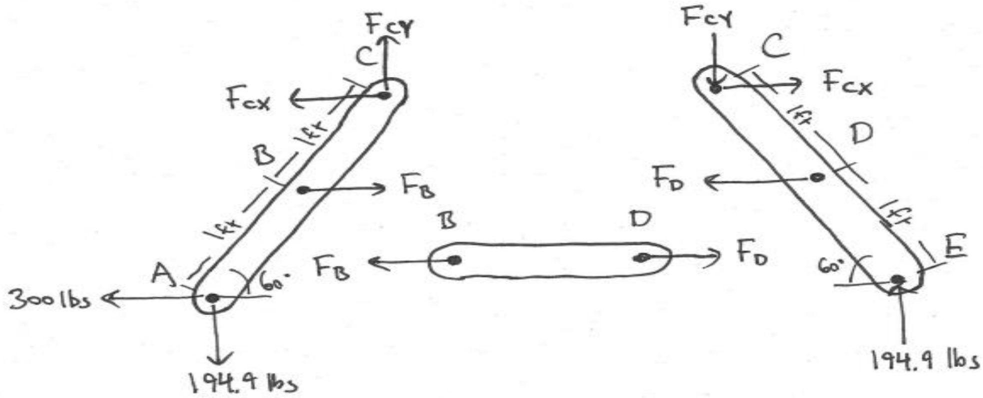
$$\sum M_A = (2)(F_{EY}) - (1.5 \sin(60))(300) = 0$$

$$F_{AX} = -300 \text{ lbs}$$

$$F_{EY} = \frac{(1.5 \sin(60))(300)}{2} = 194.9 \text{ lbs}$$

$$F_{AY} = -194.9 \text{ lbs}$$

Now break the components apart.



Member ABC

$$\sum F_x = -F_{CX} + F_B - 300 = 0$$

$$\sum F_y = F_{CY} - 194.9 = 0$$

$$\sum M_C = (1 \sin(60))(F_B) + (2 \cos(60))(194.9) - (2 \sin(60))(300) = 0$$

$$F_{CY} = 194.9 \text{ lbs}$$

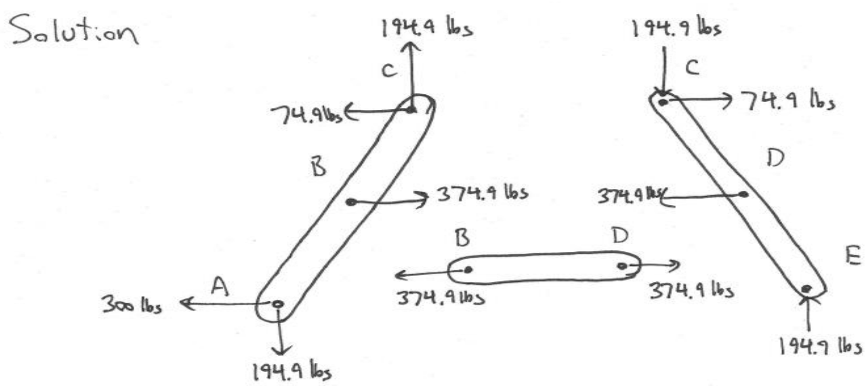
$$F_B = \frac{(-2 \cos(60))(194.9) + (2 \sin(60))(300)}{\sin(60)} = 374.9 \text{ lbs}$$

$$F_{CX} = 374.9 - 300 = 74.9 \text{ lbs}$$

Member BD

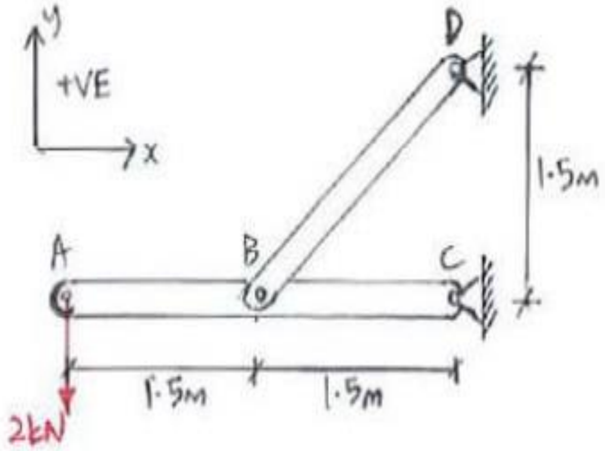
$$\sum F_x = -F_B + F_D = 0$$

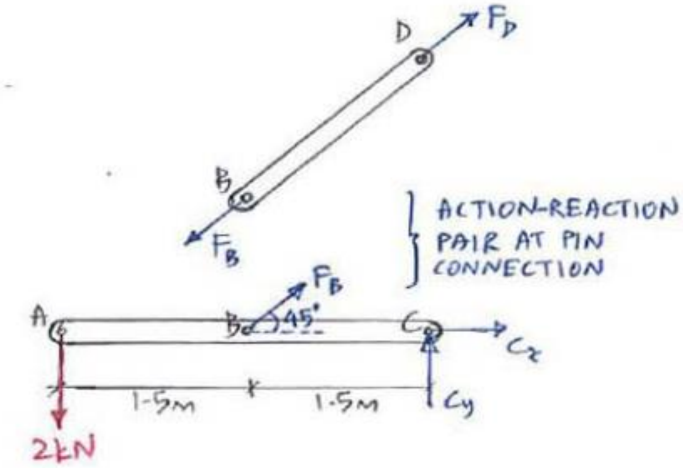
$$F_D = 374.9 \text{ lbs}$$



Examples question two

Determine the force acting the pin A, B and D

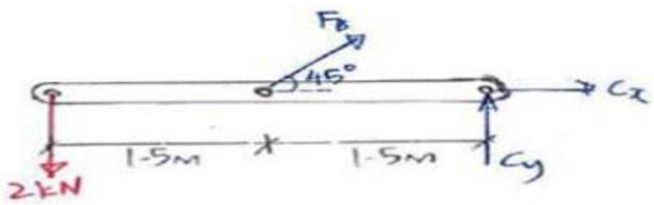




Equitation of equilibrium

$$[\overset{+}{\rightarrow}\Sigma F_x = 0] \qquad [+\uparrow \Sigma F_y = 0] \qquad [\Sigma M = 0]$$

Member AC



$$[+\circlearrowleft \Sigma M_C = 0] \quad (2\,kN \times 3\,m) - (F_B \sin 45^\circ \times 1.5\,m) = 0$$

$$F_B = \underline{5.657\,kN}$$

$$[+\uparrow \Sigma F_y = 0] \quad -2\,kN + \overbrace{5.657}^{F_B} \sin 45^\circ\,kN + C_y = 0$$

$$C_y = \underline{-2\,kN\,(\downarrow)}$$

$$[\rightarrow \Sigma F_x = 0] \quad \overbrace{5.657}^{F_B} \cos 45^\circ\,kN + C_x = 0$$

$$C_x = \underline{-4\,kN\,(\leftarrow)}$$

2. Shafts

A shaft is a rotating machine element, usually circular in cross section, which is used to transmit power from one part to another, or from a machine which produces power to a machine which absorbs power. The various members such as pulleys and gears are mounted on it.

The shafts are usually cylindrical, but may be square or cross-shaped in section. They are solid in cross-section but sometimes hollow shafts are also used

Types of Shafts The following two types of shafts are important from the subject point of view:

- **Transmission shafts**. These shafts transmit power between the source and the machines absorbing power. The counter shafts, line shafts, overhead shafts and all factory shafts are transmission shafts. Since these shafts carry machine parts such as pulleys, gears etc., therefore they are subjected to bending in addition to twisting.
- **Machine shafts**. These shafts form an integral part of the machine itself. The crank shaft is an example of machine shaft

Material Used for Shafts The material used for shafts should have the following properties:

- ✓ It should have high strength.
- ✓ It should have good machinability.
- ✓ It should have low notch sensitivity factor.
- ✓ It should have good heat treatment properties.
- ✓ It should have high wear resistant properties.

Materials

The material used for ordinary shafts is mild steel. When high strength is required, an alloy steel such as nickel, nickel-chromium or chromium-vanadium steel is used.

Standard Sizes of Transmission Shafts The standard sizes of transmission shafts are:

25 mm to 60 mm with 5 mm steps; 60 mm to 110 mm with 10 mm steps; 110 mm to 140 mm with 15 mm steps; and 140 mm to 500 mm with 20 mm steps. The standard length of the shafts are 5 m, 6 m and 7 m.

Design of Shafts

The shafts may be designed on the basis of

1. Strength, and

2. Rigidity and stiffness.

In designing shafts on the basis of strength, the following cases may be considered:

i. Shafts subjected to twisting moment or torque only,
ii. Shafts subjected to bending moment only,
iii. Shafts subjected to combined twisting and bending moments, and
iv. Shafts subjected to axial loads in addition to combined torsional and bending loads.

Stresses in Shafts

The following stresses are induced in the shafts:

a) Shear stresses due to the transmission of torque (i.e. due to torsional load).
b) Bending stresses (tensile or compressive) due to the forces acting upon machine elements like gears, pulleys etc. as well as due to the weight of the shaft itself.
c) Stresses due to combined torsional and bending loads

3. Bearings

Bearing is a mechanical element that permits motion between two parts, such as the shaft and the housing, with minimum friction.

The functions of the bearing are as the follows:

- ✓ The bearing ensures free rotation of the shaft or the axle with minimum friction.
- ✓ The bearing supports the shaft or the axle and holds it in the correct position.
- ✓ The bearing takes up the forces that act on the shaft or the axle and transmits tem to the frame or the foundation.

Classification of Bearings

- **Depending upon the direction of load to be supported.** The bearings under this group are classified as: (a) Radial bearings, and (b) Thrust bearings. In radial bearings, the load acts perpendicular to the direction of motion of the moving element.as shown in Fig. (a) and (b). In thrust bearings, the load acts along the axis of rotation as shown in Fig (c)

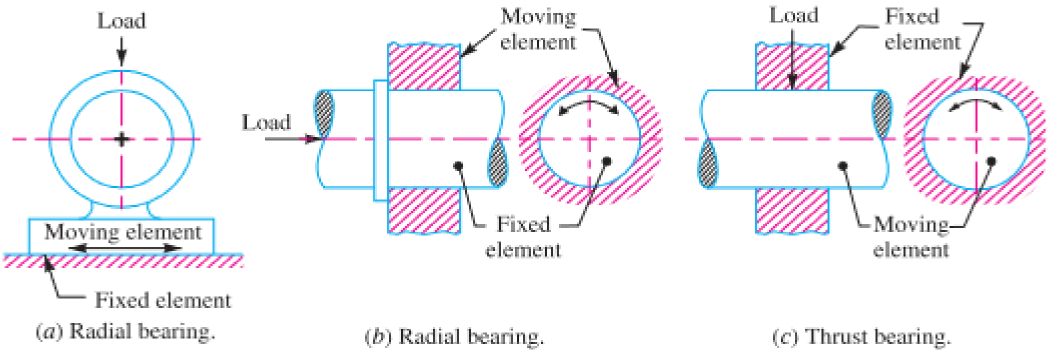


Figure 8.Radial and thrust bearings

➢ **Depending upon the nature of contact**. The bearings under this group are classified as: (a) **Sliding contact bearings, and (b) Rolling contact bearings**. In sliding contact bearings, as shown in Fig. the sliding takes place along the surfaces of contact between the moving element and the fixed element. The sliding contact bearings are also known as plain bearings.

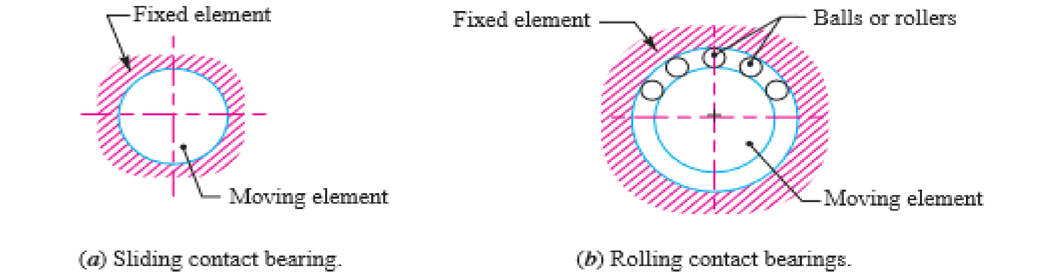


Figure 9.Sliding and rolling contact bearings

4. Axle

An axle is a central shaft for a rotating wheel or gear. On wheeled vehicles, the axle may be fixed to the wheels, rotating with them, or fixed to the vehicle, with the wheels rotating around the axle.[1] In the former case, bearings or bushings are provided at the mounting points where the axle is supported. In the latter case, a bearing or bushing sits inside a central hole in the wheel to allow the wheel or gear to rotate around the axle. Sometimes, especially on bicycles, the latter type axle is referred to as a *spindle* Drive axle

Figure 10.Splines on a front drive axle.

An axle that is driven by the engine or prime mover is called a *drive axle*.

Modern front-wheel drive cars typically combine the transmission (gearbox and differential) and front axle into a single unit called a *transaxle*. The drive axle is a split axle with a differential and universal joints between the two half axles. Each half axle connects to the wheel by use of a constant velocity (CV) joint which allows the wheel assembly to move freely vertically as well as to pivot when making turns.

In rear-wheel drive cars and trucks, the engine turns a driveshaft (also called a *propeller shaft* or *tail shaft*) which transmits rotational force to a drive axle at the rear of the vehicle. The drive axle may be a live axle, but modern rear wheel drive automobiles generally use a split axle with a differential. In this case, one half-axle or half-shaft connects the differential with the left rear wheel, a second half-shaft does the same with the right rear wheel; thus the two half-axles and the differential constitute the rear axle.

Some simple vehicle designs, such as leisure go-karts, may have a single driven wheel where the drive axle is a split axle with only one of the two shafts driven by the engine, or else have both wheels connected to one shaft without a differential (kart racing). However, other go-karts have two rear drive wheels too.

Dead axle (lazy axle)

A dead axle, also called a *lazy axle*, is not part of the drivetrain, but is instead free-rotating. The rear axle of a front-wheel drive car is usually a dead axle. Many trucks and trailers use dead axles for strictly load-bearing purposes. A dead axle located immediately in front of a drive axle is called a *pusher axle*. A tag axle is a dead axle situated behind a drive axle. Dead axles are also found on semi-trailers, farm equipment, and certain heavy construction machinery serving the same function. On some vehicles (such as motorcoaches), the tag axle may be steerable. In some designs the wheels on a lazy axle only come into contact with ground when the load is significant, thus saving unnecessary tire wear.

IN Generally a STATIONARY member used as a support for rotating members such as bearings, wheels, idler gears, etc.

5. **Keys**

Key (engineering)

In mechanical engineering, a key is a machine element used to connect a rotating machine element to a shaft. The key prevents relative rotation between the two parts and may enable torque transmission. For a key to function, the shaft and rotating machine element must have a keyway and a keyseat, which is a slot and pocket in which the key fits. The whole system is called a keyed joint. A keyed joint may allow relative axial movement between the parts.

There are five main types of keys: *sunk*, saddle, tangent, round, and spline.

- **Sunk key**

Types of sunk keys: rectangular, square, parallel sunk, gib-head, feather, and Woodruff.

- **Parallel key**

Parallel keys are the most widely used. They have a square or rectangular cross-section. Square keys are used for smaller shafts and rectangular faced keys are used for shaft diameters over 6.5 in (170 mm) or when the wall thickness of the mating hub is an issue. Set screws often accompany parallel keys to lock the mating parts into place. The keyway is a longitudinal slot in both the shaft and mating part.

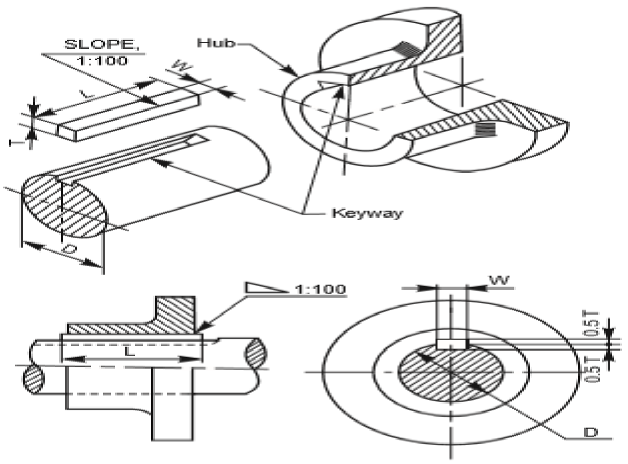


Figure 11.Parallel key

Cross-section of a parallel keyed joint

$W = d/4$

$H = 2d/3$

where

W is the key width

H is the key height

d is the shaft diameter

- **Woodruff keys**

It is a sunk key, in the form of a segment of a circular disc of uniform thickness (Fig. a). As the bottom surface of the key is circular, the keyway in the shaft is in the form of a circular recess to the same curvature as the key. A keyway is made in the hub of the mounting, in the usual manner. Woodruff key is mainly used on tapered shafts of machine tools and automobiles. Once placed in position, the key tilts and aligns itself on the tapered shaft (Fig.b). The following are the proportions of woodruff keys:

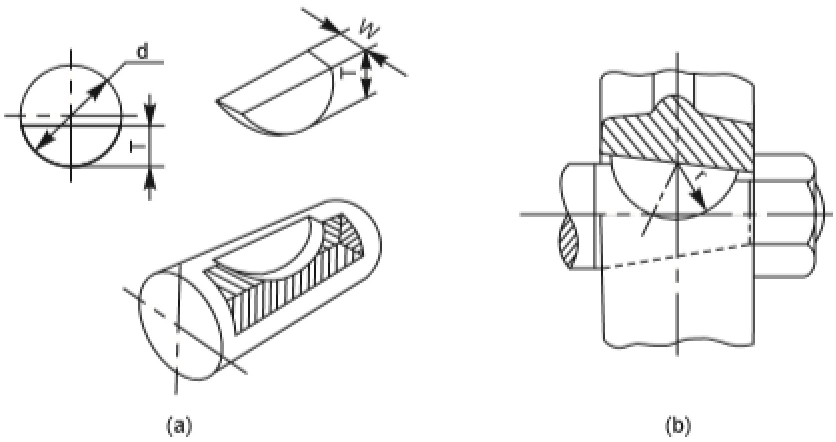


Figure 12.Woodruff keys

If D is the diameter of the shaft,

Thickness of key, W = 0.25 D

Diameter of key, d = 3 W

Height of key, T = 1.35 W

Depth of the keyway in the hub, $T1$ = 0.5 W + 0.1 mm

Depth of keyway in shaft, $T2$ = 0.85 W

The advantages of a Woodruff key

- It accommodates itself to any taper in the hub or boss of the mating pieces.
- It is useful on tapering shaft ends. Its extra depth in the shaft prevents any tendency to turn over in its keyway.

The disadvantages of Woodruff key

- The depth of the keyway weakens the shaft.
- It cannot be used as a feather.

- **Round Keys**

Round keys are of circular cross-section, usually tapered (1:50) along the length. A round key fits in the hole drilled partly in the shaft and partly in the hub (Figure below). The mean diameter of the pin may be taken as 0.25 D, where D is shaft diameter. Round keys are generally used for light duty, where the loads are not considerable.

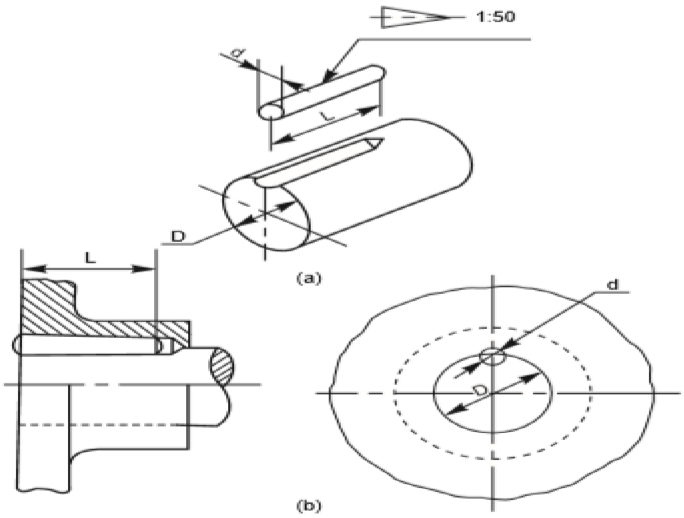


Figure 13.Round Keys

Saddle keys

These types of keys are generally attached to the driving member (e.g. shafts). These types of keys have less strength as compared with the sunk keys. These are rarely used keys, to transmit lower power to the driven members (e.g. couplings)

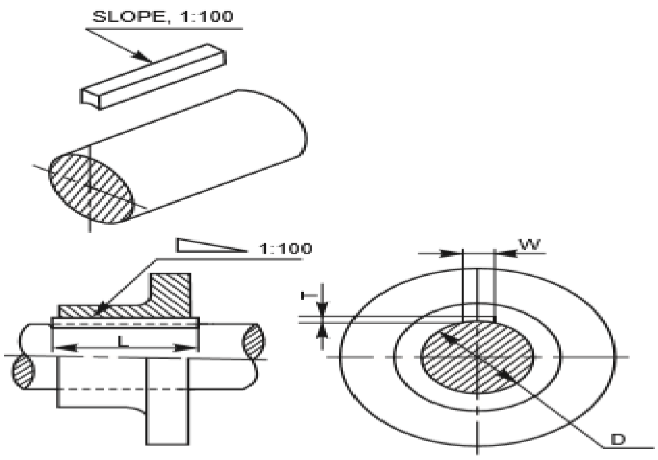


Figure 14.Saddle keys

- **Tangent Keys**

Tangent keys are used in high-torque heavy-duty applications. The keyway is similar to a parallel key, except it extends tangentially out of the external shaft into the internal shaft. What would have been the side of each keyway forms heels against which the key sits, and transfers force compressively. This latter point means that for reversible motion of the shaft, another key along a tangent outwards in the opposing direction is needed. Typically this will be offset by 90° or 180° on the shaft. The key may be wedge, rectangular, or square shaped, but particularly rectangular double-taper keys are used.

6. **Spline key**

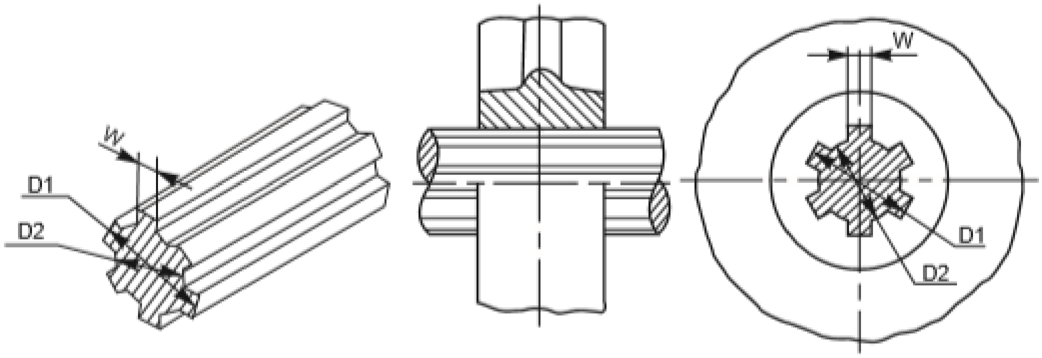


Figure 15.Spline

This type of key uses multiple keyways in the hub to transmit high power.
Splines are ridges or teeth on a drive shaft that mesh with grooves in a mating piece and transfer torque to it, maintaining the angular correspondence between them. For instance, a gear mounted on a shaft might use a male spline on the shaft that matches the female spline on the gear. The splines on the pictured drive shaft match with the female splines in the center of the clutch plate, while the smooth tip of the axle is supported in the pilot bearing in the flywheel. An alternative to spline is a keyway and key, thought splines provide a longer fatigue life.

7. **Cotter pin**

A cotter is a flat wedge shaped piece, made of steel. It is uniform in thickness but tapering in width, generally on one side; the usual taper being 1:30. The lateral (bearing) edges of the cotter and the bearing slots are generally made semi-circular instead of straight

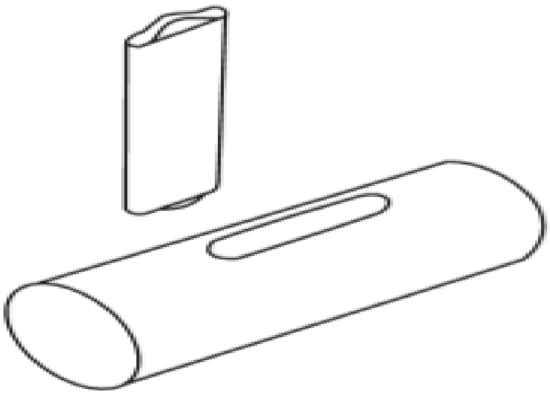


Figure 16.Cotter pin

This increases the bearing area and permits drilling while making the slots. The cotter is locked in position by means of a screw as shown in Fig.17 Cotter joints are used to connect two rods, subjected to tensile or compressive forces along their axes. These joints are not suitable where the members are under rotation.

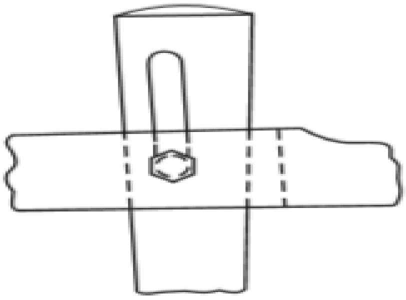


Figure 17.Cotter joints

The commonly used cotter joints are following

i. Cotter Joint with Sleeve

This is the simplest of all cotter joints, used for fastening two circular rods. To make the joint, the rods are enlarged at their ends and slots are cut. After keeping the rods butt against each other, a sleeve with slots is placed over them. After aligning the slots properly, two cotters are driven-in through the slots, resulting in the joint (Figure below). The rod ends are enlarged to take care of the weakening effect caused by the slots.

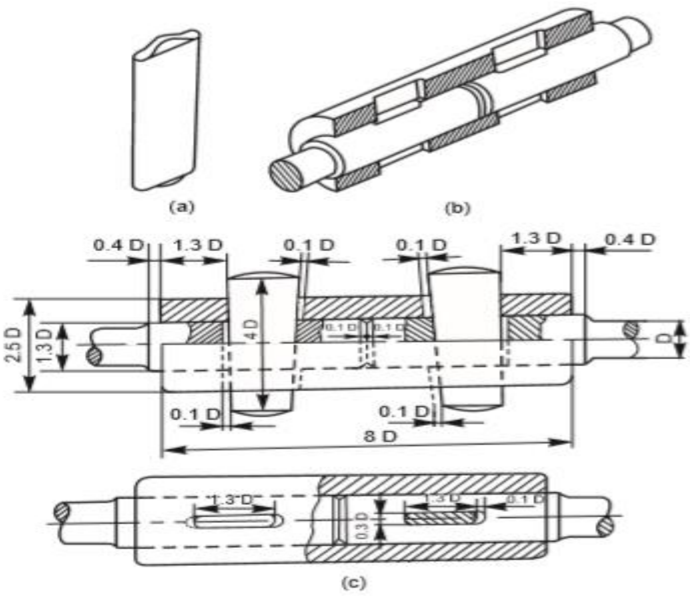


Figure 18.Cotter Joint with Sleeve

The slots in the rods and sleeve are made slightly wider than the width of cotter. The relative positions of the slots are such, that when a cotter is driven into its position, it permits wedging action and pulls the rod into the sleeve.

ii. Cotter Joint with Socket and Spigot Ends

This joint is also used to fasten two circular rods. In this, the rod ends are modified instead of using a sleeve. One end of the rod is formed into a socket and the other into a spigot (Figure below) and slots are cut. After aligning the socket and spigot ends, a cotter is driven-in through the slots, forming the joint.

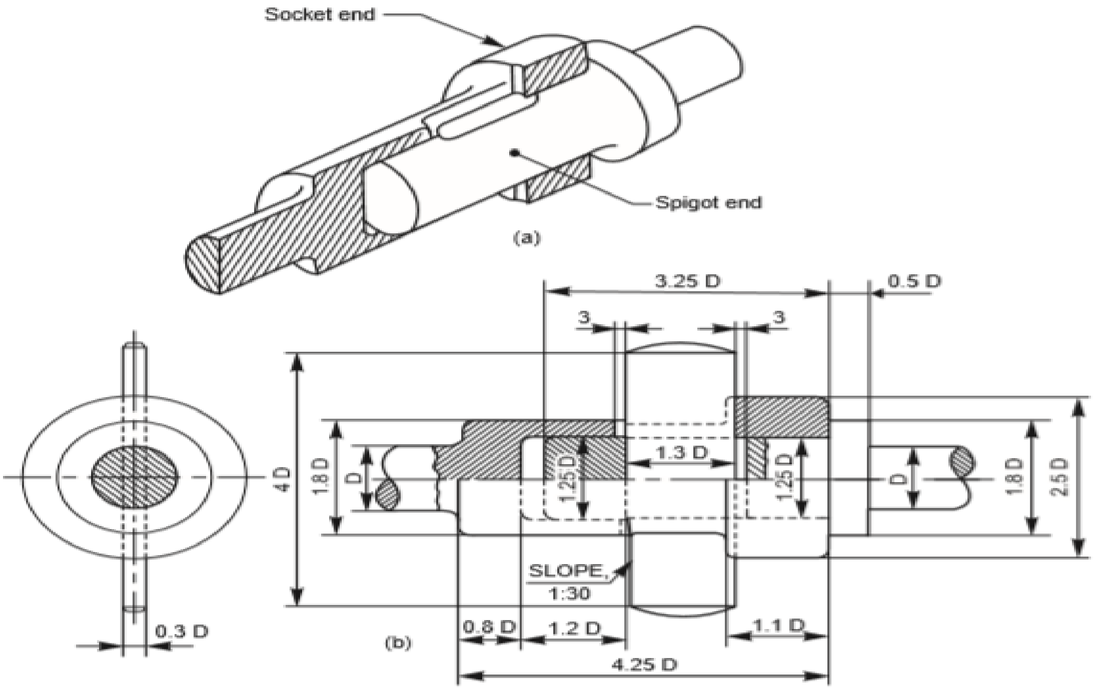


Figure 19.Cotter joint with socket and spigot ends

iii. Cotter Joint with a Gib

This joint is generally used to connect two rods of square or rectangular cross-section. To make the joint, one end of the rod is formed into a U-fork, into which, the end of the other rod fits in. When a cotter is driven-in, the friction between the cotter and straps of the U-fork, causes the straps to open. This is prevented by the use of a gib. A gib is also a wedge shaped piece of rectangular cross-section with two rectangular projections called lugs. One side of the gib is tapered and the other straight. The tapered side of the gib bears against the tapered side of the cotter such that, the outer edges of the cotter and gib as a unit are parallel. This facilitates making of slots with parallel edges, unlike the tapered edges in case of ordinary cotter joint. Further, the lugs bearing against the outer surfaces of the fork, prevents the opening tendency of the straps.

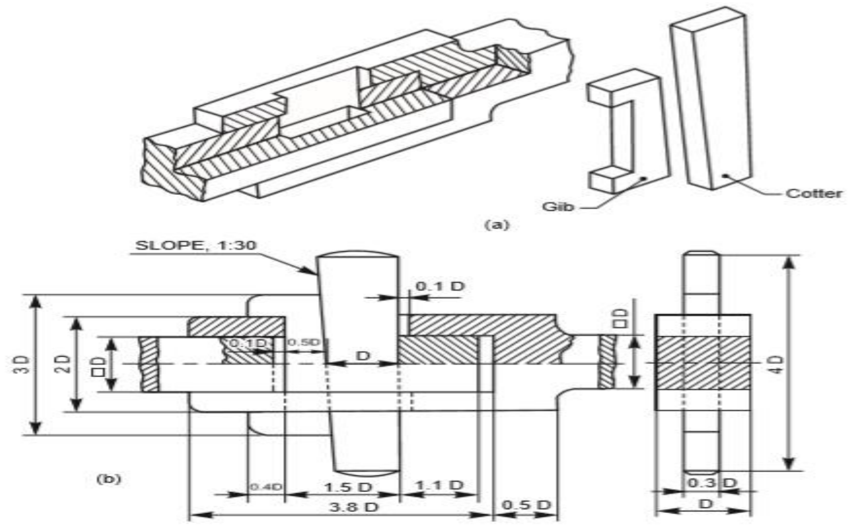


Figure 20.Cotter Joint with a Gib

iv. pin joint

In a pin joint, a pin is used to fasten two rods that are under the action of a tensile force; although the rods may support a compressive force if the joint is guided. Some pin joints such as universal joints, use two pins and are used to transmit power from one rotating shaft to another. A pin joint permits a small amount of flexibility or one rod may be positioned at an angle (in the plane containing the rods) with respect to the other rod, after providing suitable guides. Unlike in cotter joints, the pin in a pin joint is not driven-in with a force fit, but is inserted in the holes with a clearance fit. The pin is held in position, by means of a taper pin or a split pin provided at its end.

v. Knuckle Joint

A knuckle joint is a pin joint used to fasten two circular rods. In this joint, one end of the rod is formed into an eye and the other into a fork (double eye). For making the joint, the eye end of the rod is aligned into the fork end of the other and then the pin is inserted through the holes and held in position by means of a collar and a taper pin (Figure below). Once the joint is made, the rods are free to swivel about the cylindrical pin. Knuckle joints are used in suspension links, air brake arrangement of locomotives.

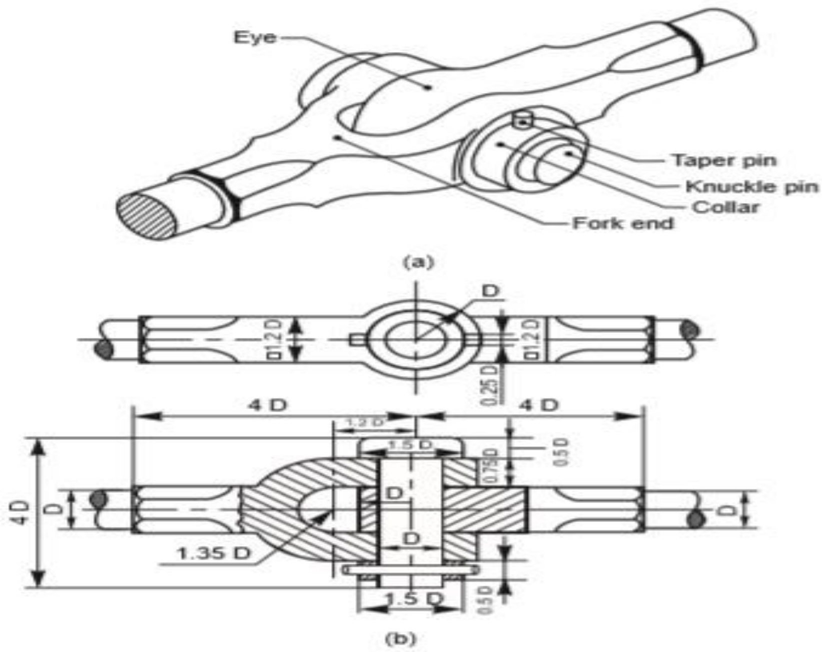


Figure 21. Knuckle Joint

8. FASTRENERS

Definitions: A machine element used for holding or joining two or more parts of a machine or structure is known as fastener. The process of joining the parts is called Fastening .

The fasteners are of two types:

Permanent and removable(Temporary).

Riveting and welding processes are used for fastening permantly. Screwed fasteners such as bolts, studs and nuts in combination, machine screws, set screws, etc., and keys, cotters, couplings, etc., are used for fastening components that require frequent assembly and dissembly. Screwed fasteners occupy the most prominent place among the removable fasteners.

In general, screwed fasteners are used :

(i) to hold parts together,

(ii) (ii) to adjust parts with reference to each other and

(iii) (iii) to transmit power

❖ **screw thread nomenclature**

A screw thread is obtained by cutting a continuous helical groove on a cylindrical surface (external thread). The threaded portion engages with a corresponding threaded hole (internal thread); forming a screwed fastener.

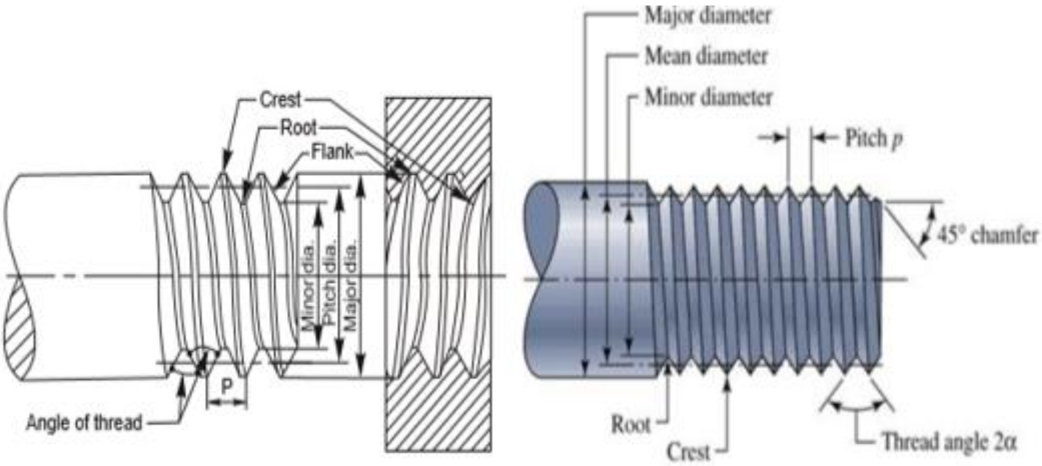


Figure 22.screw thread nomenclature

1. **Major (nominal) diameter**: This is the largest diameter of a screw thread, touching the crests on an external thread or the roots of an internal thread.
2. **Minor (core) diameter** This is the smallest diameter of a screw thread, touching the roots or core of an external thread (root or core diameter) or the crests of an internal thread.
3. **Pitch diameter** This is the diameter of an imaginary cylinder, passing through the threads at the points where the thread width is equal to the space between the threads.
4. **Pitch** It is the distance measured parallel to the axis, between corresponding points on adjacent screw threads.
5. **Lead** It is the distance a screw advances axially in one turn.
6. **Flank Flank** is the straight portion of the surface, on either side of the screw thread.
7. **Crest** It is the peak edge of a screw thread, that connects the adjacent flanks at the top.
8. **Root** It is the bottom edge of the thread that connects the adjacent flanks at the bottom.
9. **Thread angle** This is the angle included between the flanks of the thread, measured in an axial plane.

❖ **form of thread**

Bureau of Indian Standards (BIS) adapts ISO (International Organisation for Standards) metric threads which are adapted by a number of countries apart from India. The design profiles of external and internal threads are shown in Fig. The following are the relations between the various parameters marked in the figure:

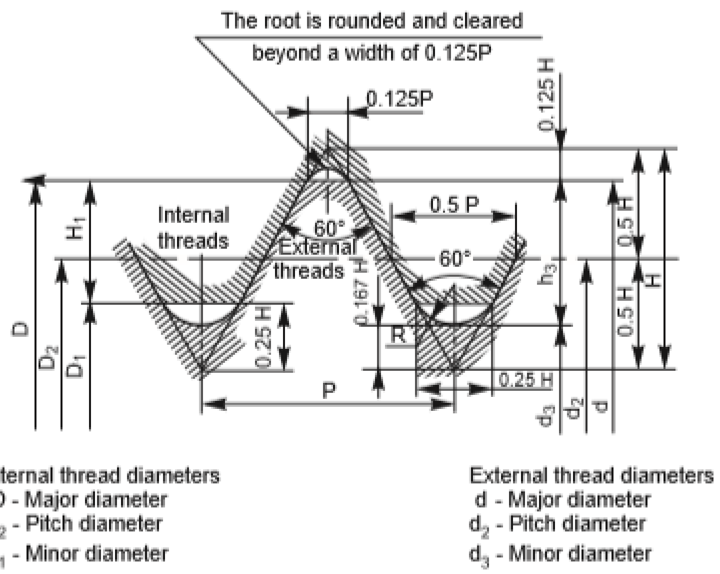


Fig. 5.2 Metric screw thread

P = Pitch

$H = 0.86\,P$

$D = d$ = Major diameter

$D_2 = d_2 = d - 0.75H$

$D_1 = d_2 - 2(H/2 - H/4) = d - 2H_1$

$d_3 = d_2 - 2\,(H/2 - H/6)$

$\quad = d - 1.22P$

$H_1 = (D - D_1)/2 = 5H/8 = 0.54P$

$h_3 = (d - d_3)/2 = 17/24H = 0.61P$

$R = H/6 = 0.14P$

Figure 23.form of thread

It may be noted from the figure that in order to avoid sharp corners, the basic profile is rounded at the root (minor diameter) of the design profile of an external thread. Similarly, in the case of internal thread, rounding is done at the root (major diameter) of the design profile.

other thread profile

- V-threads(sharp)
- British standard with worth(B.S.W) thread
- Buttress thread
- Square thread
- ACME thread
- Worm thread

❖ **RIGHT HAND AND LEFT HAND THREAD**

Screw threads may be right hand or left hand, depending on the direction of the helix. A right hand thread is one which advances into the nut, when turned in a clockwise direction and a left hand thread is one which advances into the nut when turned in a counter clockwise direction. An abbreviation LH is used to indicate a left hand thread. Unless otherwise stated, a thread should be considered as a right hand one. Figure below illustrates both right and left hand thread forms.

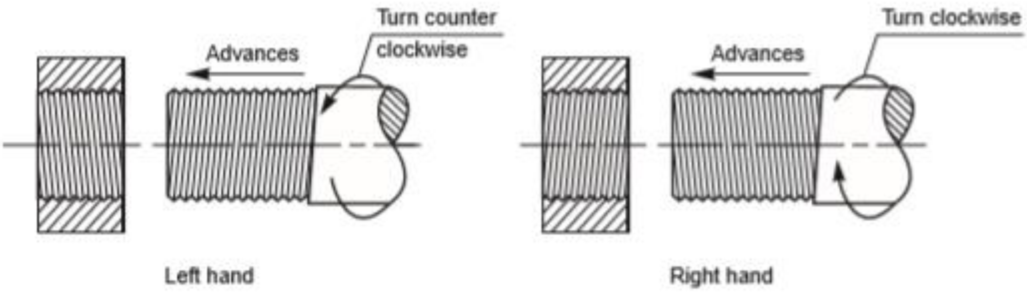


Figure 24.Right hand and left hand threads

bolt joint

A bolt and nut in combination (Fig. is a fastening device used to hold two parts together. The body of the bolt, called shank is cylindrical in form, the head; square or hexagonal in shape, is formed by forging. Screw threads are cut on the other end of the shank. Nuts in general are square or hexagonal in shape. The nuts with internal threads engage with the corresponding size of the external threads of the bolt. However, there are other forms of nuts used to suit specific requirements.

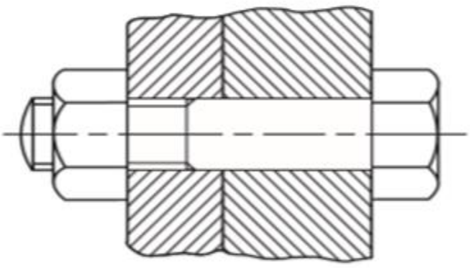


Figure 25.Bolted joint

However, square nuts are used when frequent loosening and tightening is required, for example on job holding devices like vices, tool posts in machines, etc. The sharp corners on the head of bolts and nuts are removed by chamfering.

Types of bolts

- **Hexagonal headed bolt**

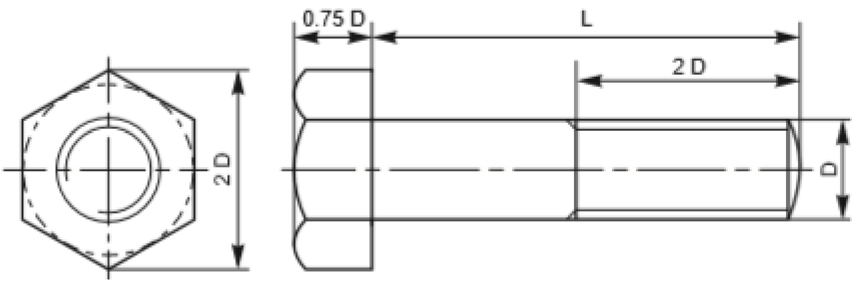


Figure 26. Hexagonal headed bolt

- **Square headed bolt**

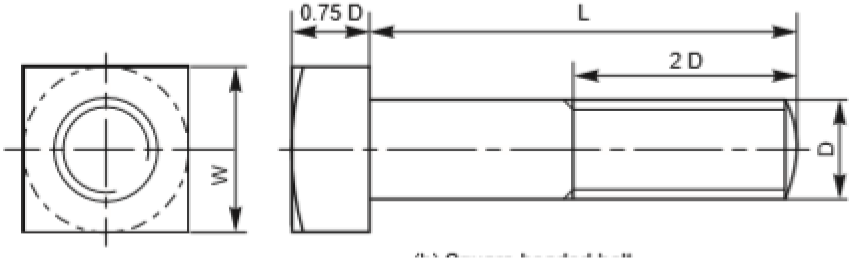


Figure 27. Square headed bolt

- **Square headed bolt with square Neck**

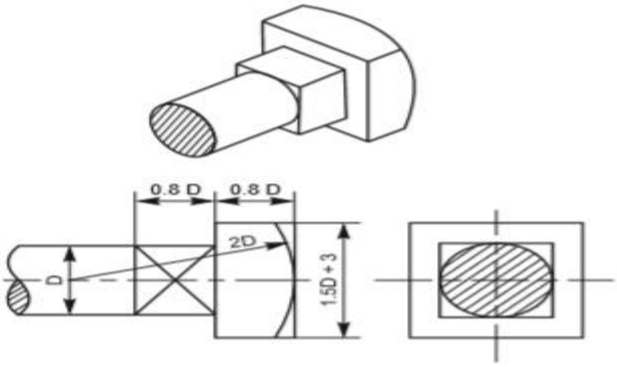


Figure 28.Square headed bolt with square Neck

It is provided with a square neck, which fits into a corresponding square hole in the adjacent part, preventing the rotation of the bolt.

- **T-Headed Bolt with Square Neck**

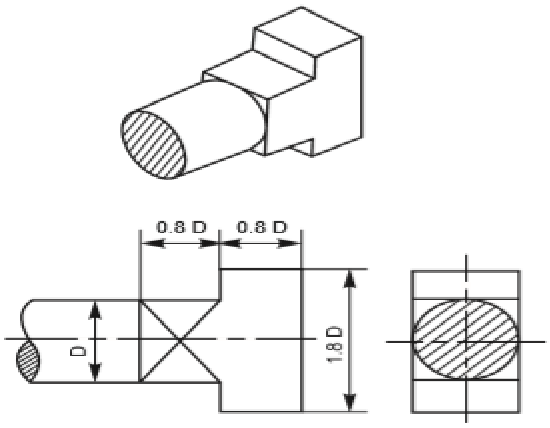


Figure 29. T-Headed Bolt with Square Neck

In this, a square neck provided below the head, prevents the rotation of the bolt. This type of bolt is used for fixing vices, work pieces, etc., to the machine table having T-slots.

- **Hook Bolt**

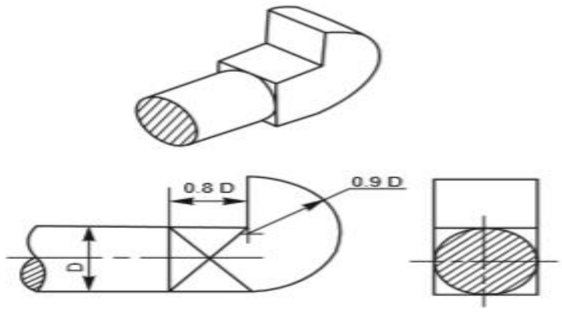


Figure 30.Hook Bolt

This bolt passes through a hole in one part only, while the other part is gripped by the hook shaped bolt head. It is used where there is no space for making a bolt hole in one of the parts. The square neck prevents the rotation of the bolt.

- **Eye Bolt**

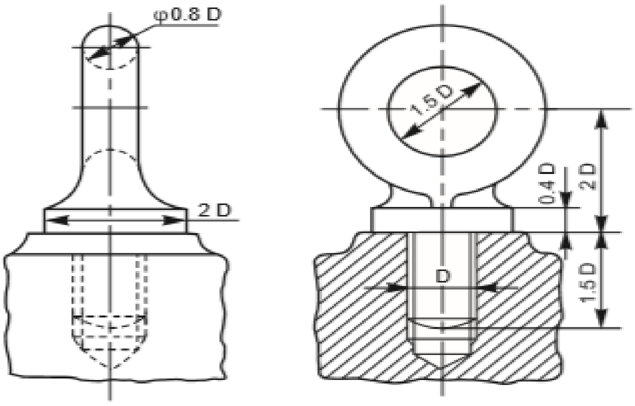


Figure 31.Eye Bolt

In order to facilitate lifting of heavy machinery, like electric generators, motors, turbines, etc., eye bolts are screwed on to their top surfaces. For fitting an eye bolt, a tapped hole is provided, above the centre of gravity of the machine.

- **Stud Bolt or Stud**

It consists of cylindrical shank with threads cut on both the ends (Fig. a). It is used where there is no place for accommodating the bolt head or when one of the parts to be joined is too thick to use an ordinary bolt. The stud is first screwed into one of the two parts to be joined, usually the thicker one. A stud driver, in the form of a thick hexagonal nut with a blind threaded hole is used for the purpose. After placing the second part over the stud, a nut is screwed-on over the nut end. It is usual to provide in the second part, a hole which is slightly larger than the stud nominal diameter. Figure. b shows a stud joint.

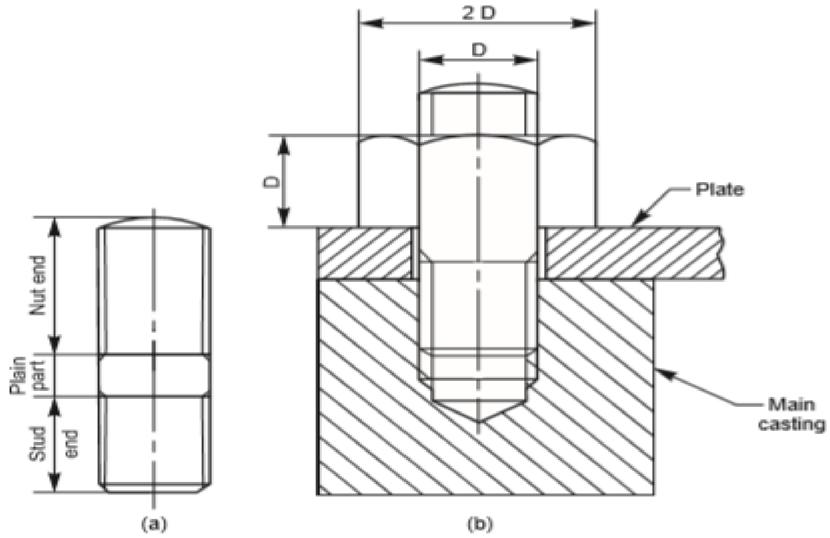


Figure 32. Stud Bolt or Stud

- **Cap Screws and Machine Screws**

Cap screws and machine screws are similar in shape, differing only in their relative sizes. Machine screws are usually smaller in size, compared to cap screws. These are used for fastening two parts, one earancehole and the other with tapped hole.

- **Set Screws**

These are used to prevent relative motion between two rotating parts, such as the movement of pulley on shaft. For this, a set screw is screwed into the pulley hub so that its end-point bears firmly against the shaft (Figure below). The fastening action is by friction between the screw and the shaft.

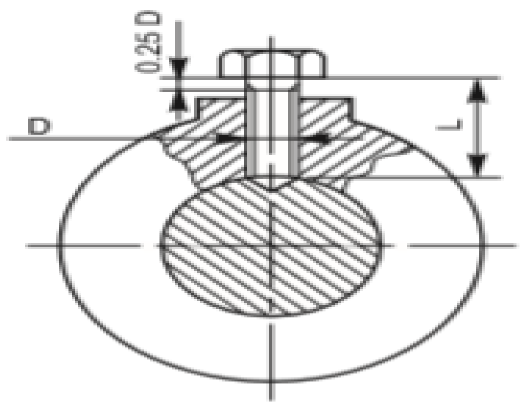


Figure 33

Set screws are not efficient and so are used only for transmitting very light loads. For longer life, set screws are made of steel and case hardened. Further, for better results, the shaft surface is suitably machined for providing more grip, eliminating any slipping tendency. Figure below shows different forms of set screws.

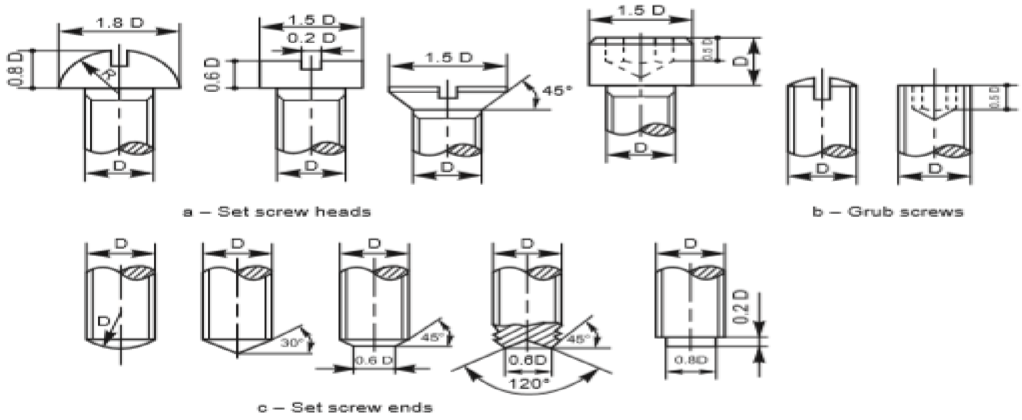


Figure 34.Set screws

- **Hexagonal Nut**

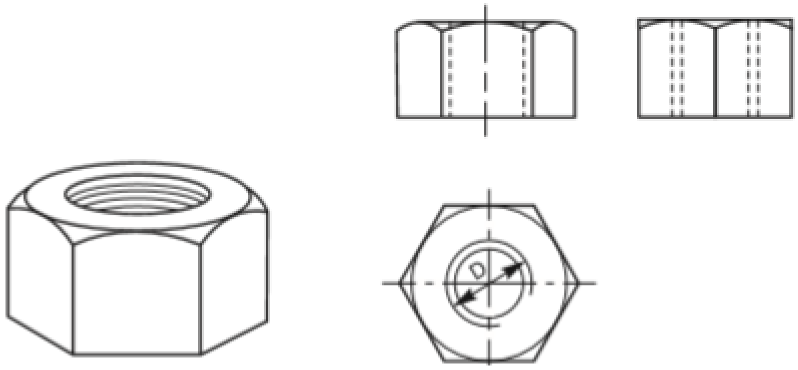


Figure 35. Hexagonal Nut

Figure 31: Hexagonal Nut

Empirical relations:

Major or nominal diameter of bolt = D

Thickness of nut, T = D

Width of nut across flat surfaces W = 1.5D + 3 mm

Radius of chamfer, R = 1.5D

- **Square (Bolt Head) Nut**

A square bolt head and nut may be drawn, showing either across flats or corners. Following relations may be adopted for the purpose:

Major or nominal diameter of bolt	= D
Thickness of nut, T	= D
Width of the nut across flats, W	= 1.5 D + 3 mm
Radius of chamfer arc, R	= 2 D

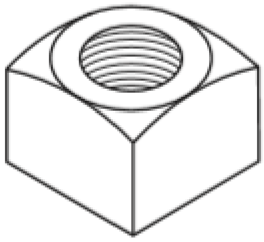


Figure 36. Square (Bolt Head) Nut

- **Flanged Nut**

This is a hexagonal nut with a collar or flange, provided integral with it. This permits the use of a bolt in a comparitively large size hole.

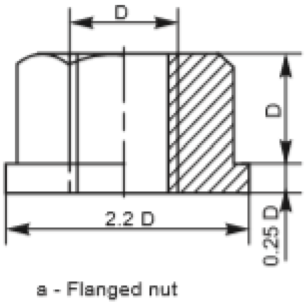


Figure 37. Flanged Nut

- **Cap Nut**

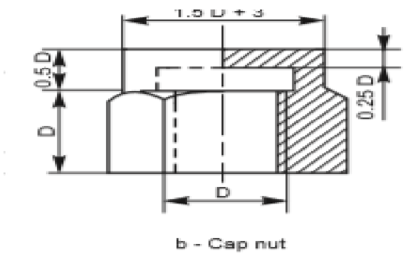


Figure 38.Cap Nut and Flanged Nut

It is a hexagonal nut with a cylindrical cap at the top. This design protects the end of the bolt from corrosion and also prevents leakage through the threads. Cap nuts are used in smoke boxes or locomotive and steam pipe connections

- **Dome Nut**

It is another form of a cap nut, having a spherical dome at the top

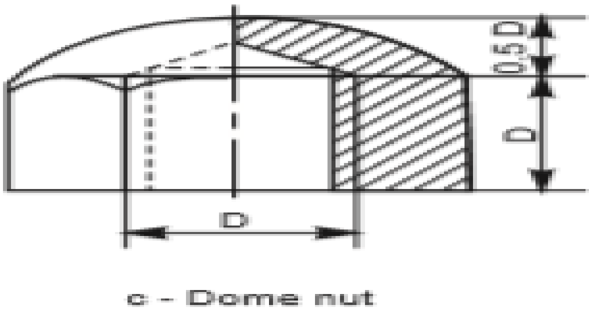


Figure 39. Dome Nut

- **Capstan Nut**

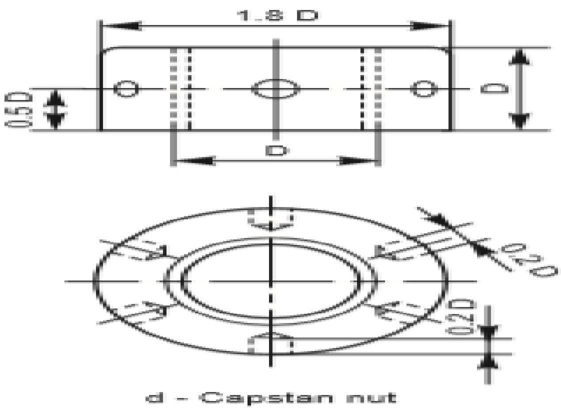


Figure 40.Capstan Nut and Dome Nut

This nut is cylindrical in shape, with holes drilled laterally in the curved surface. A tommy bar may be used in the holes for turning the nut. Holes may also be drilled in the upper flat face of the nut.

- **Ring Nut or Slotted Nut**

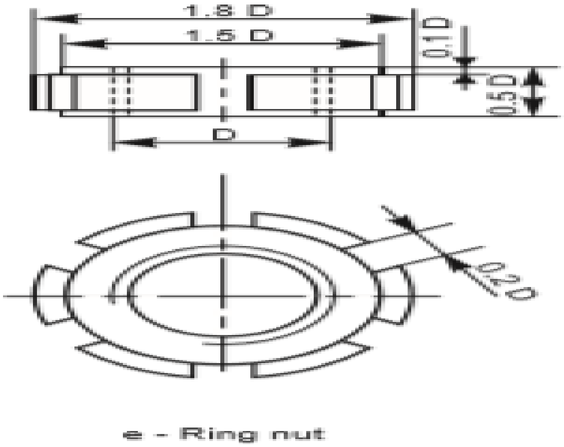


Figure 41.Ring Nut or Slotted Nut

This nut is in the form of a ring, with slots in the curved surface, running parallel to the axis. A special C-spanner is used to operate the nut. These nuts are used on large screws, where the use of ordinary spanner is inconvenient.

- **Wing Nut**

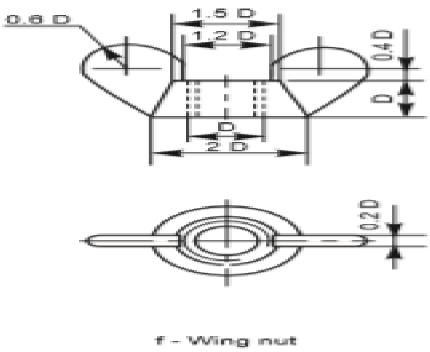


Figure 42.Wing Nut

This nut is used when frequent removal is required, such as inspection covers, lids, etc. It is operated by the thumb.

- **Washers**

A washer is a cylindrical piece of metal with a hole to receive the bolt. It is used to give a perfect seating for the nut and to distribute the tightening force uniformly to the parts under the joint. It also prevents the nut from damaging the metal surface under the joint. Figure below shows a washer, with the proportions marked.

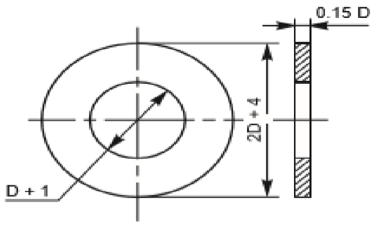


Figure 43. Washers

Figure below illustrates the views of a hexagonal headed bolt with a nut and a washer in position.

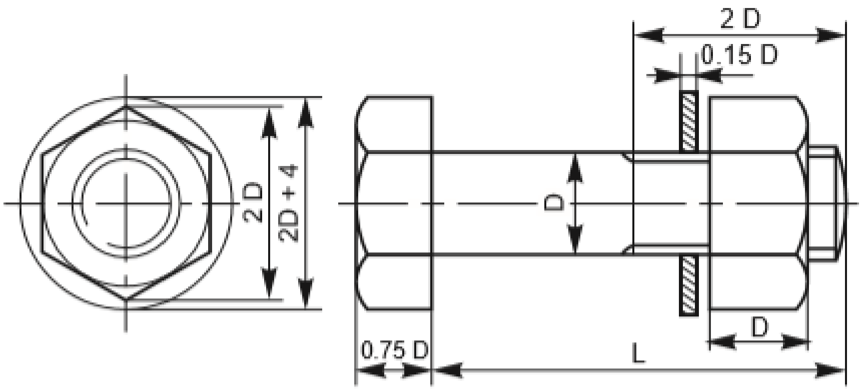


Figure 44.A hexagonal headed bolt with a nut and a washer in position

9. **Seals**

Term “seal” is used to refer to a variety of products including notary seals. Its tools used to prevent objects from being opened without authorization or detect and indicate the presence of tampering A seals is a category and a gasket is one of the important types of seal. A part from the gasket, you can find several other form of seal such as shaft seals, mechanical seal, valve stem seal, liquid sealant, rotary seals and many more.Therefore, the following article will only focus on mechanical seals and will not cover notary seals or security seals.

Mechanical seals are devices used to seal the openings of and joints between mechanisms, equipment, or systems by preventing or limiting process fluid leakage, containing pressure, and forming a barrier which protects against external contamination. Applications for these devices span across a wide range of industries, including in **electronics, marine, medical, and military sealing operations**.

Commonly used in both rotary and reciprocating equipment—such as **pumps, motors, and compressors—and stationary equipment,** seals are characterized in a variety of ways including by construction material, fluid application, and design and configuration.

Some of the types of mechanical seals available include:

- Gaskets
- O-rings
- Bellow seals
- Cartridge seals
- Labyrinth seals
- Radial shaft seals
- Axial shaft seals

Both, gasket and seal are used to avoid movement of fluid from one direction to other because of forces. A gasket is a non-moving object and once it is place in a particularly position .it can note be moved. on the other hand, oil seal can move very quickly from one to other.

One more fundamental difference between gasket and seal is that gasket is usually made from hard plastic or asbestos material. A seals’ especial oil seal is manufactured from the materials such as synthetic rubbers.

The following sections provide a brief description of each of the above-mentioned types.

Gaskets

Made of flat, elastomeric or similarly flexible material, gaskets are used to fill in the space between the flat surfaces of stationary components in a joint to prevent fluid leakage. These devices are available in a wide range of materials, shapes, and sizes, and are suitable for use as secondary seals in static applications.

- **O-Rings**

O-rings, also referred to as packing, are circular, ring-shaped mechanical gaskets that can be used in both static and dynamic applications. Designed to sit within the groove or housing between two components, the compression of the O-ring creates a fluid-tight seal capable of withstanding extreme temperatures and pressures.

- **Bellow Seals**

Bellow seals function as both a dynamic primary seal and a spring for reciprocating equipment applications. These devices also contain a secondary static seal and can be made with metal, plastic, or elastomeric bellow components.

- **Cartridge Seals**

Cartridge seals are pre-assembled and self-contained mechanical sealing devices which consist of a shaft sleeve, seal, and gland plates. These devices allow for quick installation and removal and are available in single, double, and tandem configurations.

- **Labyrinth Seals**

Labyrinth seals are composed of an inner and outer ring between which is a grooved path which hinders the escape or entrance of process fluids. These types of seals provide non-contact sealing action on rotating components.

- **Radial Shaft Seals**

Radial shaft seals, also referred to as lip seals, are used for sealing the bearing components of rotating shafts and other elements, as well as preventing the intrusion of contaminants into the equipment or system. Typically consisting of a metal insert and an elastomeric diaphragm with a spring-loaded sealing lip, these stationary seals wrap around the circumference of the rotating component to prevent leakage.

- **Axial Shaft Seals**

Axial shaft seals, also referred to as face seals, are—like radial shaft seals—used for preventing the leakage of process fluids along rotating shafts and elements and the intrusion of contaminants into the equipment or system. Consisting of a stationary and rotating component, spring-loaded seal, and secondary seal (e.g., O-ring), these seals are placed on and rotate along with the shaft and can tolerate small discrepancies in placement and shaft movement.

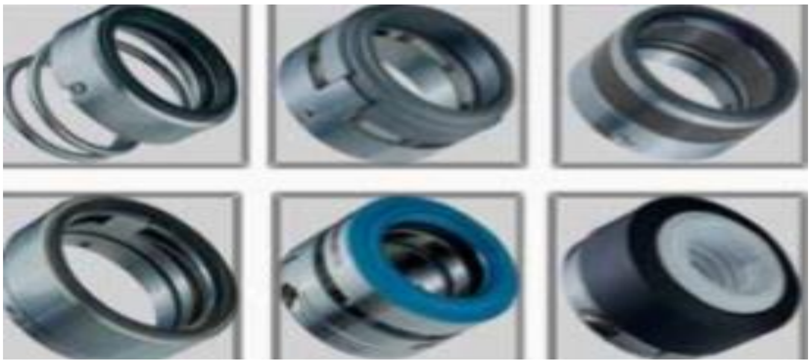


Figure 45.mechanical seals

Good sealing is essential. Seals should:

- Keep dirt out, thereby preventing premature wear of the bearing.
- Keep lubricant in, ensuring that the bearing (and possibly other components) will not run short of lubricant.

seal materials

A good sealing function requires perfect planarity of the seal faces, even under consistent gradients of temperature.Moreover, the high relative speed and pressure at which seal faces have to operate require an optimized lubrication and cooling. The combination of the above factors brings to the selection of suitable materials conveniently designed and machined.When the fluid to be sealed is abrasive it is recommended to install two hard faces such as silicon carbide or tungsten carbide. In this latter case particular care should be taken to prevent the possibility of transitory dry running which can lead to permanent damage of the seal.

- Graphite
- PTFE
- Stellite
- Chromium steel
- Ceramic
- Tungsten carbide
- Silicon carbi

Content/Topic3: Mechanisms and movement transmission

Mechanism is a device that transforms input forces and movement into a desired set of output forces and movement.

1. **Gear trains**

A gear train consists of two or more gears transmitting power from the driving shaft to the driven shaft.

The gear trains are classified into the following categories:

(i) Simple gear train
(ii) Compound gear train
(iii) Reverted gear train
(iv) Epicyclical gear train

simple gear train is one in which each shaft carries only one gear. In this type of train, the velocity ratio is equal to the number of teeth on the last driven gear to the number of teeth on the first driving gear. For example, the velocity ratio for the gear train illustrated in Fig.is given by,

$$\frac{n_1}{n_4} = \frac{z_4}{z_1}$$

The gears other than driving and driven gears are called *idler gears*. The functions of idler gear are as follows:

- Idler gears fi ll the space between the driving and driven gears.
- Idler gears change the direction of rotation of the last driven shaft relative to the fi rst driving shaft.

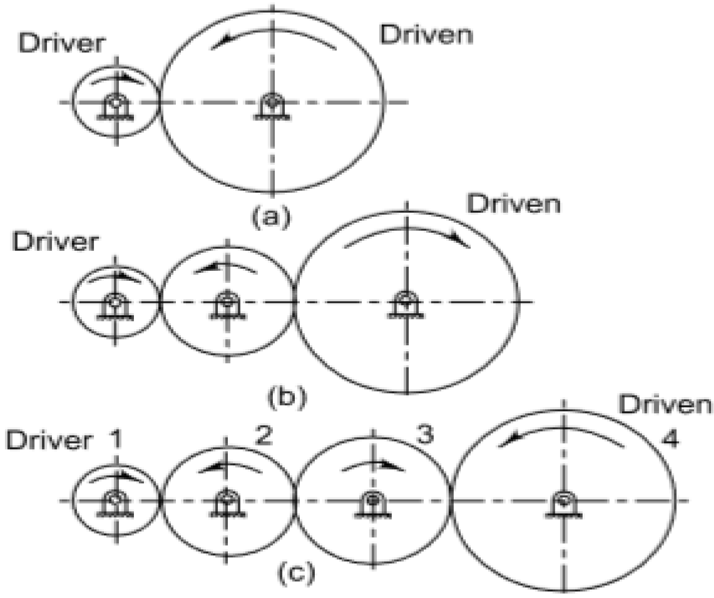


Figure 46.simple gear trains

The rules regarding direction of rotation are as follows:

- If an odd number of idler gears is used, the first and last shafts rotate in the same direction.
- If even (or zero) number of idler gears is used, the first and last shafts rotate in the opposite direction.

The main drawback of simple gear train is its large overall dimensions and weight. A compound gear train is one in which at least one shaft carries two gears. A compound gear train is illustrated in Fig. In this figure,

the intermediate shaft has two gears, one meshing with the gear on the driving shaft and the other meshing with the gear on the drive shaft. The angular velocity of two gears mounted on the intermediate shaft is the same. The velocity reduction is done in two stages. The velocity ratio is given by,

$$\frac{n_1}{n_4} = \left(\frac{z_2}{z_1}\right)\left(\frac{z_4}{z_3}\right)$$

- **Compound gear train** is compact in construction compared with simple gear train. When the number of teeth on various gears in compound gear train are selected in such a way that the centre distance between gears 1 and 2 is equal to the centre distance between gears 3 and 4 then the driving and driven shafts can be located on the same centre line. This type of arrangement is called '*reverted*' gear train.

$$m_1 (z_1 + z_2) = m_2 (z_3 + z_4)$$

where,

m_1 = module for gears 1 and 2

m_2 = module for gears 3 and 4

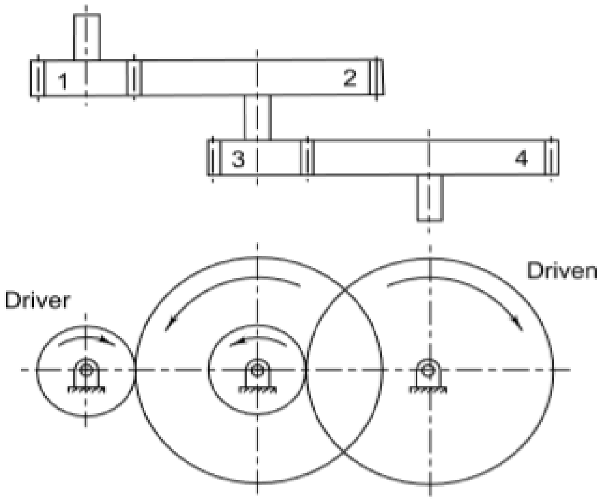


Figure 47. Compound gear train

- **Reverted gear train** it is very useful in clocks and instrument where it is desirable tohave two pointers on concentric shafts moving with specifi c velocity ratio. Reverted gear train is the most compact gearbox.

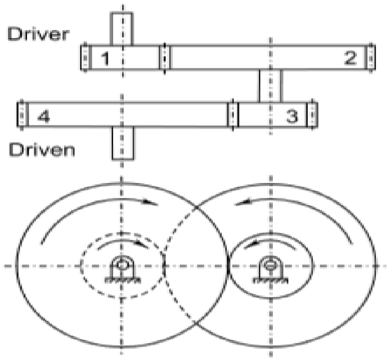


Figure 48.Reverted gear train

- **An epicyclic gear train** is illustrated in Fig. It is a gear train in which one gear is fixed and the meshing gear has a motion composed of two parts, namely, a rotation about its own axis and a rotation about the axis of the fixed gear. This type of train is also called a '*planetary*' gear train. The fixed gear is called the *sun gear,* while the revolving gear is called the *planet gear*. In the arrangement shown in the figure, the sun gear is the driving gear and the crank is connected to the driven shaft. The crank is also called the *planet carrier*. In some arrangements, there are three planet gears gears and a fixed ring gear. The epicyclic gear train has compact construction.

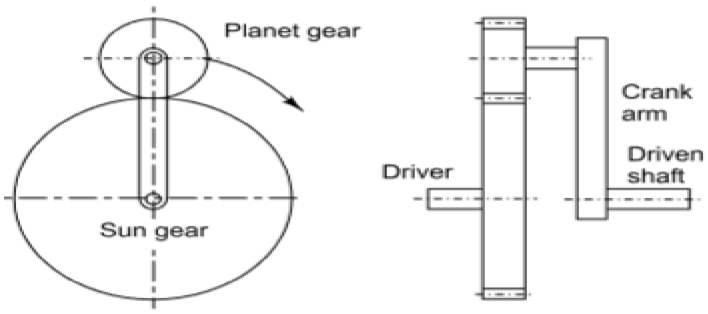


Figure 49.epicyclic gear train

2. chain

A chain drive consists of an endless chain wrapped around two sprockets as shown in Fig.41 *chain can be defined as a series of links connected by pin joints*. The sprocket is a toothed wheel with a special profile for the teeth. The chain drive is intermediate between belt and gear drives.

It has some features of belt drives and some of gear drives.

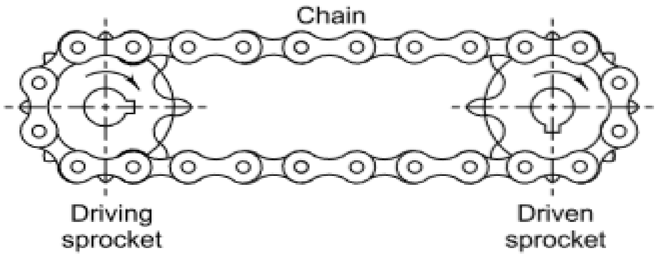


Figure 50.chain driver

The advantages of chain drives compared with belt and gear drives are as follows:

- Chain drives can be used for long as well as short centre distances. They are particularly suitable for medium centre distance, where gear drives will require additional idler gears. Thus, chain drives can be used over a wide range of centre distances.
- As shown in Fig. a number of shafts can be driven in the same or opposite direction by means of the chain from a single driving sprocket.

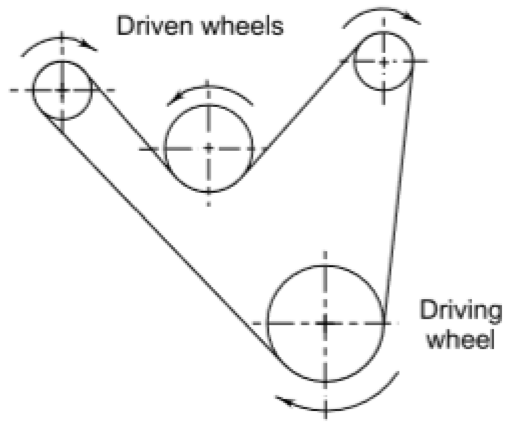


Figure 51.Chain drives

- Chain drives have small overall dimensions than belt drives, resulting in compact unit.
- A chain does not slip and to that extent, chain drive is a positive drive.
- The efficiency of chain drives is high. For properly lubricated chain, the efficiency of chain drive is from 96% to 98%.
- Chain does not require initial tension. Therefore, the forces acting on shafts are reduced.
- Chains are easy to replace.
- Atmospheric conditions and temperatures do not affect the performance of chain drives. They do not present any fi re hazard.

The disadvantages of chain drives are as follows:

(i) Chain drives operate without full lubricant fi lm between the joints unlike gears. This results in more wear at the joints. The wear increases the pitch of the chain. The chain is stretched out and may leave the sprocket, if tension is not adjusted from time to time.

(ii) Chain drives are not suitable for non-parallel shafts. Bevel and worm gears and quarterturn belt drives can be used for non-parallel shafts.

(iii) Chain drive is unsuitable where precise motion is required due to polygonal effect. The velocity of the chain is not constant resulting in non-uniform speed of the driven shaft.

(iv) Chain drives require housing.

(v) Compared with belt drives, chain drives require precise alignment of shafts. However, the centre distance is not as critical as in the case of gear drive.

(vi) Chain drives require adjustment for slack, such as a tensioning device. Compared with the belt drive, chain drives require proper maintenance, particularly lubrication.

(vii) Chain drives generate noise.

Chain drives are popular in the transportation industry, such as bicycle, motorcycle and automobile vehicle. They are used in metal and wood working machinery for the transmission of power. They are widely used in agricultural machinery, oil-well drilling rigs, building construction and materials handling equipment. Chain drives are used for velocity ratios less than 10: 1 and chain velocities of up to 25 m/s. In general, they are recommended to transmit power up to 100 kW.

There are different types of chains. With respect to their purpose, chains are classified into the following three groups:

1. Load lifting chains
2. Hauling chains
3. Power transmission chains

❖ **Load lifting chains** are used for suspending, raising or lowering loads in materials handling equipment. The popular example of this category is a '*link*' chain as illustrated in Fig.

Link chains are used in low capacity hoists, winches and hand operated cranes.

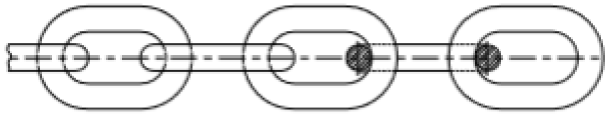


Figure 52.Link chains

They offer the following advantages:

(i) They have good flexibility in all directions.
(ii) Link chains can operate with small diameter pulleys and drums.
(iii) They are simple to design and easy to manufacture.
(iv) They produce low noise and are practically noiseless at low speeds of less than 0.1 m/s.

The disadvantages of link chains are as follows:

(i) Link chain is heavy in weight.
(ii) It is susceptible to jerks and overloads.
(iii) The failure of link chain is sudden and total.
(iv) Link chains operate at low speed.

❖ **Hauling chains** are used for carrying materials continuously by sliding, pulling or carrying in conveyors. The popular example of this category is a '*block*' chain as illustrated in Fig. 44 It consists of side plates of simple shapes and pins. It operates at medium velocities of up to 2 to 4 m/s. In general, hauling chains have long pitches because they have considerable length and mesh with sprockets whose size is not strictly limited. These chains are relatively noisy and wear rapidly because of the impact between the blocks and the sprocket. These chains are used only for conveyor applications.

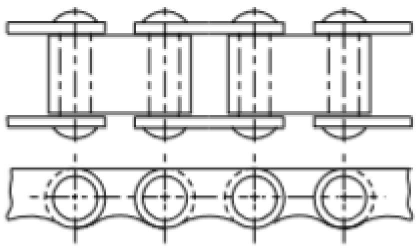


Figure 53.block chain

❖ **Power transmission chains** are used for transmitting power from one shaft to another. The discussion in this chapter is restricted to power transmission chains.

ROLLER CHAINS

The construction of a roller chain is shown in Fig. 45. It consists of alternate links made of inner and outer link plates. A roller chain consists of following five parts:

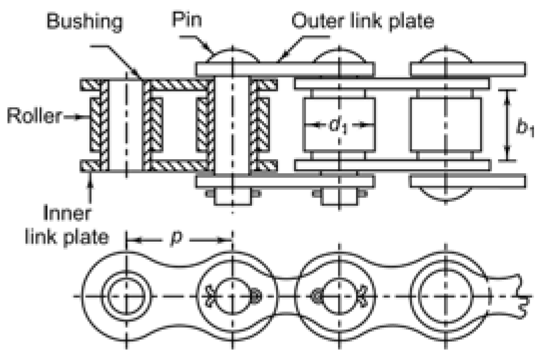


Figure 54.ROLLER CHAINS

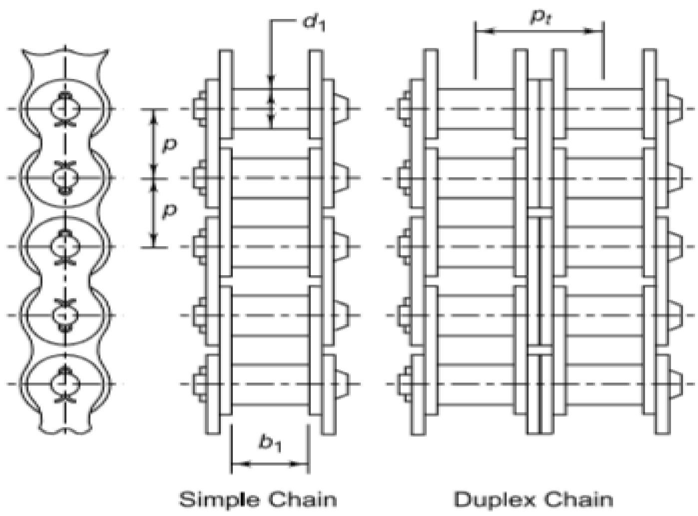


Figure 55.simple chain and duplex chain

Table1: Dimensions and breaking loads of roller chains

ISO chain number	*Pitch p (mm)*	*Roller diameter d_1 (mm) (max.)*	*Width b_1 (mm) (min.)*	*Transverse pitch p_t (mm)*	*Breaking load (min) N*		
					Simple	*Duplex*	*Triplex*
05B	8.00	5.00	3.00	5.64	4 400	7 800	11 100
06B	9.525	6.35	5.72	10.24	8 900	16 900	24 900
08A (ANSI-40)	12.70	7.95	7.85	14.38	13 800	27 600	41 400
08B	12.70	8.51	7.75	13.92	17 800	31 100	44 500
10A (ANSI-50)	15.875	10.16	9.4	18.11	21 800	43 600	65 400
10B	15.875	10.16	9.65	16.59	22 200	44 500	66 700
12A (ANSI-60)	19.05	11.91	12.57	22.78	31 100	62 300	93 400
12B	19.05	12.07	11.68	19.46	28 900	57 800	86 700
16A (ANSI-80)	25.40	15.88	15.75	29.29	55 600	111 200	166 800
16B	25.40	15.88	17.02	31.88	42 300	84 500	126 800

20A (ANSI-100)	31.75	19.05	18.90	35.76	86 700	173 500	260 200
20B	31.75	19.05	19.56	36.45	64 500	129 000	193 500
24A (ANSI120)	38.10	22.23	25.22	45.44	124 600	249 100	373 700
24B	38.10	25.40	25.40	48.36	97 900	195 700	293 600
28A (ANSI-140)	44.45	25.40	25.22	48.87	169 000	338 100	507 100
28B	44.45	27.94	30.99	59.56	129 000	258 000	387 000
32A (ANSI-160)	50.80	28.58	31.55	58.55	222 400	444 800	667 200
32B	50.80	29.21	30.99	58.55	169 000	338 100	507 100
40A (ANSI-200)	63.50	39.68	37.85	71.55	347 000	693 900	1040 900
40B	63.50	39.37	38.10	72.29	262 400	524 900	787 300
48A	76.20	47.63	47.35	87.83	500 400	1000 800	1501 300
48B	76.20	48.26	45.72	91.21	400 300	800 700	1201 000
64B	101.60	63.50	60.96	119.89	711 700	1423 400	—

The engagement of chain on sprocket wheel is shown in Fig. *D* is the pitch circle diameter of the sprocket and is called the *pitch angle*. The pitch circle diameter of the sprocket is defined as the diameter of an imaginary circle that passes through the centres of link pins as the chain is wrapped on the sprocket.

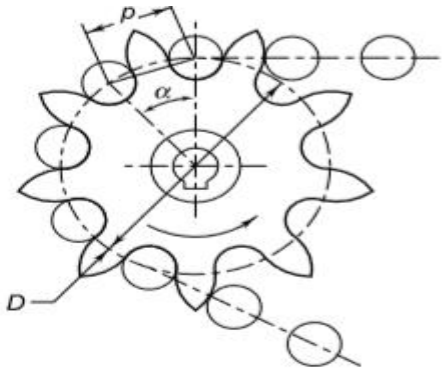


Figure 56.sprocket

$$\alpha = \frac{360}{z}$$

where ***z*** is the number of teeth on the sprocket. From the fi gure, it can be proved that

$$\sin\left(\frac{\alpha}{2}\right) = \frac{(p/2)}{(D/2)} \quad \text{or} \quad D = \frac{p}{\sin\left(\frac{\alpha}{2}\right)}$$

$$\therefore \quad D = \frac{p}{\sin\left(\frac{180}{z}\right)}$$

The velocity ratio *i* of the chain drives is given

$$i = \frac{n_1}{n_2} = \frac{z_2}{z_1}$$

where$\boldsymbol{n_1, n_2}$ = speeds of rotation of driving and driven shafts (rpm)

z_1, z_2 = number of teeth on driving and driven sprockets.

The average velocity of the chain is given by,

$$v = \frac{\pi D n}{60 \times 10^3} \qquad v = \frac{zpn}{60 \times 10^3}$$

where v is the average velocity in m/s.

The length of the chain is always expressed in terms of the number of links,

$$L = L_n \times p$$

where

L = length of the chain (mm)

L_n = number of links in the chain

The number of links in the chain is determined by the following approximate relationships:

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right)$$

where,

a = centre distance between axes of driving and driven sprockets (mm)

z_1= number of teeth on the smaller sprocket

z_2= number of teeth on the larger sprocket

The above formula is derived by analogy with the length of the belt. The first two terms represent the number of links when ($z_1 = z_2$) and the sides of the chain are parallel. The third term takes into consideration the inclination of the sides. It is obvious that the chain should contain a whole number of links. Therefore, the number of links (L_n) is adjusted to the previous or next digit so as to get an even number. It is always preferred to have an '*even*' number of links, since the chain consists of alternate pairs of inner and outer link plates.

When the chain has an odd number of links, an additional link, called '*offset*' link, is provided. The offset link is, however, weaker than the main links. After selecting the exact number of links, the centre to centre distance between the axes of the two sprockets is calculated by the following formula:

$$a = \frac{p}{4}\left\{\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right] + \sqrt{\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right]^2 - 8\left[\frac{z_2 - z_1}{2\pi}\right]^2}\right\}$$

POWER RATING OF ROLLER CHAINS

The power transmitted by the roller chain can be expressed by the elementary equation

$$\mathrm{kW} = \frac{P_1 v}{1000}$$

where

P_1 = allowable tension in the chain (N)

v = average velocity of chain (m/s)

In practice, the power rating of the roller chain is obtained on the basis of four failure criteria, viz., wear, fatigue, impact and galling.

The wear of the chain is caused by the articulation of pins in the bushings. The wear results in elongation of the chain, or in other words, the chain pitch is increased. This makes the chain '*ride out*' on the sprocket teeth, resulting in a faulty engagement.

Fatigue As the chain passes around the sprocket wheel, it is subjected to a tensile force, which varies from a maximum on the tight side to a minimum on the loose side. The chain link is, therefore, subjected to one complete cycle of fluctuating stresses during every revolution of the sprocket wheel. This results in a fatigue failure of side link plates. For infinite life, the tensile stress should be lower than the endurance limit of the link plates.

Impact The engagement of rollers with the teeth of the sprocket results in impact. When excessive, this may lead to the breakage of roller or bushing. Increasing the number of teeth on the sprocket or reducing chain tension and speed reduces the magnitude of the impact force.

Galling Galling is a stick-slip phenomenon between the pin and the bushing. When the chain tension is high, welds are formed at the high spots of the contacting area. Such microscopic welds are immediately broken due to relative motion of contacting surfaces and leads to excessive wear, even in the presence of the lubricant.

For a given application, thePower(kW) rating of the chain is determined by the following relationship:

kW rating of chain

$$= \frac{(\text{kW to be transmitted}) \times K_s}{K_1 \times K_2}$$

where

K_s= service factor

K_1 = multiple strand factor

K_2 = tooth correction factor

Table 2. *Power rating of simple roller chain*

Pinion speed (rpm)	*Power (kW)*								
	06 B	*08A*	*08 B*	*10A*	*10 B*	*12A*	*12 B*	*16A*	*16 B*
50	0.14	0.28	0.34	0.53	0.64	0.94	1.07	2.06	2.59
100	0.25	0.53	0.64	0.98	1.18	1.74	2.01	4.03	4.83
200	0.47	0.98	1.18	1.83	2.19	3.40	3.75	7.34	8.94
300	0.61	1.34	1.70	2.68	3.15	4.56	5.43	11.63	13.06
500	1.09	2.24	2.72	4.34	5.01	7.69	8.53	16.99	20.57
700	1.48	2.95	3.66	5.91	6.71	10.73	11.63	23.26	27.73
1000	2.03	3.94	5.09	8.05	8.97	14.32	15.65	28.63	34.89
1400	2.73	5.28	6.81	11.18	11.67	14.32	18.15	18.49	38.47
1800	3.44	6.98	8.10	8.05	13.03	10.44	19.85	—	—
2000	3.80	6.26	8.67	7.16	13.49	8.50	20.57	—	—

Table3: service factor

Type of driven load	*Type of input power*		
	IC engine with hydraulic drive	*Electric motor*	*IC engine with mechanical drive*
(i) *Smooth:* agitator, fan, light conveyor	1.0	1.0	1.2
(ii) *Moderate shock:* machine tools, crane, heavy conveyor, food mixer, grinder	1.2	1.3	1.4
(iii) *Heavy shock:* punch press, hammer mill, reciprocating conveyor, rolling mill drive	1.4	1.4	1.7

Table 5: tooth correction factor (K_2)

Number of teeth on the driving sprocket	K_2
15	0.85
16	0.92
17	1.00
18	1.05
19	1.11
20	1.18
21	1.26
22	1.29
23	1.35
24	1.41
25	1.46
30	1.73

Table 4: multiple strand factor (K_1)

Number of strands	K_1
1	1.0
2	1.7
3	2.5
4	3.3
5	3.9
6	4.6

SPROCKET WHEELS

There are different constructions for sprocket wheels as shown in Fig. 1. Small sprockets up to 100 mm in diameter are usually made of a disk or a solid disk with a hub on one side

They are machined from low carbon steel bars. Large sprockets with more than 100 mm diameter are either welded to a steel hub or bolted to a cast iron hub.

In general, sprockets are made of low carbon or medium carbon steels. In certain applications, stainless steel is used for sprockets. When the chain velocity is less than 180 m/min, the teeth of the sprocket wheel are heat-treated to obtain a hardness of 180 BHN.

The teeth are hardened either by carburizing in case of low carbon steel or by quenching and tempering in case of high carbon steel.

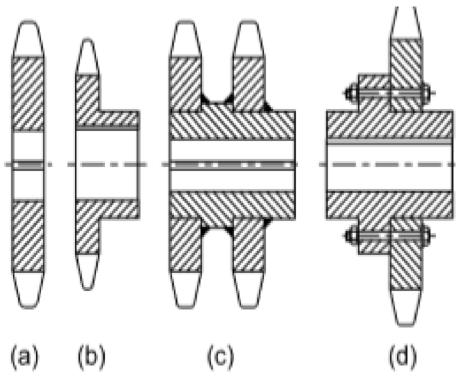


Figure 57.construction of sprocket wheel

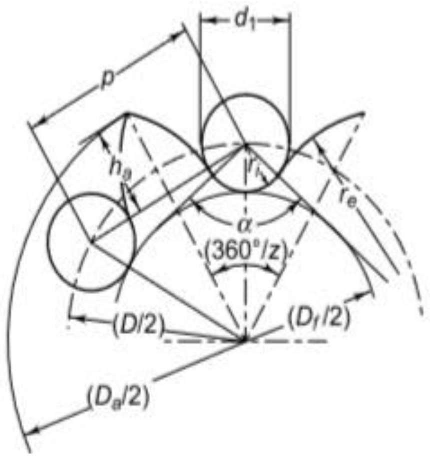


Figure 58.Tooth Profile le of Sprocket

The difference between the gear and the sprocket is as follows:

1. A gear meshes with another gear. A sprocket meshes with an '*intermediate*' link, namely chain, which in turn meshes with another sprocket.
2. The face width of gear is usually more with respect to its diameter. The sprockets are comparatively thin so as to fit between inner link plates of the chain.
3. The teeth of gears have involute profile, while circular arcs are used for the profile of sprocket teeth.

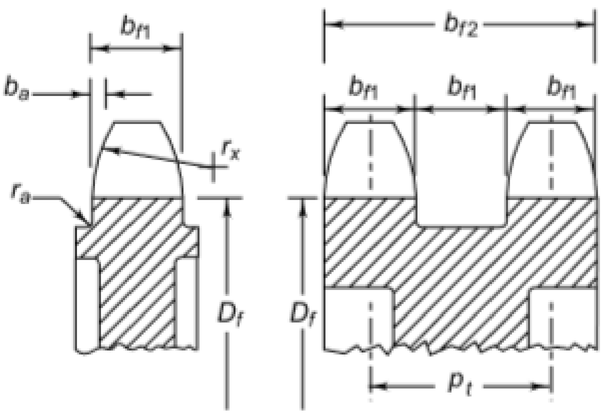


Figure 59.Rim profile of sprocket

Table 6

Dimension	Notation	Equation
1. Chain pitch	p	(Table 14.1)
2. Pitch circle diameter	D	$D = \dfrac{p}{\sin\left(\dfrac{180}{z}\right)}$
3. Roller diameter	d_1	(Table 14.1)
4. Width between inner plates	b_1	(Table 14.1)
5. Transverse pitch	p_t	(Table 14.1)
6. Top diameter	D_a	$(D_a)_{max.} = D + 1.25p - d_1$ $(D_a)_{min.} = D + p\left(1 - \dfrac{1.6}{z}\right) - d_1$
7. Root diameter	D_f	$D_f = D - 2r_i$
8. Roller seating radius	r_i	$(r_i)_{max.} = (0.505d_1 + 0.069\sqrt[3]{d_1})$ $(r_i)_{min.} = 0.505d_1$
9. Tooth flank radius	r_e	$(r_e)_{max.} = 0.008d_1\,(z^2 + 180)$ $(r_e)_{min.} = 0.12d_1\,(z + 2)$
10. Roller seating angle	α	$\alpha_{max.} = \left[120 - \dfrac{90}{z}\right]$ $\alpha_{min.} = \left[140 - \dfrac{90}{z}\right]$
11. Tooth height above the pitch polygon	h_a	$(h_a)_{max.} = 0.625p - 0.5d_1 + \dfrac{0.8p}{z}$ $(h_a)_{min.} = 0.5(p - d_1)$
12. Tooth side radius	r_x	$(r_x)_{min.} = p$
13. Tooth width	b_{f1}	$b_{f1} = 0.93b_1$ if $p \le 12.7$ mm $b_{f1} = 0.95b_1$ if $p > 12.7$ mm
14. Tooth side relief	b_a	$b_a = 0.1p$ to $0.15p$

There are two methods for adjustment of chain tension. They are as follows:

1. Change the centre distance by moving the axis of one of the sprockets.
2. Provide an adjustable idler sprocket.

The objectives of chain lubrication are as follows:

(i) To reduce the wear of chain components

(ii) To protect the chain against rust and corrosion

(iii) To carry away the frictional heat

(iv) To prevent seizure of pins and bushes

(v) To cushion shock loads and protect the chain

3.Belt and pulley

Belt, chain and rope drives are called '*flexible*' drives.

There are two types of drives:

- **Rigid and flexible**. Gear drives are called rigid or non-flexible drives. In gear drives, there is direct contact between the driving and driven shafts through the gears.

In flexible drives, there is an intermediate link such as belt, rope or chain between the driving and driven shafts.In flexible drives, the rotary motion of the driving shaft is first converted into translator motion of the belt or chain and then again converted into rotary motion of the driven shaft.

The advantages of flexible drives over rigid drives are as follows:

- Flexible drives transmit power over a comparatively long distance due to an intermediate link between driving and driven shafts.
- Since the intermediate link is long and flexible, it absorbs shock loads and damps vibrations.
- Flexible drives provide considerable flexibility in the location of the driving and driven shafts. The tolerances on the centre distance are not critical as compared with a gear drive.
- Flexible drives are cheap compared to gear drives. Their initial and maintenance costs are low.

The disadvantages of flexible drives are as follows:

- They occupy more space.
- The velocity ratio is relatively small.
- In general, the velocity ratio is not constant.

A belt drive consists of three elements—driving and driven pulleys and an endless belt, which envelopes them.

Belt drives offer the following advantages compared with other types of drives:

a. Belt drives can transmit power over considerable distance between the axes of driving and driven shafts.
b. The operation of belt drive is smooth and silent
c. They can transmit only a definite load, which if exceeded, will cause the belt to slip over the pulley, thus protecting the parts of the drive against overload.
d. They have the ability to absorb the shocks and damp vibration.
e. They are simple to design.
f. They have low initial cost.

The disadvantages of belt drives compared to other types of drives are as follows:

a. Belt drives have large dimensions and occupy more space.
b. The velocity ratio is not constant due to belt slip.
c. They impose heavy loads on shafts and bearings.
d. There is considerable loss of power resulting in low efficiency.
e. Belt drives have comparatively short service life.

Belt drives are mainly used in electric motors, automobiles, machine tools and conveyors.

Depending upon the shape of the cross-section, belts are classified as **flat belts** and **V-belts**.

Flat belts have a narrow rectangular cross-section, while *V-belts* have a trapezoidal cross-section.

Flat belts offer the following advantages over V-belts:

(i) They are relatively cheap and easy to maintain. Their maintenance consists of periodic adjustment in the centre distance between two shafts in order to compensate for stretching and wear. They do not require precise alignment of shafts and pulleys. When worn out, they are easy to replace.

(ii) A flat belt drive can be used as a clutch by making a simple provision of shifting the belt from tight to loose pulley and vice versa.

(iii) Different velocity ratios can be obtained by using a stepped pulley, where the belt is shifted from one step to another, having different diameters.

(iv) They can be used in dusty and abrasive atmosphere and require no closed casing.

(v) The design of flat-belt drive is simple and inexpensive.

The major disadvantage of flat belt drives over V-belt drives are as follows:

(i) The power transmitting capacity of flat-belt drive is low.

(ii) The velocity ratio of flat belt-drive is lower than V-belt drive.

(iii) Flat-belt drives have large dimensions and occupy more space compared to V-belt drives.

(iv) Flat belts generate more noise than V-belts.

(v) In general, flat-belt drives are horizontal and not vertical

V-belts are very popular where an electric motor is used as the prime mover to drive compressors, pumps, fans, positive displacement pumps, blowers and machine tools. They are also popular in automobiles to drive accessories on petrol or diesel engines.

The velocity ratio for flat belt is up to 4:1. For V-belts the velocity ratio is up to 7:1. For chain drives it can be up to 15:1.

Flat and V-belts are widely used. However, there are certain applications, where '*round*' belts are used. The round belt is illustrated in Fig. Round belts are made of leather, canvas or rubber.

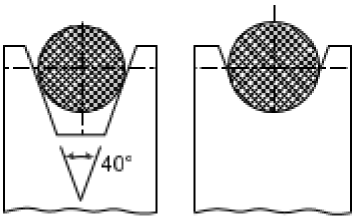


Figure 60.Round belts

There are two types of grooves for pulley

- trapezoidal with an angle of 40° between the sides and
- half round with a radius equal to that of the belt.

The advantages of round belts are as follows:

- Round belts can operate satisfactorily over pulleys in several different planes. They are suitable for 90° twist, reverse bends or serpentine drives.
- They can be stretched over the pulley and snapped into the groove very easily. This makes the assembly and replacement simple.
- Round belts are limited to light duties. They are used in dishwasher drives, sewing machines, vacuum cleaners and light textile machinery.

BELT CONSTRUCTIONS

The desirable properties of belt materials are as follows:

(i) The belt material should have high coefficient of friction with the pulleys.

(ii) The belt material should have high tensile strength to withstand belt tensions.

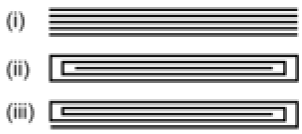
(iii) The belt material should have high wear resistance.

(iv) The belt material should have high flexibility and low rigidity in bending in order to avoid bending stresses while passing over the pulley.

The Belts are made of leather, canvas, rubber or rubberized fabric and synthetic materials.

There are two types of flat belts—*leatherbelt* and *fabric rubber belt*. The leather belt is made of the best quality leather obtained from either sides of the backbone of a steer.

The main advantage of leather belt is the high coefficient of friction and consequently, high power transmitting capacity.



(b) Fabric Rubber Belt

Figure 61.Flat belt

The fabric rubber belts are made from several layers of canvas or cotton-duck impregnated with rubber as shown in Fig. The fabric transmits major portion of the load. The rubber protects the fabric against damage and increases the coefficient of friction.

V-belts are made of polyester fabric and cords, with rubber reinforcement. The cross-section of the V-belt is shown in Fig.50. It consists of the following three parts

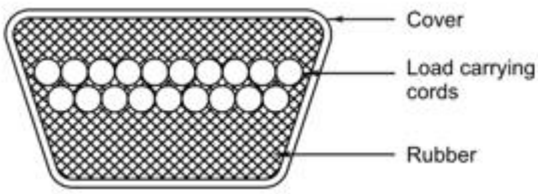


Figure 62.Cross-section of V-belt

GEOMETRICAL RELATIONSHIPS

There are two types of belt construction—**open** and **crossed**

The difference between these two constructions is as follows:

- An open belt drive is a belt drive in which the belt proceeds from the top of one pulley to the top of another without crossing. A crossed belt drive is a belt drive in which the belt proceeds from the top of one pulley to the bottom of another and crosses over itself. In both cases, the driving and driven shafts are parallel.
- In an open belt drive, both driving and driven pulleys rotate in the same direction. In a crossed belt drive, driving and driven pulleys rotate in the opposite direction.
- In crossed belt drive, the angle of wrap is more. Therefore, power transmitting capacity of a crossed belt drive is more than that of an open belt drive.
- In crossed belt drive, the belt rubs against itself while crossing. Also, the belt has to bend in two different planes. These two factors increase the wear and reduce the life of the belt.

- In open belt drives, when the centre distance is more, the belt whips, i.e., vibrates in a direction perpendicular to the direction of motion. When the centre distance is small, the belt slip increases. Both these factors limit the use of an open belt drive. Crossed belt drives do not have these limitations.
- Open belt drives are more popular than crossed belt drives.

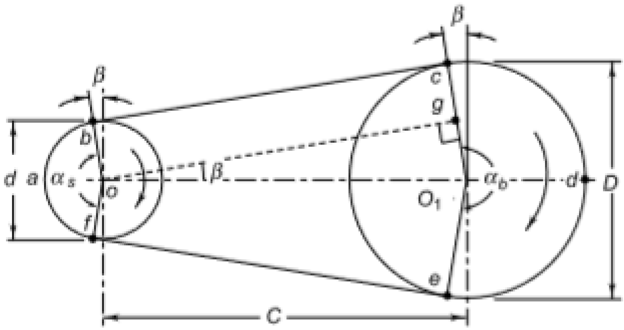


Figure 63.Open belt drives

D = diameter of big pulley (mm)
d = diameter of small pulley (mm)
C = centre distance (mm)

Construction Draw a line $\overline{og}$ perpendicular to the line $\overline{o_1c}$. The area *ogcb* is a rectangle.

$$\therefore \quad ob = gc$$

From triangle ogo_1,

$$\sin\beta = \frac{o_1g}{oo_1} = \frac{o_1c - gc}{oo_1} = \frac{o_1c - ob}{oo_1}$$

$$= \frac{D/2 - d/2}{C} = \frac{D-d}{2C}$$

$$\therefore \quad \sin\beta = \frac{D-d}{2C}$$

also $\alpha_s = (180 - 2\beta)$ and $\alpha_b = (180 + 2\beta)$

Therefore, $$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right) \qquad (13.1)$$

$$\alpha_b = 180 + 2\sin^{-1}\left(\frac{D-d}{2C}\right) \qquad (13.2)$$

Refer to Fig.13.5 again. The length of the belt (L) is given by,

$$L = \text{arc}\ (fab) + \overline{bc} + \text{arc}\ (cde) + \overline{ef}$$

$$= \frac{d}{2}(\alpha_s) + \overline{og} + \frac{D}{2}(\alpha_b) + \overline{og}$$

$$= \frac{d}{2}(\pi - 2\beta) + C\cos\beta + \frac{D}{2}(\pi + 2\beta) + C\cos\beta$$

or $$L = \frac{\pi(D+d)}{2} + \beta(D-d) + 2C\cos\beta \qquad \text{(a)}$$

For small values of β,

$$\beta = \sin\beta = \left(\frac{D-d}{2C}\right)$$

and $$\cos\beta = 1 - 2\sin^2\left(\frac{\beta}{2}\right) = 1 - \frac{\beta^2}{2}$$

$$= 1 - \frac{(D-d)^2}{8C^2}$$

Substituting these values of β and $\cos\beta$ in Eq. (a), we get,

$$L = \frac{\pi(D+d)}{2} + \left(\frac{D-d}{2C}\right)(D-d) + 2C\left[1 - \frac{(D-d)^2}{8C^2}\right]$$

$$= \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{2C} + 2C - \frac{(D-d)^2}{4C}$$

$$\therefore \quad L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C} \qquad (13.3)$$

A crossed belt drive is shown in Fig. 13.6.

Construction Draw a line $\overline{og}$ perpendicular to the line $\overline{o_1c}$ The area $ofcg$ is a rectangle.

$$\therefore \quad cg = of$$

Cross belt driver

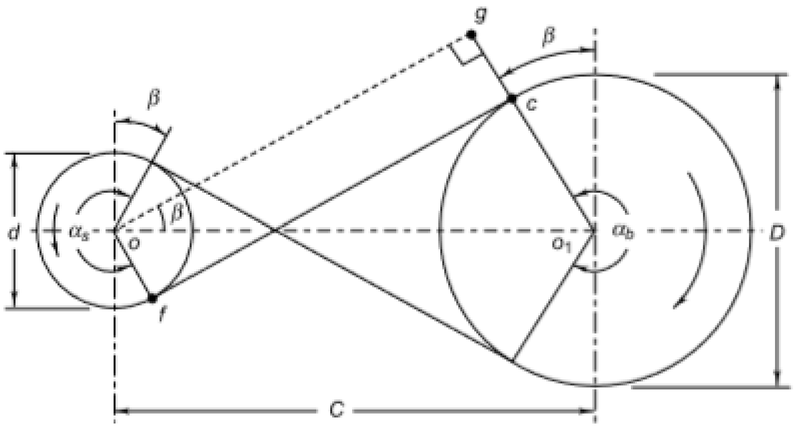


Figure 64.Cross belt driver

From triangle ogo_1,

$$\sin\beta = \frac{o_1g}{oo_1} = \frac{o_1c + cg}{oo_1} = \frac{o_1c + of}{oo_1}$$

$$= \frac{D/2 + d/2}{C} = \frac{D+d}{2C}$$

$$\sin\beta = \frac{D+d}{2C}$$

$$\alpha_s = \alpha_b = (180^\circ + 2\beta)$$

$$\alpha_s = \alpha_b = 180 + 2\sin^{-1}\left(\frac{D+d}{2C}\right)$$

The procedure used for open belt drive can be used for crossed belt drive and it can be proved that the belt length L for a crossed belt drive is given by

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D+d)^2}{4C}$$

It should be noted that in the above expressions, *D* and *d* are pitch diameters of pulleys while *L* is the pitch length of the belt.

The power losses in the belt drive are made up of the following factors

Power loss due to belt creep on the pulley.

(i) Power loss due to internal friction between the particles of the belt in alternate bending and unbending over the pulley.

(ii) Power loss due aerodynamic resistance to the motion of pulleys and belt.

(iii) Power loss due to friction in bearings of pulleys.

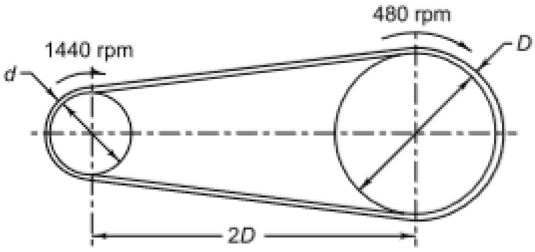
Example question.

The layout of a leather belt drive transmitting 15 kW of power is shown in Fig. 13.13. The centre distance between the pulleys is twice the diameter of the bigger pulley. The belt should operate at a velocity of 20 m/s approximately and the stresses in the belt should not exceed

2.25 N/mm^2. The density of leather is 0.95 g/cc and the coefficient of friction is 0.35. The thickness of the belt is 5 mm. Calculate: (i) the diameter of pulleys;

(ii) the length and width of the belt; and

(iii) the belt tensions.



Given kW = 15 $v = 20$ m/s $C = 2D$
$t = 5$ mm $\rho = 0.95$ g/cc $\sigma = 2.25$ N/mm^2
$f = 0.35$

Step I *Diameter of pulleys*

$$v = \frac{\pi d n}{60(1000)}$$

or
$$d = \frac{60(1000)v}{\pi n} = \frac{60(1000)(20)}{\pi(1440)}$$
$$= 265.26 \text{ mm (or 270 mm)}$$

The diameters of pulleys are

$$d = 270 \text{ mm}$$

and
$$D = \frac{270(1440)}{(480)} = 810 \text{ mm} \quad (a$$

Step II *Belt length*

$$C = 2D = 2(810) = 1620 \text{ mm}$$

From Eq. (13.3),

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$
$$= 2(1620) + \frac{\pi(810+270)}{2} + \frac{(810-270)^2}{4(1620)}$$
$$= 4981.46 \text{ mm} \quad (b)$$

Step III *Belt width and belt tensions*

The correct belt velocity is given by,

$$v = \frac{\pi dn}{60(1000)} = \frac{\pi(270)(1440)}{60(1000)} = 20.36 \text{ m/s}$$

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$= 180 - 2\sin^{-1}\left(\frac{810-270}{2\times 1620}\right) = 160.8°$$

or $\alpha_s = \left(\frac{160.8}{180}\right)\pi = 2.81 \text{ rad}$

The density of leather is given as 0.95 g per cubic cm. The volume of 1 metre belt in cubic cm is given by,

$$\text{volume} = (\text{length}) \times (\text{width}) \times (\text{thickness})$$

$$= (100)\left(\frac{b}{10}\right)\left(\frac{5}{10}\right)$$

where b is the width in mm.

$$\therefore \quad m = (0.95)(100)\left(\frac{b}{10}\right)\left(\frac{5}{10}\right) \text{ g/m}$$

$$= (0.95)(100)\left(\frac{b}{10}\right)\left(\frac{5}{10}\right)(10^{-3}) \text{ kg/m}$$

$$= (4.75 \times 10^{-3})b \text{ kg/m}$$

or $mv^2 = (4.75 \times 10^{-3})b\,(20.36)^2 = 1.97b$

also $e^{f\alpha} = e^{(0.35)(2.81)} = 2.67$

From Eq. (13.6),

$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha} \quad \text{or} \quad \frac{P_1 - 1.97b}{P_2 - 1.97b} = 2.67$$

or $$P_1 - 2.67P_2 + 3.29b = 0 \qquad \text{(i)}$$

The maximum stress in the belt is given as 2.25 N/mm².

$$\sigma = \frac{\text{maximum tension}}{\text{area of cross-section of belt}} = \frac{P_1}{A}$$

$$\therefore \quad P_1 = \sigma A = 2.25(5b) = (11.25\, b) \text{ N} \qquad \text{(ii)}$$

From Eq. (13.8),

$$P_1 - P_2 = \frac{1000(\text{kW})}{v} = \frac{1000(15)}{20.36} = 736.74 \text{ N} \qquad \text{(iii)}$$

Solving Eqs (i), (ii) and (iii) simultaneously, we have

$$b = 127.02 \text{ mm or } 130 \text{ mm} \qquad \text{(b2)}$$

$$P_1 = 1428.98 \text{ N} \quad \text{and} \quad P_2 = 692.26 \text{ N} \qquad \text{(c)}$$

PULLEYS FOR FLAT BELTS

The pulleys for flat belts consist of three parts, viz., rim, hub and arms or web. The rim carries the belt. The hub connects the pulley to the shaft. The arms or web join the hub with the rim. There are two types of pulleys that are used for flat belts:

cast iron pulleys and mild steel pulley. The pulley diameters are calculated in belt drive design

The minimum pulley diameter depends upon the following two factors:

1.The number of plies in the belt

2.The belt speed

There is a specific term, '*crowning*' of pulley in flat belt drive. The thickness of the rim is slightly increased in the centre to give it a convex or conical shape. This is called '*crown*' of the pulley. The crown is provided only on one of the two pulleys.

The objectives of providing crown are as follows:

(i) The crown on the pulley helps to hold the belt on the pulley in running condition.

(ii) The crown on the pulley prevents the belt from running off the pulley.

(iii) The crown on the pulley brings the belt to running equilibrium position near the mid plane of the pulley.

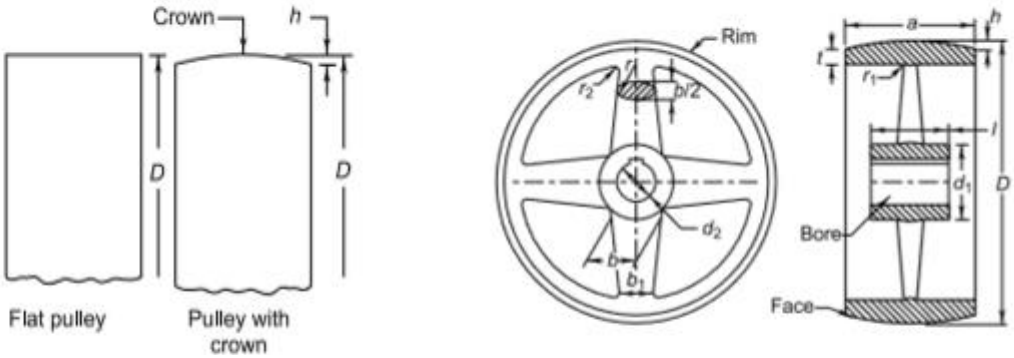


Figure 65.The crown for the pulley and cast iron pulley

V-BELTS

The dimensions for the cross-section of V-belt are shown in Fig.The following notations are used for the dimensions of the cross-section:

✓ ***Pitch Width (W_p)***It is the width of the belt at its pitch zone. This is the basic dimension for standardization of belt and corresponding pulley groove.

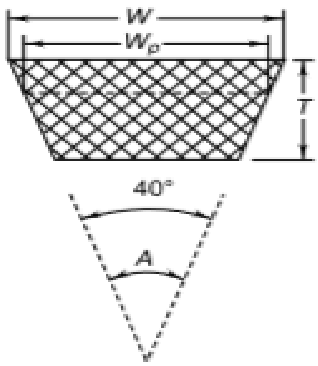


Figure 66.V-BELTS

✓ ***Nominal Top Width (W)*** It is the top width of the trapezium outlined on the cross-section of the belt.

✓ ***Nominal Height (T)*** It is the height of the trapezium outlined on the cross-section of the belt.

- ✓ ***Angle of Belt (A)*** It is the included angle obtained by extending the sides of the belt. The standard value of the belt angle is 40°.
- ✓ ***Pitch Length (L_p)*** It is the length of the pitch line of the belt. This is the circumferential length of the belt at the pitch width.

There are six basic symbols— *Z*, *A*, *B*, *C*, *D* and *E*—for the cross-section of Vbelts. Z-section belts are occasionally used for low power transmission and small pulley diameters, while *A*, *B*, *C*, *D* and *E* section belts are widely used as general purpose belts. V-belts are designated by the symbol of cross-section followed by nominal pitch length along with symbol L_p., e.g., a V-belt of cross-section *B* and with pitch length 4430 mm is designated as *B* 4430L_p. The recommended values of standard pitch lengths (L_p) are given in Table

Belt section	*Pitch width W_p (mm)*	*Nominal top width W(mm)*	*Nominal Height T (mm)*	*Recommended Minimum pitch diameter of pulley (mm)*	*Permissible Minimum pitch diameter of pulley (mm)*
Z	8.5	10	6	85	50
A	11	13	8	125	75
B	14	17	11	200	125
C	19	22	14	315	200
D	27	32	19	500	355
E	32	38	23	630	500

Table. ***Dimensions of standard cross-section***

The basic procedure for the selection of V-belts consists of the following steps:

- Determine the correction factor according to service (F_a) from Table It depends upon the type of driving unit, the type of driven machine and the operational hours per day
- Calculate the design power by the following relationship:
- Plot a point with design power as *X* coordinate and input speed as *Y* co-ordinate in Fig. The location of this point decides the type of cross-section of the belt. In a borderline case, such as the point located on the borderline of cross-sections *B* and *C*, alternative

(ii) Calculate the design power by the following relationship:

$$\text{Design power} = F_a \text{ (transmitted power)}$$

Determine the recommended pitch diameter of the smaller pulley from Table It depends upon the cross-section of the belt. Calculate the pitch diameter of the bigger pulley by the following relationship:

$$D = d\left[\frac{\text{speed of smaller pulley}}{\text{speed of bigger pulley}}\right] = d\left[\frac{\text{input speed}}{\text{output speed}}\right]$$

The above values of *D* and *d* are compared with the preferred pitch diameters given in Table .In case of non-standard value, the nearest values of *d* and *D* should be taken from Table.

Determine the pitch length of belt *L* by the following relationship,

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

Compare the above value of *L* with the preferred pitch length *L* in Table in case of a non-standard value, the nearest value of pitch length from Table. should be taken.
Find out the correct centre distance *C* by substituting the above value of *L* in the following equation:

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

Calculate the arc of contact for the smaller pulley by the following relationship:

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

(v) Depending upon the type of cross-section, refer to the respective table from Table and determine the power rating (P_r) of single V-belt. It depends upon three factors—speed of faster shaft, pitch diameter of smaller pulley and the speed ratio.

(vi) The last step in the selection procedure is to find out the number of belts. It depends upon the design power and the power transmitting capacity of one belt. The number of belts is obtained by the following relationship:

$$\text{Number of belts} = \frac{P \times F_a}{P_r \times F_c \times F_d}$$

The dimensions of V-grooved pulley[1] for V-belts are given in Table. Such pulleys are usually made of grey cast iron of Grade-FG 250. In some applications, the pulleys are made of carbon steel casting. The notations used in the table are as follows:

l_p = Pitch width of pulley groove or pitch width of belt. It is the width of the belt at its neutral axis. The line in the V-belt, whose length remains unchanged when the belt is deformed under tension, is called its neutral axis. *b* = Minimum height of groove above the pitch line

h = Minimum depth of groove below the pitch line

e = Centre to centre distance of adjacent grooves.

f = Distance of the edge of pulley to first groove center Groove angle

d_p = Pitch diameter of pulley. It is diameter of the pulley measured at the pitch width of the groove

g = Minimum top width of the groove

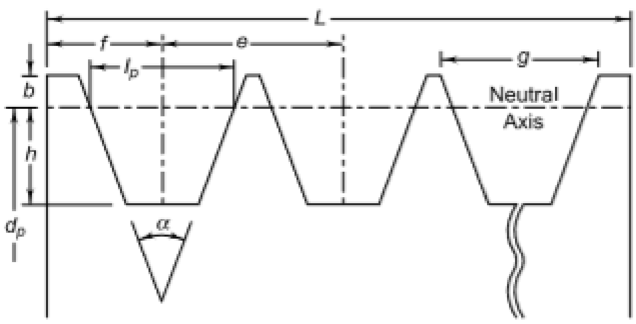


Table 8. ***Dimensions of V-grooved pulleys***

Groove section	l_p mm	b mm	h mm	e mm	f mm	α^o	d_p mm	g mm	Outside diameter
Z	8.5	2.00	9	12 ± 0.3	7–9	34	Up to 80	9.7	$d_p + 4.0$
A	11	2.75	11	15 ± 0.3	9–12	34	Up to 118	12.7	$d_p + 5.5$
B	14	3.50	14	19 ± 0.4	11.5–14.5	34	Up to 190	16.1	$d_p + 7.0$
C	19	4.80	19	25.5 ± 0.5	16–19	34	Up to 315	21.9	$d_p + 9.6$
D	27	8.10	19.9	37 ± 0.6	23–27	36	Up to 499	21.9	$d_p + 16.2$
E	32	9.60	23.4	44.5 ± 0.7	28–33	36	Up to 629	38.2	$d_p + 19.2$
$L = (x-1)e + 2f$ where x is the number of grooves									

Example question: The following data is given for an open-type V-belt drive:

Diameter of driving pulley = 200 mm

Diameter of driven pulley = 600 mm

Groove angle for sheaves = 34°

Mass of belt = 0.5 kg/m

Maximum permissible tension in belt = 500 n

Coefficient of friction = 0.2

Contact angle for smaller pulley = 157°

Speed of smaller pulley = 1440 rpm

Power to be transmitted = 10 kw

How many V-belts should be used, assuming each belt takes its proportional part of the load?

solution

Given $kW = 10$ $n = 1440$ rpm $D = 600$ mm

$d = 200$ mm $\theta = 34^o$ $m = 0.5$ kg/m $f = 0.2$

$\alpha_s = 157^o$ allowable belt tension = 500 N

Step I *Belt tensions*

$$\frac{f\alpha}{\sin\left(\frac{\theta}{2}\right)} = \frac{0.2\left(\frac{157}{180}\right)\pi}{\sin\left(\frac{34}{2}\right)} = 1.874$$

$$e^{f\alpha/\sin(\theta/2)} = e^{1.874} = 6.52$$

$$v = \frac{\pi dn}{60 \times 10^3} = \frac{\pi(200)(1440)}{60 \times 10^3} = 15.08 \text{ m/s}$$

$$mv^2 = 0.5(15.08)^2 = 113.70$$

From Eq. (13.7),

$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha/\sin(\theta/2)}$$

$$\therefore \quad \frac{P_1 - 113.70}{P_2 - 113.70} = 6.52$$

$$P_1 - 6.52P_2 + 627.61 = 0 \qquad \text{(i)}$$

From Eq. (13.8),

$$\text{kW} = \frac{(P_1 - P_2)v}{1000} \quad \text{or} \quad 10 = \frac{(P_1 - P_2)(15.08)}{1000}$$

$$P_1 - P_2 - 663.13 = 0$$

Solving Eq. (i) and (ii),

$$P_1 = 896.96 \text{ N} \quad \text{and} \quad P_2 = 233.83 \text{ N}$$

Step II *Number of belts*

$$\text{Number of belts} = \frac{\text{Maximum tension in belt}}{\text{Allowable belt load}}$$

$$= \frac{896.96}{500} = 1.79 \text{ or } 2 \text{ belts}$$

RIBBED V-BELTS

Ribbed V-belts are fl at belts with a series of evenly spaced teeth on the inside of the circumference, which mesh with the teeth in the pulley or sprocket as shown in Fig. Therefore, they can maintain exactly the same angular position of the driven shaft with respect to driving shaft. A ribbed V-belt is a positive drive. These belts combine the high velocity characteristic of the belt drive with positive power transmission of the chain drive.

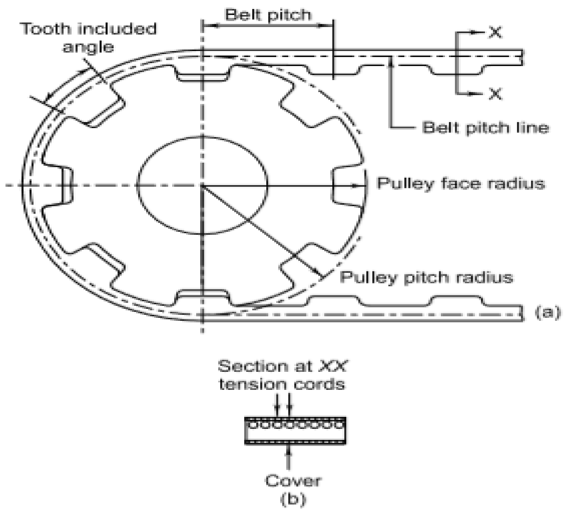


Figure 67.ribbed v-belts

Ribbed V-belts offer the following advantages:

(i) It is a positive drive; there is no slip and no variation in output speed.

(ii) It has high strength to weight ratio, which allows for high pitch line velocities of up to 80 m/s.

(iii) The belt is thin and flexible, which permits the use of small diameter pulleys as small as 15 mm in diameter.

(iv) The length of the belt does not increase appreciably during service due to steel cords. Therefore, no tensioning device is required like flat belt drive.

(v) The ribbed V-belt drive does not require any lubrication like chain drive.

(vi) The ribbed V-belt does not require initial tension like flat belt. This reduces bearing reactions.

The disadvantages of ribbed V-belt drives are as follows:

(i) It is costly compared with flat or V-belts.

(ii) The construction of sprocket is difficult compared with the pulleys for flat or V-belts.

(iii) It is more sensitive to misalignment than V or flat belts.

4.Cable drives / Rope Drives

Rope Drives The rope drives are widely used where a large amount of power is to be transmitted, from one pulley to another, over a considerable distance. It may be noted that the use of flat belts is limited for the transmission of moderate power from one pulley to another when the two pulleys are not more than 8 metres apart. If large amounts of power are to be transmitted, by the flat belt, then it would result in excessive belt cross-section. The ropes drives use the following two types of ropes:

1. Fibre ropes, and
2. Wire ropes.

The fibre ropes operate successfully when the pulleys are about 60 metres apart, while the wire ropes are used when the pulleys are upto 150 metres apart.

Fibre Ropes

The ropes for transmitting power are usually made from fibrous materials such as hemp, manila and cotton.

When the hemp and manila ropes are bent over the sheave, there is some sliding of the fibres, causing the rope to wear and chafe internally. In order to minimise this defect, the rope fibres are lubricated with a tar, tallow or graphite. The lubrication also makes the rope moisture proof. The hemp ropes are suitable only for hand operated hoisting machinery and as tie ropes for lifting tackle, hooks etc. The cotton ropes are very soft and smooth. The lubrication of cotton ropes is not necessary. But if it is done, it reduces the external wear between the rope and the grooves of its sheaves. It may be noted that the manila ropes are more durable and stronger than cotton ropes. The cotton ropes are costlier than manila ropes.

The fibre rope drives have the following advantages:

1. They give smooth, steady and quiet service.
2. They are little affected by outdoor conditions.
3. The shafts may be out of strict alignment.
4. The power may be taken off in any direction and in fractional parts of the whole amount.
5. They give high mechanical efficiency

Ratio of Driving Tensions for Fibre Rope,

The fibre ropes are designed in the similar way as V-belts, the Ratio is

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta$$

where μ, θ and β have usual meanings.

Wire Ropes: When a large amount of power is to be transmitted over long distances from one pulley to another (i.e. when the pulleys are upto 150 metres apart), then wire ropes are used. The wire ropes are widely used in elevators, mine hoists, cranes, conveyors, hauling devices and suspension bridges. The wire ropes run on grooved pulleys but they rest on the bottom of the *grooves and are not wedged between the sides of the grooves.

The wire ropes are made from cold drawn wires in order to have increase in strength and durability.

The wire ropes have the following advantages as compared to fibre ropes.

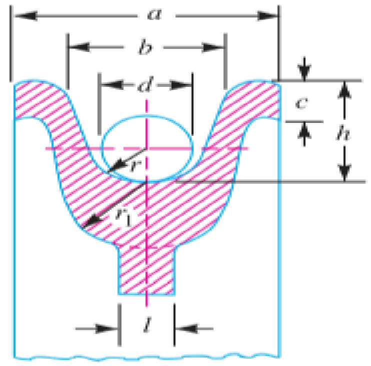
- ✓ These are lighter in weight,
- ✓ These offer silent operation,

- ✓ These can withstand shock loads
- ✓ These are more reliable
- ✓ These are more durable
- ✓ They do not fail suddenly
- ✓ The efficiency is high
- ✓ The cost is low.

Classification of Wire Ropes

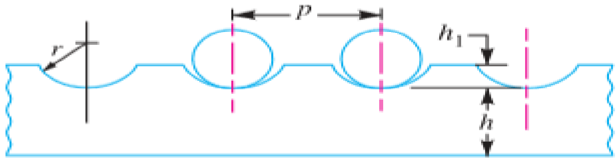
1. Cross or regular lay ropes
2. Parallel or lang lay ropes.
3. Composite or reverse laid ropes.

The sheave groove has a great influence on the life and service of the rope. If the groove is bigger than rope, there will not be sufficient support for the rope which may, therefore, flatten from its normal circular shape and increase fatigue effects. On the other hand, if the groove is too small, then the rope will be wedged into the groove and thus the normal rotation is prevented



$r = 0.53\ d;\ r_1 = 1.1\ d; a = 2.7\ d; b = 2.1\ d;$
$c = 0.4\ d; h = 1.6\ d; l = 0.75\ d$

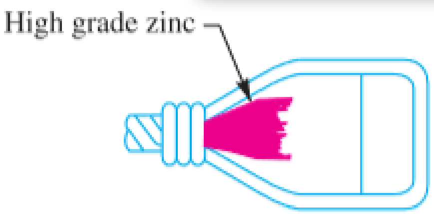
(*a*) Wire rope sheave rim.



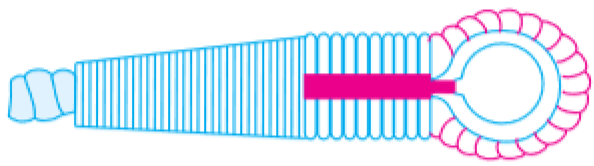
$p = 1.15\ d\ ; h_1 = 0.25\ d; r = 0.53\ d; h = 1.1\ d$

(*b*) Grooved rope drum.

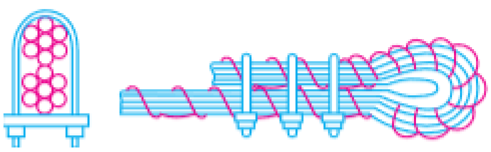
Wire Rope Fasteners

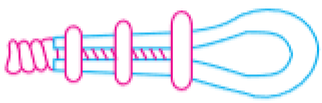


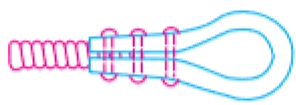
(*a*)



(*b*)







Direct stress due to axial load lifted and weight of the rope Let

1. W = Load lifted,
2. w = Weight of the rope, and
3. A = Net cross-sectional area of the rope

$$\sigma_d = \frac{W + w}{A}$$

Bending stress when the rope winds round the sheave or drum

$$\sigma_b = \frac{E_r \times d_w}{D}$$

$$W_b = \sigma_b \times A = \frac{E_r \times d_w \times A}{D}$$

E_r = Modulus of elasticity of the wire rope,
d_w = Diameter of the wire,
D = Diameter of the sheave or drum, and
A = Net cross-sectional area of the rope.

Stresses during starting and stopping.

$$W_a = \frac{W + w}{g} \times a$$

$$\sigma_a = \frac{W + w}{g} \times \frac{a}{A}$$

a = Acceleration of the rope and load, and g = Acceleration due to gravity.

a = v / 60 t

The impact load on starting may be obtained by the impact equation, i.e.

$$W_{st} = (W + w)\left[1 + \sqrt{1 + \frac{2a \times h \times E_r}{\sigma_d \times l \times g}}\right]$$

and velocity of the rope (v_r) at the instant when the rope is taut,

$$v_r = \sqrt{2a \times h}$$

where a = Acceleration of the rope and load,
h = Slackness in the rope, and
l = Length of the rope.

When there is no slackness in the rope, then $h = 0$ and $v_r = 0$, therefore

Impact load during starting,

$$W_{st} = 2\,(W + w)$$

and the corresponding stress,

$$\sigma_{st} = \frac{2\,(W + w)}{A}$$

Stress due to change in speed

$$a = (v_2 - v_1)\,/\,t$$

where $(v_2 - v_1)$ is the change in speed in m/s and t is the time in seconds.

Effective stress.

Effective stress in the rope during normal working

$$= \sigma_d + \sigma_b$$

Effective stress in the rope during starting

$$= \sigma_{st} + \sigma_b$$

and effective stress in the rope during acceleration of the load

$$= \sigma_d + \sigma_b + \sigma_a$$

A pulley used to transmit power by means of ropes has a diameter of 3.6 metres and has 15 grooves of 45° angle. The angle of contact is 170° and the coefficient of friction between the ropes and the groove sides is 0.28. The maximum possible tension in the ropes is 960 N and the mass of the rope is 1.5 kg per metre length. Determine the speed of the pulley in r.p.m. and the power transmitted if the condition of maximum power prevail.

Solution.

Given : d = 3.6 m ; n = 15 ; 2 β = 45° or β = 22.5° ; θ = 170° = 170 × π/ 180 = 2.967 rad ; μ = 0.28 ; T = 960 N ; m = 1.5 kg / m

N = Speed of the pulley in r.p.m.

We know that for maximum power, speed of the pulley

$$v = \sqrt{\frac{T}{3m}} = \sqrt{\frac{960}{3 \times 1.5}} = 14.6 \text{ m/s}$$

We also know that speed of the pulley (v),

$$14.6 = \frac{\pi d \cdot N}{60} = \frac{\pi \times 3.6 \times N}{60} = 0.19\ N$$

$$\therefore \quad N = 14.6 / 0.19 = 76.8 \text{ r.p.m. } \textbf{Ans.}$$

Power transmitted

We know that for maximum power, centrifugal tension,

$$T_C = T/3 = 960/3 = 320 \text{ N}$$

∴ Tension in the tight side of the rope,

$$T_1 = T - T_C = 960 - 320 = 640 \text{ N}$$

Let T_2 = Tension in the slack side of the rope.

We know that

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta = 0.28 \times 2.967 \times \operatorname{cosec} 22.5° = 2.17$$

$$\therefore \quad \log\left(\frac{T_1}{T_2}\right) = \frac{2.17}{2.3} = 0.9435 \quad \text{or} \quad \frac{T_1}{T_2} = 8.78 \qquad \text{...(Taking antilog of 0.9435)}$$

and

$$T_2 = T_1 / 8.78 = 640 / 8.78 = 73 \text{ N}$$

∴ Power transmitted,

$$P = (T_1 - T_2)\, v \times n = (640 - 73)\, 14.6 \times 15 = 124\,173 \text{ W}$$

$$= 124.173 \text{ kW } \textbf{Ans.}$$

5.Linkages

Linkages are mechanical structures with revolute, linear and cam joints for transmitting movements or for guiding points or planes.
Linkages (couplings) are often combined with cam gears, servo drives or gearboxes. They transfer the movement from a cam to the possibly distant working tool and thus to the point of action. At the same time, they also translate the cam stroke.
In machines, four-bar linkages are often used. Servo drives are often combined with slider-cranks to create an oscillating stroke from the input rotation of the servo drive or gearbox output.
In mechanical presses, sometimes multi-bar linkages are used to achieve a press ram motion that is favorable for the forming process.
But you also know some examples of linkages from everyday life:

- ✓ Mechanics of umbrellas
- ✓ Harbor cranes
- ✓ Excavator
- ✓ Hinges in kitchen cabinets
- ✓ Windshield wipers
- ✓ Toy pantograph
- ✓ Mechanics on the wheels of a steam locomotive
- ✓ Mechanics on cabriolet covers
- ✓ Loppers
- ✓ Lifters

Four-bar linkages

Probably the most well-known and most studied coupling mechanism is the flat four-bar linkage, a coupling gear with four bars (parts) exclusively with revolute joints. The frame counts as a bar

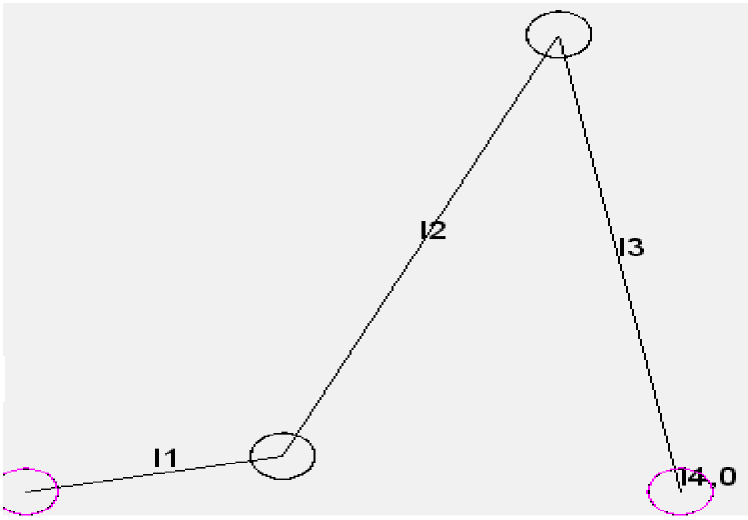


Figure 68.Four-bar linkages

Four-bar linkages are used to **transmit motions** and to **guide points and planes**.
Their simplicity is based on the principle: the less moving parts, the better.
Four-bar linkages often serve as the basis for more complex mechanism structures.
At the time of the graphical transmission analysis, four-bar linkages were often used as replacement to study, for example, characteristics of cam gears.

The First Mechanism: The Lever is a 2-bar Linkage

•A lever (link) can be used with a fulcrum (pivot) against the ground (link) to allow a small force moving over a large distance to create a large force moving over a short distance… –When one considers the means to input power, a lever technically becomes a 4-bar linkage

•The forces are applied through pivots, and thus they may not be perpendicular to the lever – Torques about the fulcrum are thus the best way to determine equilibrium, and torques are best calculated with vector cross product

. A link is a nominally rigid body that possess at least 2 nodes.

A node is an attachment point to other links via joints. The order of a link indicates the number of joints to which the link is connected (or the number of nodes per link). There are binary (2 nodes), ternary (3 nodes), and quaternary (4 nodes) links. A joint is a connection between two or more links at their nodes, which allows motion to occur between the links.

There are many different processes for designing linkages. Synthesis is the process used to create a linkage. Number synthesis is the determination of the number and order of links needed to produce desired motion. Kinematic synthesis is the determination of the size and configuration of links needed to produce desired motion. In either method, precision points are the defined desired position and orientations of a link at a point in its motion.

The simplest linkage with at least one degree of freedom (motion) is thus a 4-bar linkage!

A 3-bar linkage will be rigid, stable, not moving unless you bend it, break it, or throw it!

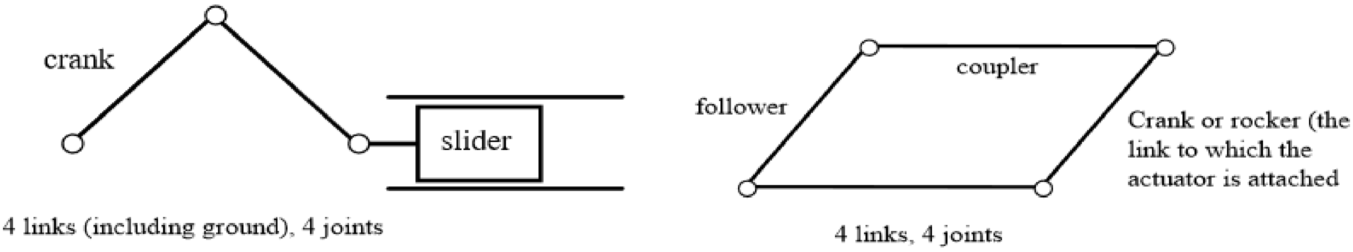


Figure 69.link joint

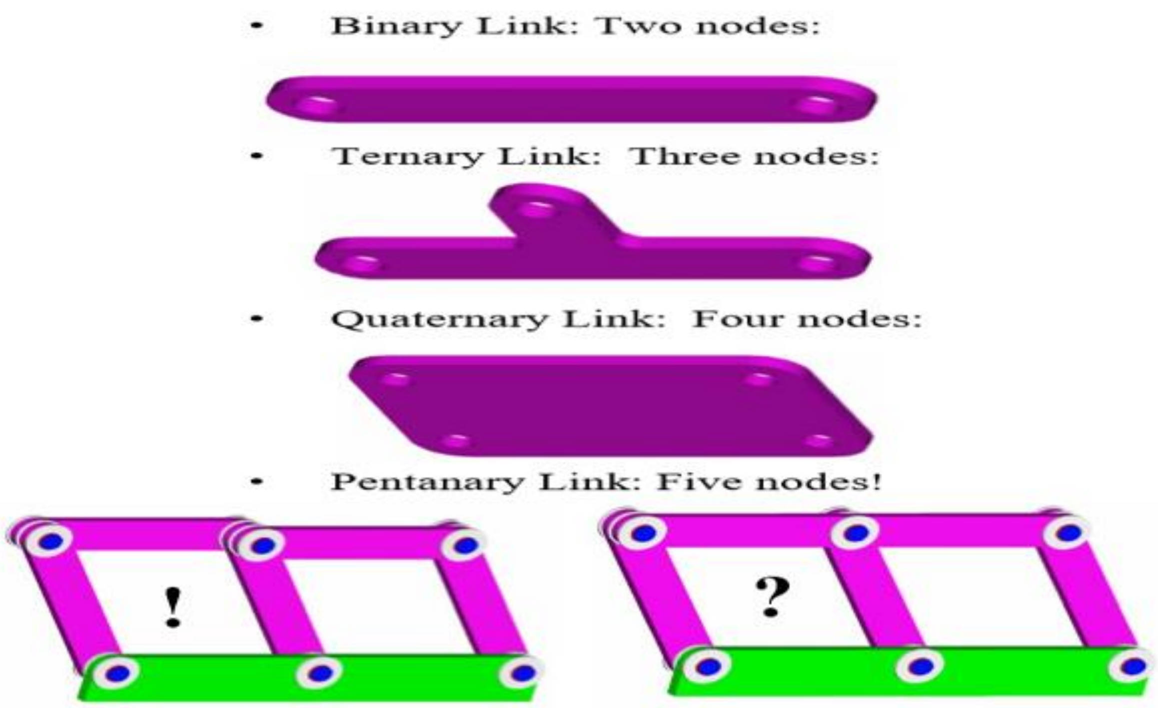


Figure 70.nodes per link

6. Cam and follower systems

A cam is a rotating or sliding piece in a mechanical linkage that drives a mating component known as a follower. From a functional viewpoint, a cam-and-follower arrangement is very similar to the linkages. The cam accepts an input motion (rotary motion or linear motion) and imparts a resultant motion (linear motion or rotary motion) to a follower.

Cam Nomenclature

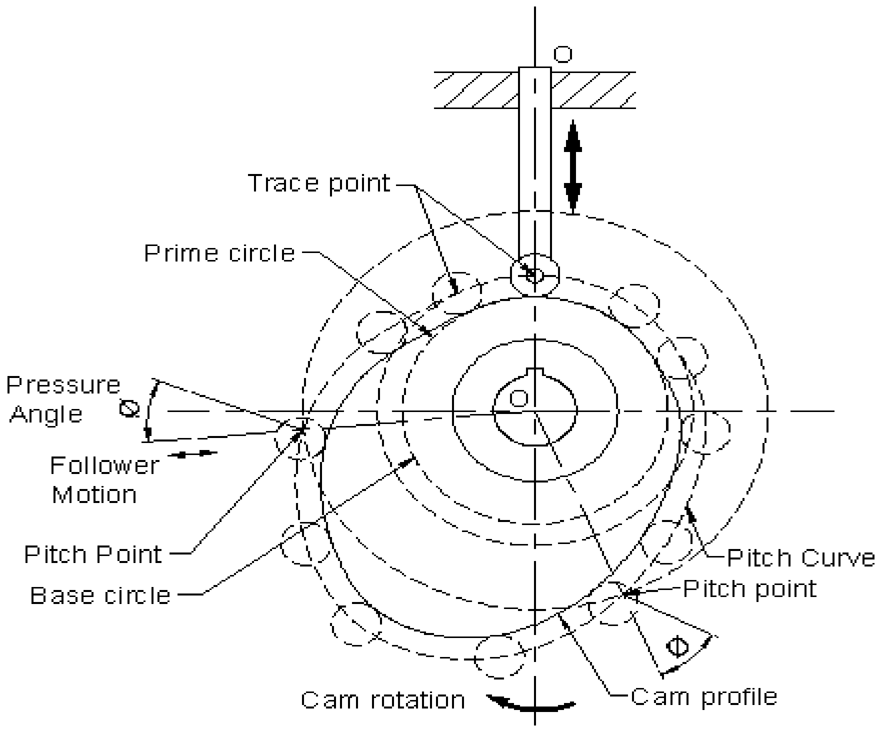


Figure 71.CAM nomenclature

1. ***Cam profile:*** Cam profile is outer surface of the disc cam.
2. ***Base circle:*** Base circle is the smallest circle, drawn tangential to the cam profile.
3. **Trace point*:*** Trace point is a point on the follower, trace point motion describes the movement of the follower.
4. **Pitch curve:** Pitch curve is the path generated by the trace point as the follower is rotated about a stationery cam.
5. **Prime circle:** Prime circle is the smallest circle that can be drawn so as to be tangential to the pitch curve, with its centre at the cam centre.
6. **Pressure angle:** The pressure angle is the angle between the direction of the follower movement and the normal to the pitch curve.
7. **Pitch point:** Pitch point corresponds to the point of maximum pressure angle.
8. **Pitch circle:** A circle drawn from the cam center and passes through the pitch point is called Pitch circle.
9. **Stroke:** The greatest distance or angle through which the follower moves or rotates.

Cams can be classified into the following three types based on their shapes.

Figure 72.Various custom cams

1. **Plate or disk cams:** Plate or disk cams are the simplest and most common type of cam. A plate cam is illustrated in figure 3 (a). This type of cam is formed on a disk or plate. The radial distance from the center of the disk is varied throughout the circumference of the cam. Allowing a follower to ride on this outer edge gives the follower a radial motion.
2. **. Cylindrical or drum cam:** A cylindrical or drum cam is illustrated in figure 3 **(b). This type of cam** is formed on a cylinder. A groove is cut into the cylinder, with a varying location along the axis of rotation. Attaching a follower that rides in the groove gives the follower motion along the axis of rotation.
3. **Linear cam:** A *linear cam* is illustrated in figure 3 (c). This type of cam is formed on a translated block.

A groove is cut into the block with a distance that varies from the plane of translation. Attaching a follower that rides in the groove gives the follower motion perpendicular to the plane of translation.

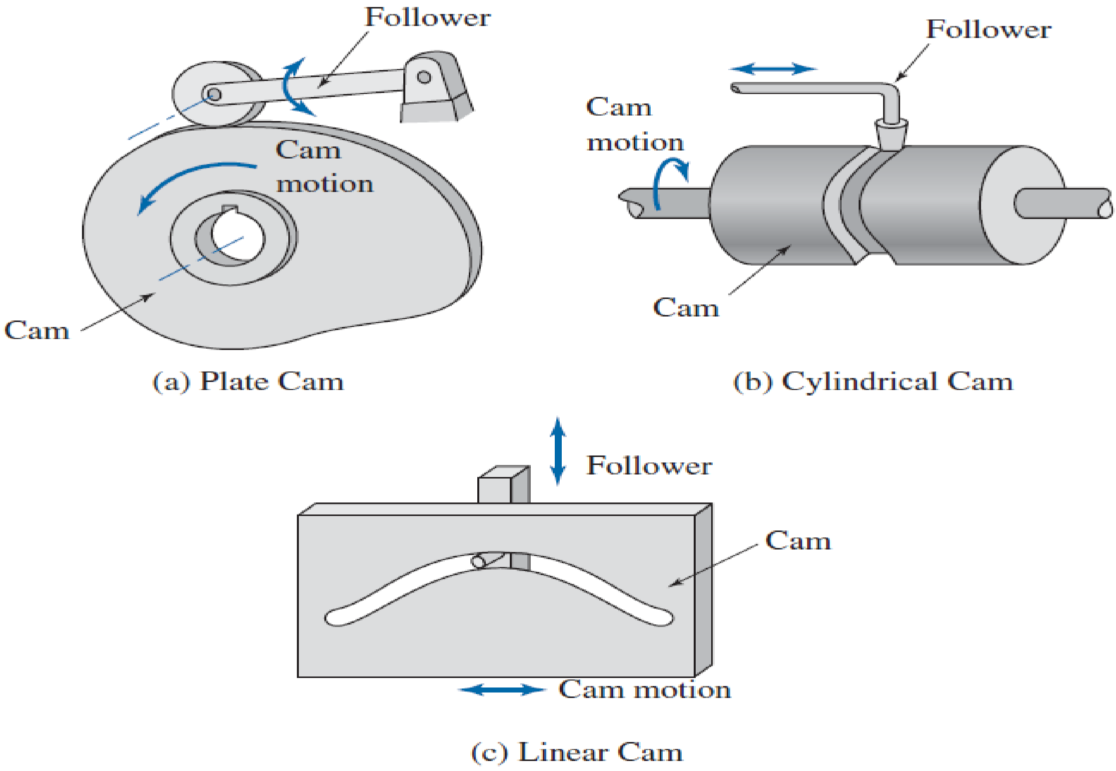


Figure 73.Cam types

Types of Followers

Followers are classified based on their motion, position and shape. The details of follower classifications are shown in the figure 4 and discussed below

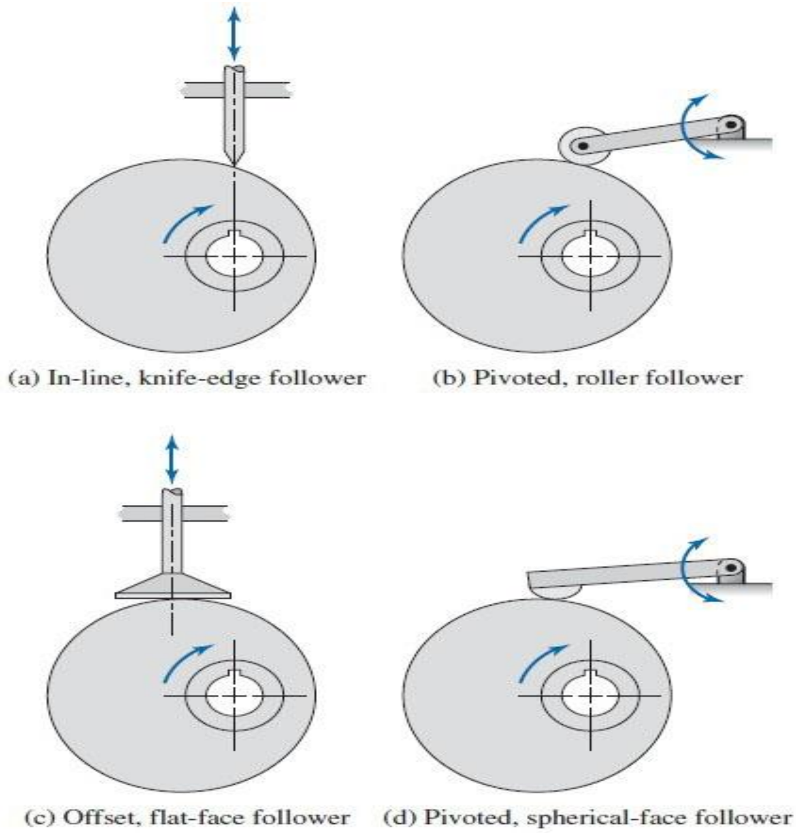


Figure 74.follower types

1. Based on Follower Motion

Based on the follower motion, followers can be classified into the following two categories:

Translating followers are constrained to motion in a straight line *followers* are constrained to rotational motion and *Swinging arm or pivoted* are shown in figure 74 (b) and 4 (d).

1. **Based on Follower Position**

Based on the follower position, relative to the center of rotation of the cam, is typically influenced by any spacing requirements of the machine.

The position of translating followers can be classified into the following two categories:

(i). An *in-line follower* exhibits straight-line motion, such that the line of translation extends through the center of rotation of the cam and is shown in figure 74(a).

(ii). An *offset follower* exhibits straight-line motion, such that the line of the motion is offset from the center of rotation of the cam and is shown in figure 74 (c).

In the case of pivoted followers, there is no need to distinguish between in-line and offset followers because they exhibit identical kinematics.

2. **Based on Follower Shape**

Finally, the follower shape can be classified into the following four categories:

(i). A *knife-edge follower* consists of a follower that is formed to a point and drags on the edge of the cam. The follower shown in figure 4 (a) is a knife-edge follower. It is the simplest form, but the sharp edge produces high contact stresses and wears rapidly. Consequently, this type of follower is rarely used.

(ii). A *roller follower* consists of a follower that has a separate part, the roller that is pinned to the follower stem. The follower shown in figure 4 (b) is a roller follower. As the cam rotates, the roller maintains contact with the cam and rolls on the cam surface. This is the most commonly used follower, as the friction and contact stresses are lower than those for the knife-edge follower. However, a roller follower can possibly jam during steep cam displacements.

(iii). A *flat-faced follower* consists of a follower that is formed with a large, flat surface available to contact the cam. The follower shown in figure 4 (c) is a flat-faced follower. This type of follower can be used with a steep cam motion and does not jam. Consequently, this type of follower is used when quick motions are required. However, any follower deflection or misalignment causes high surface stresses. In addition, the frictional forces are greater than those of the roller follower because of the intense sliding contact between the cam and follower.

(iv). A *spherical-faced follower* consists of a follower formed with a radius face that contacts the cam. The follower shown in figure 61 (d) is a spherical-face follower. As with the flat-faced follower, the spherical- face can be used with a steep cam motion without jamming. The radius face compensates for deflection or misalignment. Yet, like the flat-faced follower, the frictional forces are greater than those of the roller follower.

Displacement, velocity and acceleration diagrams when the follower moves with uniform velocity

The displacement, velocity and acceleration diagrams when a knife-edged follower moves with uniform velocity (a), (b) and (c) respectively. The abscissa (base) represents the time (i.e. the number of seconds required for the cam to complete one revolution) or it may represent the angular displacement of the cam in degrees. The ordinate represents the displacement, or velocity or acceleration of the follower. Since the follower moves with uniform velocity during its rise and return stroke, therefore the slope of the displacement curves must be constant. In other words, AB1 and C1D must be straight lines. A little consideration will show that the follower remains at rest during part of the cam rotation. The periods during which the follower remains at rest are known as dwell periods, as shown by lines B1C1 and DE.(c), we see that the acceleration or retardation of the follower at the beginning and at the end of each stroke is infinite. This is due to the fact that the follower is required to start from rest and has to gain a velocity within no time. This is only possible if the acceleration or retardation at the beginning and at the end of each stroke is infinite. These conditions are however, impracticable.

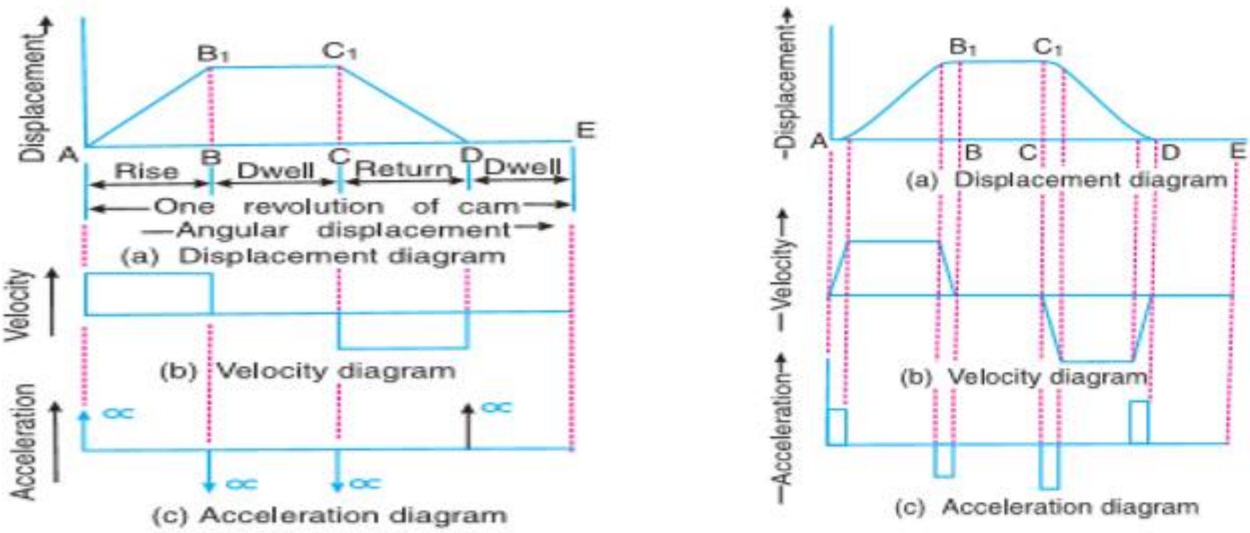


Figure 75.Displacement, velocity and acceleration diagrams when the follower moves with uniform velocity.

In order to have the acceleration and retardation within the finite limits, it is necessary to modify the conditions which govern the motion of the follower. This may be done by rounding off the sharp corners of the displacement diagram at the beginning and at the end of each stroke, (a). By doing so, the velocity of the follower increases gradually to its maximum value at the beginning of each stroke and decreases gradually to zero at the end of each stroke. (b). The modified displacement, velocity and acceleration diagrams. The round corners of the displacement diagram are usually parabolic curves because the parabolic motion results in a very low acceleration of the follower for a given stroke and cam speed.

7. BRAKES

A brake is defined as a mechanical device, which is used to absorb the energy possessed by a moving system or mechanism by means of friction. The primary purpose of the brake is to slow down or completely stop the motion of a moving system, such as a rotating drum, machine or vehicle. It is also used to hold the parts of the system in position at rest. An automobile brake is used either to reduce the speed of the car or to bring it to rest. It is also used to keep the car stationary on the downhill road.

Brakes are classified into the following three groups:

***1*.Mechanical brakes**, which are operated by mechanical means such as levers, springs and pedals. Depending upon the shape of the friction material, the mechanical brakes are classified as block brakes, internal or external shoes brakes, disk brakes and band brakes. Brakes are also classified into two groups according to the direction of the actuating force, namely, radial brakes and axial brakes. Internal and external shoe brakes are radial brakes, while disk brakes are axial brakes. The discussion in this chapter is restricted to mechanical brakes.

2.Hydraulic and pneumatic brakes, which are operated by fluid pressure such as oil pressure or air pressure.

3.Electrical brakes, which are operated by magnetic forces and which include magnetic particle brakes, hysteresis brakes and eddy current brakes.

Brake capacity depends upon the following factors:

(i) The unit pressure between braking surfaces

(ii) The contacting area of braking surface

(iii) The radius of the brake drum

(iv) The coefficient of friction

(v) The ability of the brake to dissipate heat that is equivalent to the energy being absorbed

ENERGY EQUATIONS

The braking-torque depends upon the amount of energy absorbed by the brake. When a mechanical system of mass *m* moving with a velocity v_1 is slowed down to the velocity v_2 during the period of braking, the kinetic energy absorbed by the brake is given by

Displacement, velocity and acceleration diagrams when the follower moves with uniform velocity.

$$KE = \frac{1}{2} m(v_1^2 - v_2^2)$$

where

KE = kinetic energy absorbed by the brake (J)

m = mass of the system (kg)

v_1, v_2 = initial and fi nal velocities of the system (m/s)

Similarly, the kinetic energy of the rotating body is given by

$$KE = \frac{1}{2} I(\omega_1^2 - \omega_2^2)$$

$$KE = \frac{1}{2} mk^2(\omega_1^2 - \omega_2^2)$$

where,

I = mass moment of inertia of the rotating body (kg-m^2)

k = radius of gyration of the body (m) of the body (rad/s)

the brake absorbs the potential energy released by the moving weight during the braking period. When a body of

mass *m* falls through a distance *h*, the potential energy absorbed by the brake during the braking period is given by

$$PE = mgh$$

the brake absorbs the potential energy released by the moving weight during the braking period. When a body of

mass *m* falls through a distance *h*, the potential energy absorbed by the brake during the braking period is given by

$E = M_t\theta$

E = total energy absorbed by the brake (J)

M_t= braking-torque (N-m)

⍰= angle through which the brake drum rotates during the braking period (rad)

Example question

A solid cast iron disk, 1 m in diameter and 0.2 m thick, is used as a flywheel. It is rotating at 350 rpm. It is brought to rest in 1.5 s by means of a brake. Calculate

(ii) the energy absorbed by the brake; and

(iii) (ii) the torque capacity of the brake.

Solution

Given $D = 1$ m $t = 0.2$ m $n_1 = 350$ rpm

$t = 1.5$ s

Step I*Energy absorbed by brake*

The brake absorbs the kinetic energy of the rotating flywheel. The mass density of cast iron is taken as 7200 kg/m^3. The radius of gyration of a solid disk about its axis of rotation is $(d/\sqrt{8})$. (Therefore,

$$m = \frac{\pi}{4}(1)^2(0.2)(7200) = 1130.97 \text{ kg}$$

$$k^2 = \frac{D^2}{8} = \frac{1}{8} \text{ m}^2$$

$$\omega_1 = \frac{2\pi n_1}{60} = \frac{2\pi(350)}{60} = 36.65 \text{ rad/s}$$

and $\omega_2 = 0$

$$E = \frac{1}{2} mk^2(\omega_1^2 - \omega_2^2)$$

$$= \frac{1}{2}(1130.97)\left(\frac{1}{8}\right)(36.65)^2 = 94\,946.52 \text{ J (i)}$$

Step II *Torque capacity of brake*

The average velocity during the braking period is

(⍰$_1$ + ⍰$_2$)/2 or (⍰$_1$/2). Therefore,

$$\theta = \left(\frac{\omega_1}{2}\right)t = \left(\frac{36.65}{2}\right)(1.5) = 27.49 \text{ rad}$$

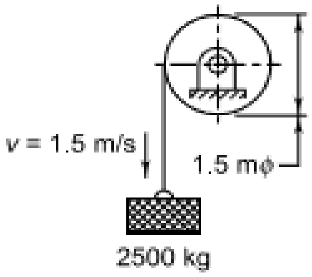
$$M_t = \frac{E}{\theta} = \frac{94\,946.52}{27.49} = 3453.86 \text{ N-m}$$

Qwestion two

A mass of 2500 kg is lowered ata velocity of 1.5 m/s from the drum as shown in Fig.The mass of the drum is 50 kg and its radius of gyration can be taken as 0.7 m. On applying the brake, the mass is brought to rest in a distance of 0.5 m. Calculate

(ii) the energy absorbed by the brake; and

(iii) *(ii) the torque capacity of the brake.*



Solution

Given $m = 2500$ kg $v = 1.5$ m/s
For drum, $m = 50$ kg $k = 0.7$ m $h = 0.5$ m
$R = 0.75$ m

Step I Energy absorbed by brake

KE of the mass

$$KE = \frac{1}{2}m(v_1^2 - v_2^2) = \frac{1}{2}(2500)(1.5)^2 = 2812.5 \text{ J}$$

KE of the drum

$$\omega_1 = \frac{v_1}{R} = \frac{1.5}{0.75} = 2 \text{ rad/s}$$

$$KE = \frac{1}{2}mk^2(\omega_1^2 - \omega_2^2) = \frac{1}{2}(50)(0.7)^2(2)^2 = 49 \text{ J}$$

PE of the mass

$$PE = mgh = (2500)(9.81)(0.5) = 12\,262.5 \text{ J}$$

$$E = 2812.5 + 49 + 12\,262.5 = 15\,124 \text{ J} \qquad \text{(i)}$$

Step II*Torque capacity of brake*

During the braking action, the mass moves through a distance of 0.5 m. If is the angle through which the drum rotates during the braking period,

 × drum radius = 0.5

or $\theta = \frac{0.5}{0.75} = 0.667$ rad

$\therefore \quad M_t = \frac{E}{\theta} = \frac{15124}{0.667} = 22\,686$ N-m

BLOCK BRAKE WITH SHORT SHOE

A block brake consists of a simple block, which is pressed against the rotating drum by means of a lever as shown in Fig. The friction between the block and the brake drum causes the retardation of the drum. This type of brake is commonly employed in railway trains. The block is either rigidly attached to the lever or, in some applications, pivoted to the lever. The angle of contact between the block and the brake drum is usually small. When it is less than 45°, the intensity of pressure between the block and brake drum is uniform.

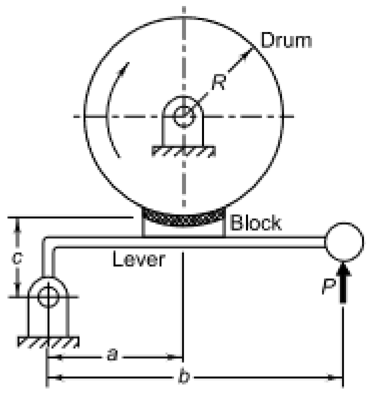


Figure 76.BLOCK BRAKE

The free-body diagram of forces acting on the drum and the lever is shown in Fig. The analysis is based on the following assumptions:

(i) The block is rigidly attached to the lever.

(ii) The angle of contact between the block and brake drum is small, resulting in uniform pressure distribution.

(iii) The brake drum is rotating in clockwise direction.

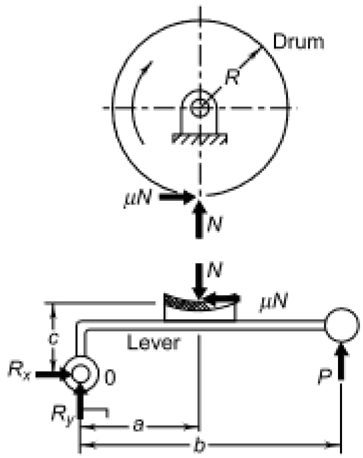


Figure 77.The free-body diagram (clocke wise rotation)

Considering the forces acting on the brake drum,

$M_t = NR$

where

M_t= braking torque (N-mm)

R = radius of the brake drum (mm)

N = normal reaction (N)

The dimensions of the block are determined by the following expression,

$N = plw$

p = permissible pressure between the block and the brake drum (N/mm^2)

l = length of the block (mm) w = width of the block (mm)

Considering the equilibrium of forces in vertical and horizontal directions,

$R_x = \mu N$

$R_y = (N - P)$

Taking moment of forces acting on the lever about the hinge point O,

$$P \times b - N \times a + \mu N \times c = 0$$

$$\text{or} \quad P = \frac{(a - \mu c)}{b} \times N$$

The above analysis was based on three assumptions and the third assumption was that the brake drum is rotating in a clockwise direction. Let us consider the free-body diagram of forces when the brake drum rotates in an anti-clockwise direction\Taking moment of forces acting on the lever about the hinge point$O$,

$$P \times b - N \times a - \mu N \times c = 0$$

$$\text{or} \quad P = \frac{(a + \mu c)}{b} \times N$$

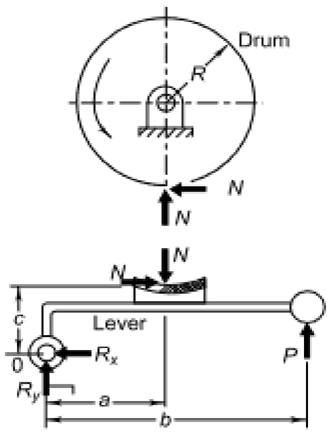


Figure 78.free-body diagram of forces (anti-clockwise direction)

Therefore, the braking effort (P) also depends upon the direction of rotation of the brake drum. Obviously, for anti-clockwise rotation, the actuating force or braking effort (P) is more than that of clockwise rotation, In design, the objective will be smaller values of braking effort.

The main disadvantage of the block brake is the tendency of the brake drum shaft to bend under the action of normal reaction. The remedy is to use two symmetrical blocks at the opposite sides of the brake drum.

- **BLOCK BRAKE WITH LONG SHOE**

In the previous section, a block brake with short shoe was discussed. The angle of contact between the block and brake drum in such cases is usually small and less than 45°. It is, therefore, reasonable to assume that the normal reaction (N) and frictional force (□N) are concentrated at the midpoint of the shoe. This assumption is not applicable for the brake with the long shoe. A block brake with long shoe is shown in Fig.

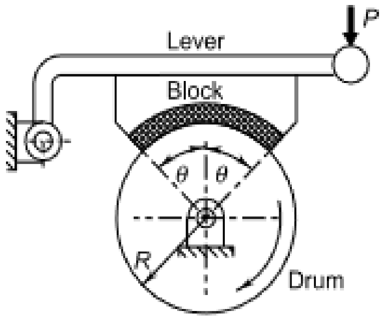


Figure 79.BLOCK BRAKE WITH LONG SHOE

The force of friction on the elementary area is given by,

$\mu dN = \mu(Rdw)p$

The torque transmitted by the force of friction on the elementary area is given by,

$M_t = \mu dNR = (R^2 dw)p$

$M_t = \mu R^2 w\ pd$

❖ PIVOTED BLOCK BRAKE WITH LONG SHOE

When the block is rigidly fixed to the lever, the tendency of the frictional force (*N*) is to unseat the block with respect to the lever as shown in Fig. 69 In case of the pivoted shoe brake, the location of the pivot can be selected in such a way that the moment of frictional force about the pivot is zero. This is the main advantage of the pivoted shoe brake. A double block brake with two symmetrical and pivoted shoes is shown in Fig.80

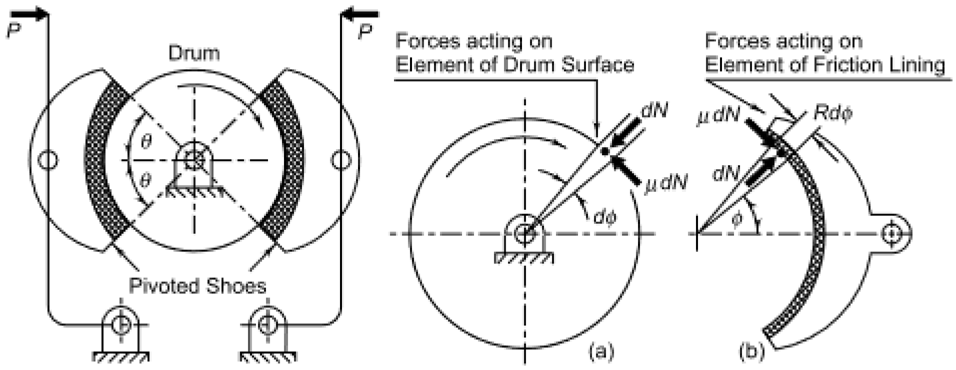


Figure 80.BLOCK BRAKE

❖ BAND BRAKES

The construction of a simple band brake is shown in Fig.81. It consists of a flexible steel strip lined with friction material, which is pressed against the rotating brake drum. When one end of the steel band passes through the fulcrum of the actuating lever, the brake is called the simple band brake.

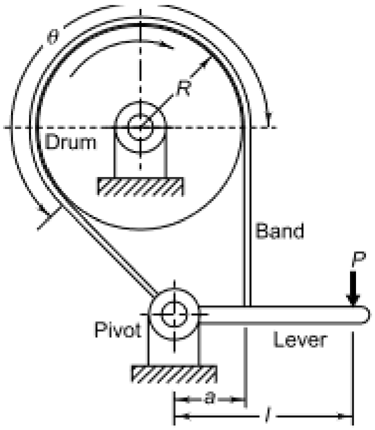


Figure 81.simple band brake.

The working of the steel band is similar to that of a fl at belt operating at zero velocity. The free-body diagram of forces acting on the band and the lever.

$$\frac{P_1}{P_2} = e^{\mu\theta}$$

where,

P_1 = tension on the tight side of the band (N)

P_2 = tension on the loose side of the band (N)

The torque M_t absorbed by the brake is given by,

$M_t = (P_1 - P_2)R$ (12.25) where,

M_t= torque capacity of the brake (N-mm)

R = radius of the brake drum (mm)

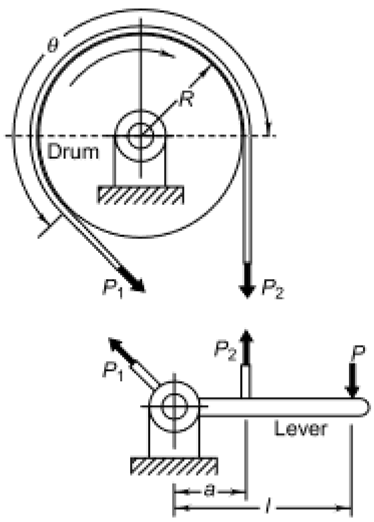


Figure 82.Free-body Diagram of Forces

Considering the forces acting on the lever and taking moments about the pivot,

$$P_2 \times a = P \times l$$

$$P = \frac{P_2 a}{l}$$

Band brake offers the following advantages:

(i) Band brake has simple construction. It has small number of parts. These features reduce the cost of band brake.
(ii) Most equipment manufacturers can easily produce band brake without requiring specialized facilities like foundry or forging shop. The friction lining is the only part which must be purchased from outside agencies.
(iii) Band brake is more reliable due to small number of parts.
(iv) Band brake requires little maintenance.

The disadvantages of band brake are as follows:

(i) The heat dissipation capacity of a band brake is poor.
(ii) The wear of friction lining is uneven from one end to the other.

Band brakes are used in applications like bucket conveyors, hoists and chain saws. They are more popular as back-stop devices.

❖ DISK BRAKES

A disk brake is similar to a plate clutch, except that one of the shafts is replaced by a fixed member. Disk brakes can be observed on the front wheel of most motorcycles. A bicycle brake is another example of a disk brake. In this case, the wheel rim constitutes the disk. The friction lining on the caliper contacts only a small portion of the rim, leaving the remaining portion to dissipate the heat to the surrounding.

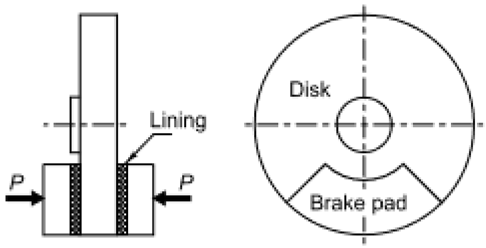


Figure 83.DISK BRAKES

Disk brakes have the following advantages:

(i) Disk brake is simple to install and service.
(ii) Disk brake has high torque transmitting capacity in small volume.
(iii) In drum brakes, as the temperature increases, the coefficient of friction decreases. Due to inherent abilityof the disk brake to dissipate heat, it is insensitive to changes in the coefficient of friction. The disk brake is easy to control.
(iv) The brake can never become self-locking.
(v) The brake is equally effective for both directions of rotation of the disk.

The disk brake has ‘linearity’, that is, the braking torque is linearly proportional to the actuating force

8. Clutch

The clutch is a mechanical device, which is used to connect or disconnect the source of power from the remaining parts of the power transmission system at the will of the operator. An automotive clutch can permit the engine to run without driving the car. This is desirable when the engine is to be started or stopped, or when the gears are to be shifted.

Very often, three terms are used together, namely, couplings, clutches and brakes. There is a basic difference between the coupling and the clutch. A coupling, such as a flange coupling, is a permanent connection. The driving and driven shafts are permanently attached by means of coupling and it is not possible to disconnect the shafts, unless the coupling is dismantled. On the other hand, the clutch can connect or disconnect the driving and driven shafts, as and when required by the operator. Similarly, there is a basic difference between initial and final conditions in clutch and brake operations.

In the operation of clutch, the conditions are as follows:

1.Initial Condition The driving member is rotating and the driven member is at rest.

2.Final Condition Both members rotate at the same speed and have no relative motion.

Clutches are classified into the following four groups:

1.Positive contact Clutches They include square jaw clutches; spiral jaw clutches and toothed clutches. In these clutches, power transmission is achieved by means of interlocking of jaws or teeth. Their main advantage is positive engagement and once coupled, they can transmit large torque with no slip.

2.Friction Clutches They include single and multi-plate clutches, cone clutches and centrifugal clutches. In these clutches, power transmission is achieved by means of friction between contacting surfaces. This chapter is restricted to friction clutches.

3.Electromagnetic Clutches They include magnetic particle clutches, magnetic hysteresis clutches and eddy current clutches. In these clutches, power transmission is achieved by means of the magnetic field. These clutches have many advantages, such as rapid response time, ease of control, and smooth starts and stops

4.Fluid Clutches and Couplings In these clutches, power transmission is achieved by means of hydraulic pressure. A fluid coupling provides extremely smooth starts and absorbs shock.

The simplest form of positive contact clutches is the square jaw clutch as shown in Fig. 11 It consists of two halves carrying projections or jaws. There are two types of jaws, namely, square and spiral. The spiral jaws can be engaged at slightly higher speed without clashing. Frequent engagement results in wear of jaws.

The jaw clutches have the following advantages:

(a) They do not slip and engagement is positive.

(b) No heat is generated during engagement or disengagement.

The jaw clutches have the following drawbacks:

(a) Jaw clutches can be engaged only when both shafts are stationary or rotate with very small speed difference.

(b) They cannot be engaged at high speeds because engagement of jaws and sockets results in shock.

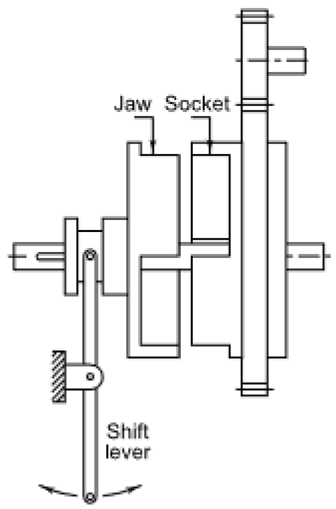


Figure 84.Square jaw clutch

In general, positive contact clutches are rarely used as compared with friction clutches. However, they have some important applications where synchronous operation is required like power presses and rolling mills.

A single plate friction clutch consisting of two flanges is shown in Fig.73. One flange is rigidly keyed to the driving shaft, while the other is connected to the driven shaft by means of splines.

The splines permit free axial movement of the driven flange with respect to the driven shaft. power is transmitted from the driven flange to the driven shaft by means of the splines. Since the power is transmitted by means of frictional force between the driving and driven flanges, the clutch is called '*friction*' clutch.

In order to disengage the clutch, a fork is inserted in the collar on the driven flange to shift it axially to the right side. This relieves the pressure between the driving and the driven flanges and no torque can be transmitted. In the working condition, the clutch is in an engaged position under the action of spring force. Levers or forks are operated to '*disengage*' the clutch.

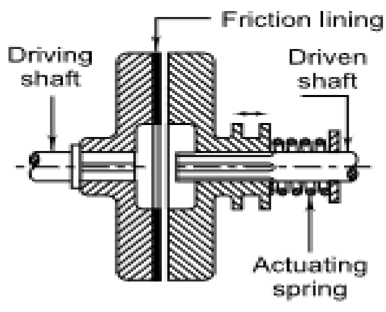


Figure 85.single plate clucth

The main advantages of friction clutch are as follows:

(i) The engagement is smooth.

(ii) Slip occurs only during engaging operation and once the clutch is engaged, there is no slip between the contacting surfaces. Therefore, power loss and consequent heat generation do not create problems, unless the operation requires frequent starts and stops.

(iii) In certain cases, the friction clutch serves as a safety device. It slips when the torque transmitted through it exceeds a safe value. This prevents the breakage of parts in the transmission chain.

Depending upon the number of friction surfaces, the friction clutches are classified as single-plate or multi-plate clutches. Depending upon the shape of the friction material, the clutches are classified as disk clutches, cone clutches or expanding shoe clutches.

The following factors should be considered while designing friction clutches:

(i) Selection of a proper type of clutch that is suitable for the given application

(ii) Selection of suitable friction material at the contacting surfaces

(iii) Designing the clutch for sufficient torque capacity

(iv) Engagement and disengagement should be without shock or jerk

(v) Provision for holding the contacting surfaces together by the clutch itself and without any external assistance

(vi) Low weight for rotating parts to reduce inertia forces, particularly in high-speed applications

(vii) Provision for taking or compensating wear of rubbing surfaces

(viii) Provision for carrying away the heat generated at the rubbing surfaces

In design of clutches, the following factors should be considered:

(iv) Service Factor

(v) Location of Clutch

(vi) The coefficient of friction

MULTI-DISK CLUTCHES

A multi-disk clutch, as shown in Fig. consists of two sets of disks—*A* and *B*. Disks of Set *A* are usually made of hardened steel, while those of Set *B* are made of bronze. Disks of Set *A* are connected to the driven shaft by means of splines. Because of splines, they are free to move in an axial direction on the splined sleeve.

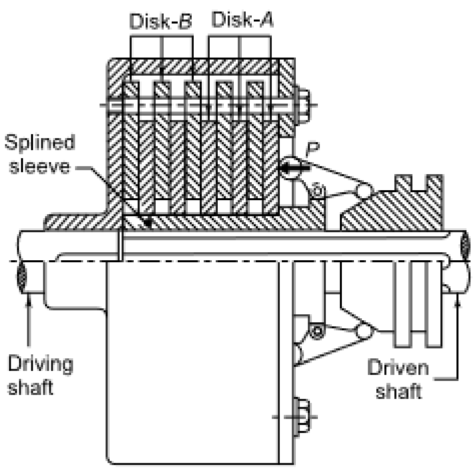


Figure 86.MULTI-DISK CLUTCHES

The difference between single and multi-plate clutches is as follows:

(i) The number of pairs of contacting surfaces in the single plate clutch is one or at the most, two. There are more number of contacting surfaces in the multi-disk clutch.

(ii) As the number of contacting surfaces is increased, the torque transmitting capacity is also increased, other conditions being equal. In other words, for a given torque capacity, the size of the multi-plate clutch is smaller than that of the single plate clutch, resulting in compact construction.

(iii) The work done by friction force during engagement is converted into heat. More heat is generated in the multi-plate clutch due to increased number of contacting surfaces

(iv) The coefficient of friction decreases due to cooling oil, thereby reducing the torque transmitting capacity of the multi-plate clutch. The coefficient of friction is high in dry single plate clutches.

(v) Single plate clutches are used in applications where large radial space is available, such as trucks and cars. Multi-disk clutches are used in applications where compact construction is desirable, e.g., scooter and motorcycle.

CONE CLUTCHES

A cone clutch, as shown in Fig. 7consists of inner and outer conical surfaces. The outer cone is keyed to the driving shaft, while the inner cone is free to slide axially on the driven shaft due to splines. The axial force required to engage the clutch is provided by means of helical compression spring. In engaged position, power is transmitted from the driving shaft to the outer cone by means of the key. Power is then transmitted from the outer cone to the inner cone by means of friction. Finally, power is transmitted from the inner cone to the driven shaft by means of the splines.

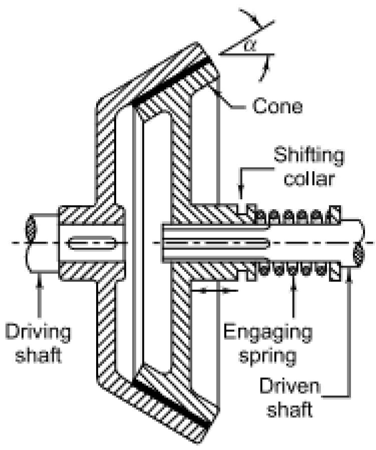


Figure 87.CONE CLUTCHES

CENTRIFUGAL CLUTCHES

Whenever it is required to engage the load after the driving member has attained a particular speed, a centrifugal clutch is used. The centrifugal clutches permit the drive-motor or engine to start, warm up and accelerate to the operating speed without load. Then the clutch is automatically engaged and the driven equipment is smoothly brought up to the operating speed. These clutches are particularly useful with internal combustion engines, which cannot start under load.

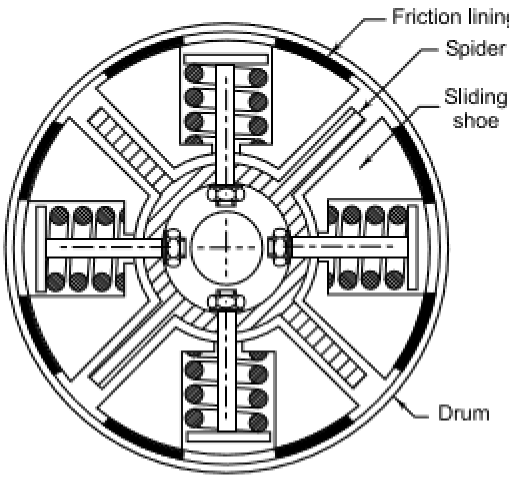


Figure 88.CENTRIFUGAL CLUTCHES

9. Friction wheels

Friction wheels are cylinders or cone wheels in contact that transmit motion by surface friction. So friction wheels motion is transmitted between the wheels by friction between them,
The high coefficient of friction of the contact face ensures high efficiency coupled with low slip.

- Industrial washing machines
- Conveyor belts
- Theatre stages
- Theme parks
- Bearing systems
- Wind turbines
- Drum drives

Types Friction wheels can be classified according to their friction surface:

• **External friction wheels** The wheels are cylinders. Their outer edges are in contact and they rotate in opposite directions.

• **Internal friction wheels** The wheels are cylinders. The inner edge of the bigger wheel is in contact with the outer edge of the smaller wheel. Both wheels rotate in the same direction.

• **Bevel friction wheels** The wheels are cones. The surfaces in contact are the surfaces of the cones. The wheels rotate in opposite directions.

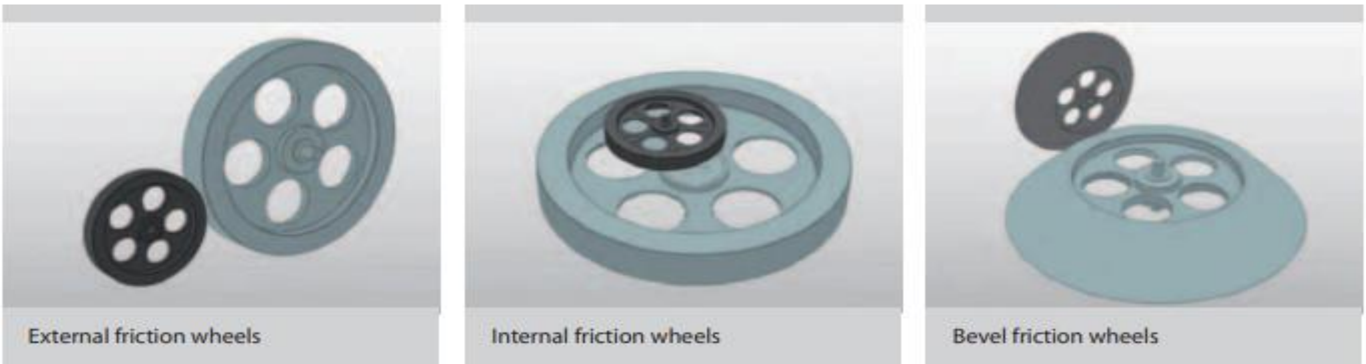


Figure 89.friction wheel

Notice that always the smallest wheel spins the fastest .in exchange it torque (rotary strength) is lowest. To calculate the speed of the driven wheel we employ the equation.

$$D_1 \cdot N_1 = D_2 \cdot N_2$$

D1=diameter of the driver wheel
D2= diameter of the driven wheel
N1= speed of driver wheel ,usually in revolution per minute (rpm)
N2=speed of the driven wheel(rpm)

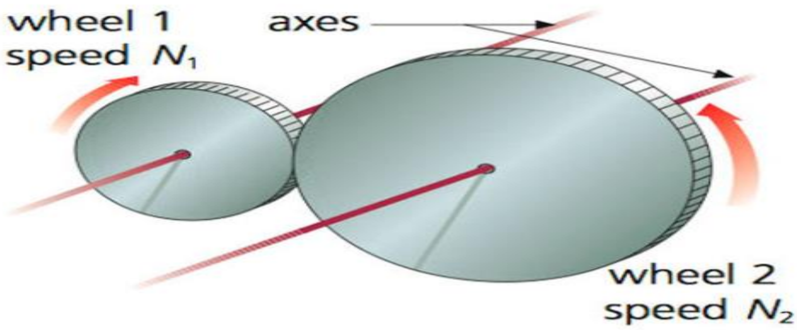


Figure 90.Transmission ration

The speed ration between the driver and driven wheel is calculating as follow

$$\mu = N2 / N1$$

10. Coupling

A coupling can be defined as a mechanical device that permanently joins two rotating shafts to each other. The most common application of coupling is joining of shafts of two separately built or purchased units so that a new machine can be formed. For example, a coupling is used to join the output shaft of an engine to the input shaft of a hydraulic pump to raise water from well. A coupling is used to join the output shaft of an electric motor to the input shaft of a gearbox in machine tools. A coupling is also used to join the output shaft of an electric motor to the input shaft of a compressor.
Requirements of a good shaft coupling
A good shaft coupling should have the following requirements:

- It should be easy to connect or disconnect.
- It should transmit the full power from one shaft to the other shaft without losses.
- It should hold the shafts in perfect alignment.
- It should have no projecting parts.

Uses

Shaft couplings are used in machinery for several purposes. The most common of which are the following.

- To transfer power from one end to another end.(ex: motor transfer power to pump through coupling) Primary function.

- To provide for the connection of shafts of units that are manufactured separately such as a motor and generator and to provide for disconnection for repairs or alterations.
- To provide for misalignment of the shafts or to introduce mechanical flexibility.
- To reduce the transmission of shock loads from one shaft to another.
- To introduce protection against overloads.
- To alter the vibration characteristics of rotating units.
- To connect driving and the driven part slips when overload occurs

6. 1. Types of shaft couplings:

Shaft couplings are divided into two main groups as follows:

1.**Rigid coupling**. It is used to connect two shafts which are perfectly aligned.

Following types of rigid coupling are important from te point of view:

- Sleeve or muff coupling
- Clamp or split –muff or compression coupling
- Flange coupling

2**.Flexible coupling**. It is used to connect two shafts having both lateral and angular misalignment. Following types of flexible coupling are important from the subject point of view:

- Bushed pin type coupling
- Universal coupling
- Oldham coupling

6.2.The difference between rigid and flexible couplings is as follows

(i) A rigid coupling cannot tolerate misalignment between the axes of the shafts. It can be used only when there is precise alignment between two shafts. On the other hand, the flexible coupling, due to provision of flexible elements like bush or disk, can tolerate 0.5° of angular misalignment and 5 mm of axial displacement between the shafts.

(ii) The flexible elements provided in the flexible coupling absorb shocks and vibrations. There is no such provision in rigid coupling. It can be used only where the motion is free from shocks and vibrations.

(iii) Rigid coupling is simple and inexpensive. Flexible coupling is comparatively costlier due to additional parts

A good coupling, rigid or flexible, should satisfy the following requirements:

(i) The coupling should be capable of transmitting torque from the driving shaft to the driven shaft.

(ii) The coupling should keep the two shafts in proper alignment.

(iii) The coupling should be easy to assemble and disassemble for the purpose of repairs and alterations.

(iv) The failure of revolving bolt heads, nuts, key heads and other projecting parts may cause accidents. They should be covered by giving suitable shape to the flanges or by providing guards.

6.3. The couplings are standardized

MUFF COUPLING

Muff coupling is also called *sleeve coupling* or *box coupling*. It is a type of rigid coupling. The construction of the muff coupling is shown in Fig80. It consists of a sleeve or a hollow cylinder, which is fi tted over the ends of input and output shafts by means of a sunk key. The torque is transmitted from the input shaft to the sleeve through the key. It is then transmitted from the sleeve to the output shaft through the key.

Muff coupling has following advantages:

(i) It is the simplest form of coupling with only two parts, viz., sleeve and key. It is simple to design and manufacture.
(ii) It has no projecting parts except the key-head. The external surface of the sleeve is smooth. This is an advantage from the standpoint of safety to the operator.
(iii) It has compact construction with small radial dimensions.
(iv) It is cheaper than other types of coupling.

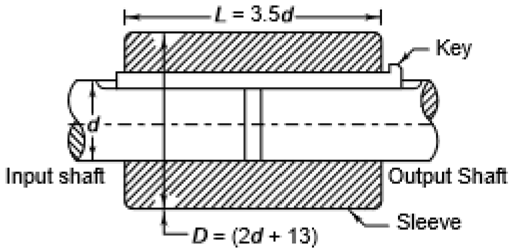


Figure 91.Muff coupling

Muff coupling has following disadvantages:

- Muff coupling is difficult to assemble or dismantle. The sleeve has to be either shifted over the shaft by half of its length or the ends of the shafts have to be drawn together or apart by half length of the sleeve.
- It is a rigid type of coupling and requires accurate alignment of shafts. It cannot tolerate misalignment between the axes of two shafts. The misalignment of shafts, caused by inaccurate assembly, induces forces, which tend to bend the shafts.
- Since there is no flexible element in the cou-pling, it cannot absorb shocks and vibrations.

For the sleeve of muff coupling, the standard proportions used in practice are as follows:

$D = (2d + 13)$ mm $\quad L = 3.5\,d$

where, D = outer diameter of the sleeve (mm)

L = axial length of the sleeve (mm)

d = diameter of the shaft (mm)

6.4.DESIGN PROCEDURE FOR MUFF COUPLING

The basic procedure for finding out the dimensions of the muff coupling consists of the following steps:

Calculate the diameter of each shaft by the following equations:

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} \quad \text{and} \quad \tau = \frac{16\, M_t}{\pi d^3}$$

Calculate the dimensions of the sleeve by the following empirical equations,

$$D = (2d + 13)\ \text{mm} \quad \text{and} \quad L = 3.5\, d$$

Also, check the torsional shear stress induced in the sleeve by the following equations:

$$\tau = \frac{M_t r}{J} \quad J = \frac{\pi(D^4 - d^4)}{32} \quad r = \frac{D}{2}$$

Determine the standard cross-section of flat sunk key. The length of the key in each shaft is one-half of the length of the sleeve. Therefore,

$$l = \frac{L}{2}$$

With these dimensions of the key, check the shear and compressive stresses in the key

$$\tau = \frac{2M_t}{dbl} \quad \text{and} \quad \sigma_c = \frac{4M_t}{dhl}$$

6.6. CLAMP COUPLING

The clamp coupling is also called compression coupling or split muff coupling. It is a rigid type of coupling. In this coupling, the sleeve is made of two halves, which are split along a plane passing through the axes of shafts. The construction of the clamp coupling is shown in Fig.The two halves of the sleeve are clamped together by means of bolts. The number of bolts can be four or eight. They are always in multiples of four. The bolts are placed in recesses formed in the sleeve halves.

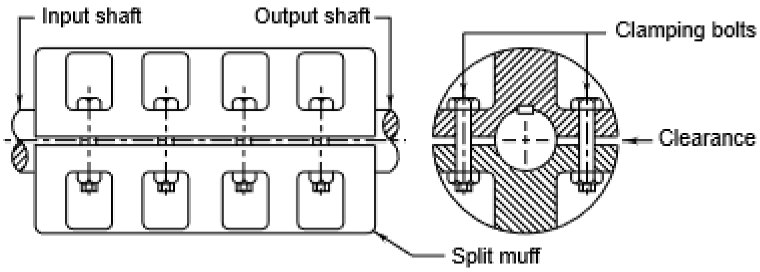


Figure 92.CLAMP COUPLING

The clamp coupling has the following advantages:

- It is easy to assemble and dismantle.
- It can be easily removed without shifting the shaft in axial direction, unlike solid muff coupling.
- As compared with flange coupling, clamp coupling has small diametral dimensions.

The disadvantages of clamp coupling are as follows:

- There is difficulty in dynamic balancing of the coupling. Therefore, it is not possible to use the clamp coupling for high-speed applications.
- Clamp coupling is unsuitable for shock loads.
- It is necessary to provide a guard for the coupling to comply with the factory regulation act.

The main application of clamp coupling is for line shaft in power transmission. Nowadays, the line shaft and the clamp coupling have become obsolete. Clamp coupling is usually designed on the basis of standard proportions for sleeve halves and clamping bolts.

For sleeve halves,

$D = 2.5d$

$L = 3.5d$

where

D = outer diameter of sleeve halves (mm)

L = length of sleeve (mm)

d = diameter of shaft (mm) For clamping bolts,

$$d_1 = 0.2d + 10 \text{ mm}$$

The frictional force is (fN) andfrictional torque is given by,

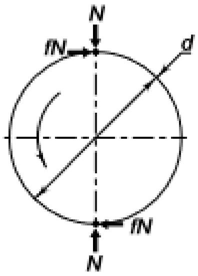
$$M_t = fN\left(\frac{d}{2}\right) + fN\left(\frac{d}{2}\right) = fNd$$

where,

f = coefficient of friction

From expression (a) and (b),

$$M_t = \frac{fd\,P_1 n}{2}$$



For clamping bolt

$$d_1 = 0.2d + 10 \text{ mm}$$

$$d < 55 \text{ mm}$$

$$d_1 = 0.15d + 15 \text{ mm}$$

$$d > 55 \text{ mm}$$

The clamping force of each bolt is given

$$P_1 = \frac{\pi}{4} d_1^2 \sigma_t$$

where P_1 = tensile force on each bolt (N)

d_1 = core diameter of clamping bolt (mm)

σ_t = permissible tensile stress (N/mm^2)

clamping force on each shaft is given by,

$$N = \frac{P_1 n}{2}$$

where,

n = total number of bolts

N = clamping force on each shaft (N)

6.7.DESIGN PROCEDURE FOR CLAMP COUPLING

The basic procedure for finding out the dimensions of clamp coupling consists of the following steps:

Calculate the diameter of each shaft by the following equations:

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} \quad \text{and} \quad \tau = \frac{16 M_t}{\pi d^3}$$

The shaft, key and clamping bolts are usually made of plain carbon steel.

Calculate the main dimensions of the sleeve halves by using the following empirical equations

$$D = 2.5\,d \quad \text{and} \quad L = 3.5\,d$$

The sleeve halves are made of grey cast iron of Grade FG 200.

Determine the standard cross-section of the flat key:The length of the key in each shaft is one-half of the length of sleeve. Therefore,

$$l = \frac{L}{2}$$

With these dimensions of the key, check the shear and compressive stresses in the key by Eqs

$$\tau = \frac{2M_t}{dbl} \quad \text{and} \quad \sigma_c = \frac{4M_t}{dhl}$$

Calculate the diameter of clamping bolts by

$$P_1 = \frac{2M_t}{fdn} \quad \text{and} \quad P_1 = \frac{\pi}{4} d_1^2 \sigma_t$$

RIGID FLANGE COUPLINGS

A flange coupling consists of two flanges—one keyed to the driving shaft and the other to the driven shaft as shown in Fig. The two flanges are connected together by means of four or six bolts arranged on a circle concentric with the axes of the shafts. Power is transmitted from the driving

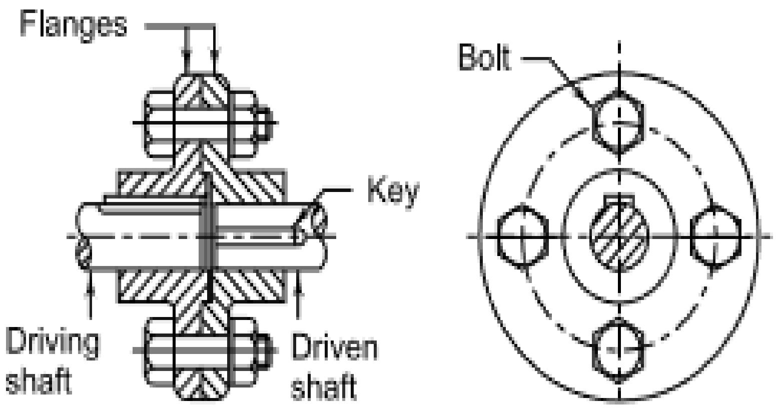


Figure 93.Unprotected Type Flange Coupling

Finally, power is transmitted from the right side flange to the driven shaft through the key. Since flange coupling is rigid type of coupling, provision should be made for precise location of the axes of two shafts.

There are two types of rigid flange couplings:

unprotected and protected. The flange coupling is the unprotected type of coupling. The revolving bolt heads and nuts are dangerous to the operator and may lead to accident.

Protected type flange coupling. In this case, protecting circumferential rims cover the bolt heads and nuts. In case of failure of bolts while the machine is being run, the broken pieces will dash against this rim and eventually fall down. This protects the operator against injuries.

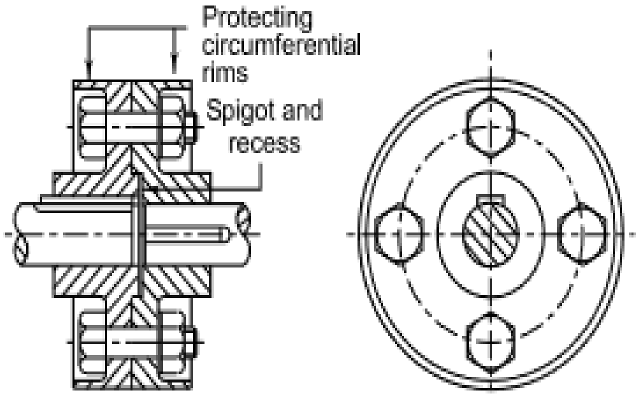


Figure 94.Protected Type Rigid Coupling

The rigid flange couplings have the following advantages:

- Rigid coupling has high torque transmitting capacity.
- Rigid coupling is easy to assemble and dismantle.
- Rigid coupling has simple construction. It is easy to design and manufacture.

The rigid flange couplings have the following disadvantages:

- It is a rigid type of coupling. It cannot tolerate misalignment between the axes of two shafts.
- It can be used only where the motion is free from shocks and vibrations.
- It requires more radial space.

The function of the hub is to transmit the torque from the shaft to the central flange and vice versa. The central portion of the flange has holes to accommodate the bolts

6.9. The standard proportions for various dimensions of the flange shown

d_h = outside diameter of hub

$$d_h = 2d$$

l_h = length of hub or effective length of key

$$l_h = 1.5\,d$$

D = pitch circle diameter of bolts

$$D = 3d$$

t = thickness of flanges

$$t = 0.5\,d$$

t_1 = thickness of protecting rim

$$t_1 = 0.25\,d$$

d_r = diameter of spigot and recess

$$d_r = 1.5\,d$$

D_0 = outside diameter of flange

$$D_0 = (4d + 2t_1)$$

6.1.1. BUSHED-PIN FLEXIBLE COUPLING

Rigid coupling can be used only when there is perfect alignment between the axes of two shafts and the motion is free from vibrations and shocks. In practice, it is impossible to obtain perfect alignment of shafts.

Misalignment exists due to the following reasons:

a. deflection of shafts due to lateral forces
b. error in shaft mounting due to manufacturing tolerances
c. use of two separately manufactured units such as an electric motor and a worm gear box; and
d. thermal expansion of the parts.

If rigid coupling is used in such circumstances, the misalignment causes excessive bearing reactions resulting in vibrations and wear. To overcome this problem, flexible couplings are used. A flexible coupling employs a flexible element like a rubber bush between the driving and the driven flanges. This flexible rubber bush not only accommodates the misalignment but also absorbs shocks and vibrations. The basic types of misalignment between axes of the input and output shafts. A flexible coupling can tolerate 0.5 mm of lateral or axial misalignment and 1.5° of angular misalignment.

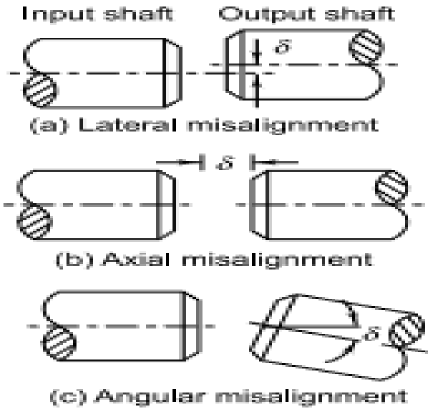


Figure 95.Types of Misalignment

The bushed-pin type flexible coupling has following advantages:

i. It can tolerate 0.5 mm of lateral or axial misalignment and 1.5° of angular misalignment.
ii. It prevents transmission of shock from one to the other and absorbs vibrations.
iii. It is simple in construction and easy to shaft assemble and dismantle. It is easy to design and manufacturing the coupling
iv. It can be used for transmitting high torques

The disadvantages of bushed-pin type flexible coupling are as follows:

(i) The cost of flexible coupling is more than that of rigid coupling due to additional parts.
(ii) It requires more radial space compared with other types of coupling.

In flexible coupling, the input flange accommodates the rubber bushes of comparatively large diameter than the diameter of the pins accommodated in the output flange as shown in Fig

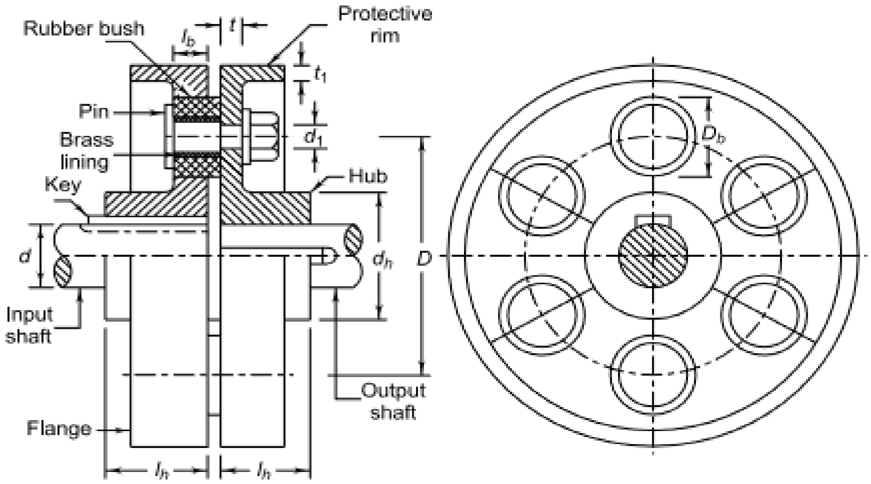


Figure 96.Flexible Coupling

The standard proportions for various dimensions of the flanges

(i) d_h = outside diameter of hub $d_h = 2d$

(ii) l_h = length of hub or effective length of key $l_h = 1.5\,d$

(iii) D = pitch circle diameter of pins

$D = 3d$ to $4d$

(iv) t = thickness of output flange

$t = 0.5\,d$

(v) t_1= thickness of protective rim $t_1 = 0.25\,d$

(vi) d_1 = diameter of pin

$$d_1 = \frac{0.5d}{\sqrt{N}}$$

where N is the number of pins and d is the shaft diameter.

6.1.2.DESIGN PROCEDURE FOR FLEXIBLE COUPLING

The basic procedure for finding out the dimensions of bushed pin type flexible coupling consists of the following steps:

(i) ***Shaft Diameter*** Calculate the shaft diameter by using the following two equations:

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} \quad \text{and} \quad \tau = \frac{16\,M_t}{\pi d^3}$$

Dimensions of Flanges Calculate the dimensions of the flanges by the following empirical relationships:

$$\begin{aligned} d_h &= 2d \\ l_h &= 1.5d \\ D &= 3d \text{ to } 4d \\ t &= 0.5d \\ t_1 &= 0.25d \end{aligned}$$

The torsional shear stress in the hub can be calculated by considering it as a hollow shaft subjected to torsional moment M_t. The inner and outer diameters of the hub are d and d_h respectively. The torsional shear stress in the hub is given by,

$$\tau = \frac{M_t r}{J}$$
$$J = \frac{\pi (d_h^4 - d^4)}{32}$$
$$r = \frac{d_h}{2}$$

The shear stress in the flange at the junction with the hub is calculated by Eq

$$M_t = \frac{1}{2} \pi d_h^2 t \tau$$

(i) ***Diameter of Pins*** The number of pins is usually 4 or 6. The diameter of the pins is calculated by the following empirical equation,

$$d_1 = \frac{0.5\, d}{\sqrt{N}}$$

Determine the shear stress in the pins by Eq.

$$\tau = \frac{8M_t}{\pi d_1^2 DN}$$

Dimensions of Bushes Calculate the outer diameter of the rubber bush by Eq.

$$M_t = \frac{1}{2} D_b^2 DN$$

Calculate the effective length of the rubber bush by the following relationship,

$$l_b = D_b$$

(iv) ***Dimensions of Keys*** Determine the standard cross-section of fl at key ,The length of the key in each shaft is l_h . Therefore,

$l = l_h$

With the above dimensions of the key, check the shear and compressive stresses in the key by Eqs

$$\tau = \frac{2M_t}{dbl}$$
$$\sigma_c = \frac{4M_t}{dhl}$$

Solid flanged coupling

Couplings for marine or automotive propeller shafts demand greater strength and reliability. For these applications, flanges are forged integral with the shafts. The flanges are joined together by means of a number of headless taper bolts.

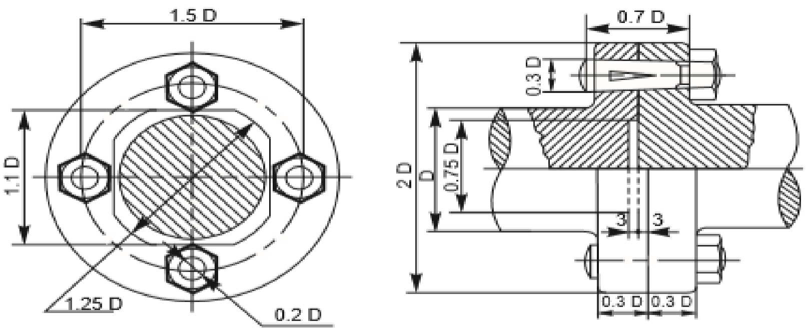


Figure 97.Solid flanged coupling

Compression coupling

This consists of a compressible steel sleeve which fits on to the ends of the shafts to be coupled. The sleeve corresponds to the shaft diameter and its outer surface is of double conical form. The sleeve has one through cut longitudinally and five other cuts, equi-spaced, but running alternately from opposite ends to about 85% of its length; making it radially flexible. The two flanges used have conical bores and are drawn towards each other by means of a number of bolts and nuts, making the sleeve firmly compressed onto the shafts. Here, the friction between the shafts and sleeve assists power transmission and the bolts do not take any load. Because of the presence of flexible sleeve, the coupling takes care of both axial and angular mis-alignment of shafts

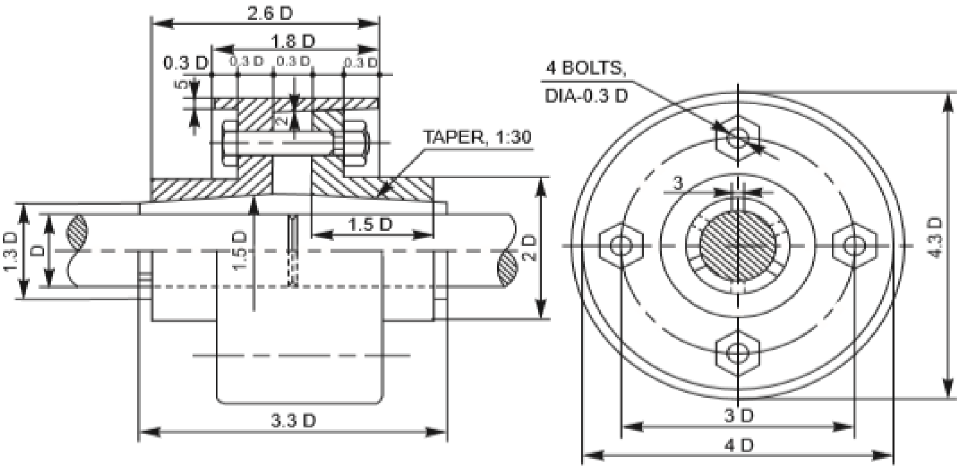


Figure 98.Compression coupling

dis- engaging coupling

Disengaging couplings are used when power transmission from one shaft to another is intermittent. With this, the shafts can be engaged or disengaged as and when required, even during rotation. A dis-engaging coupling in general consists of one part firmly fixed to the driving shaft and another one mounted with provision for sliding over the driven shaft. The part that is mounted on the driven shaft, can be made to slide at will to engage or disengage from the rotating driving shaft. The following are the examples of dis-engaging couplings.

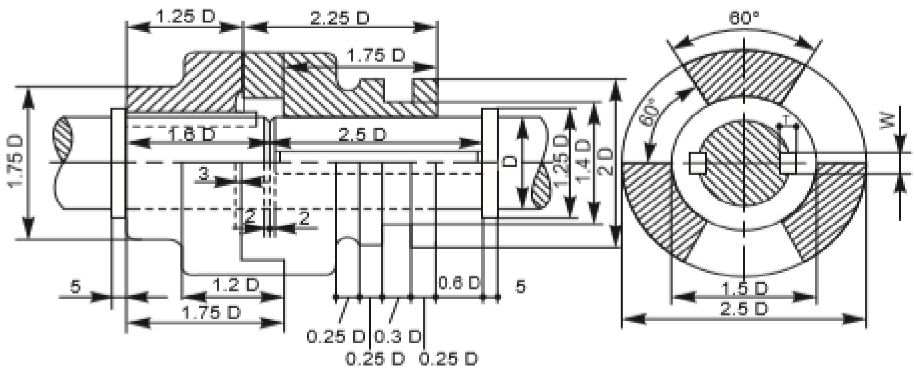


Figure 99.dis- engaging coupling

non- aligned coupling

Non-aligned couplings are used to transmit power between two shafts which are not coaxial. The following are the examples of non-aligned couplings:

universal coupling (hookes joint)

It is a rigid coupling that connects two shafts, whose axes intersect if extended. It consists of two forks which are keyed to the shafts. The two forks are pin joined to a central block, which has two arms at right angle to each other in the form of a cross (Figure below). The angle between the shafts may be varied even while the shafts are rotating.

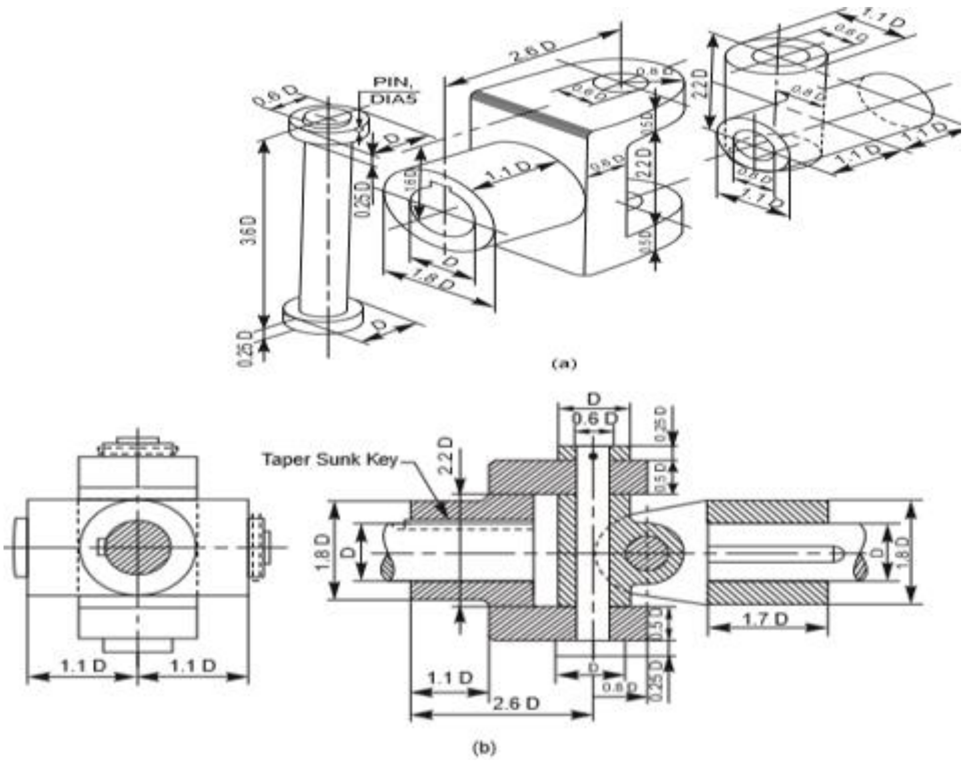


Figure 100.universal coupling (hookes joint)

oldham coupling

It is used to connect two parallel shafts whose axes are at a small distance apart. Two flanges, each having a rectangular slot, are keyed, one on each shaft. The two flanges are positioned such that, the slot in one is at right angle to the slot in the other. To make the coupling, a circular disc with two rectangular projections on either side and at right angle to each other, is placed between the two flanges. During motion, the central disc, while turning, slides in the slots of the flanges. Power transmission takes place between the shafts, because of the positive connection between the flanges and the central disc (Figure below).

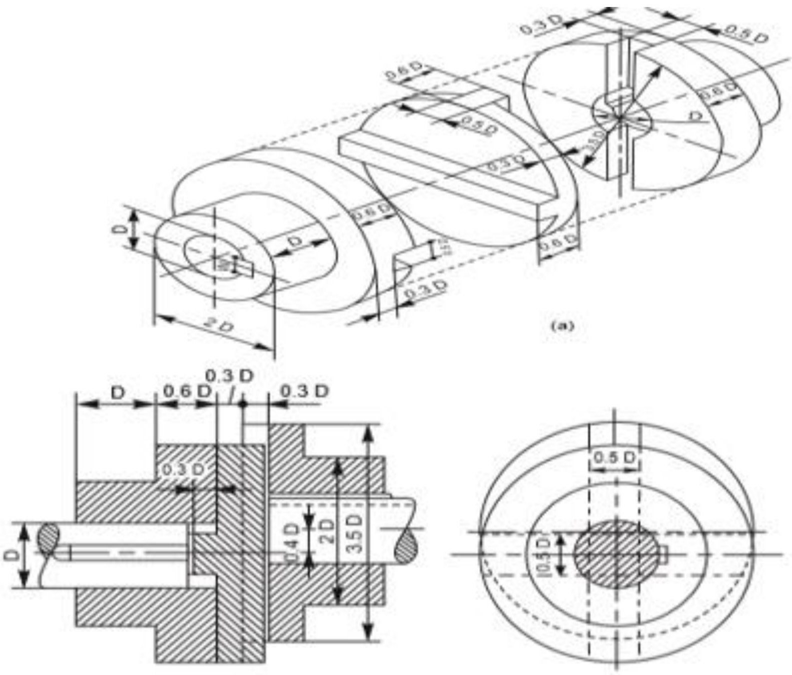


Figure 101.oldham coupling

Content/Topic4: Machine control components

1.Switches

Limit switches are used to control and monitor automated machinery and industrial equipment. As opposed to a sensor limit switches make physical contact with the target causing the circuit to turn on or off. Some key benefits of limit switches include:

- ✓ Can be used for safety and monitoring in a diverse range of applications.
- ✓ Physical contact with target offers precise operating points.
- ✓ Depending on the application a roller/plunger, lever, or whisker actuator can be used.
- ✓ Several housing styles to choose from including but not limited to die cast aluminum, silicon, plastic.
- ✓ Models are available for hazardous locations, compact areas, high pressure, extend temperature applications and more

2.Indicators

Analog measured values are indicated, evaluated and calculated by process indicators. Large displays offer easily readability at all ambient conditions. The universal inputs enable the connection of current, voltage, RTDs and TCs. Loop powered indicators require no power supply and can be universally used in current measuring circuits. They can be easily installed in intrinsically safe applications. Fieldbus indicators support all bus devices and indicate the values communicated on the bus.

Indicators with control function combine several functionalities in one device:

- Active barrier
- Transmitter
- Control unit with relay

3. Sensors

Sensors are critical for converting physical properties into discrete and analog signals, letting machine automation systems do their tasks reliably and consistently.

Machines need sensors and transmitters to operate reliably and consistently. All it takes is a small drop in air pressure or a slight jam to cause a machine to crash; these sorts of problems can often be detected before they lead to bigger issues.

Types of sensor

- ✓ **Temperature Sensors**

A temperature sensor is a device that collects information concerning the temperature from a resource and changes it to a form that can be understood by another device. These are commonly used category of sensors which detect Temperature or Heat and it also measures the temperature of a medium.:

- ✓ **Digital Temperature Sensors**

these Digital Temperature Sensors are silicon-based temperature- sensing ICs that provide accurate output through digital representations of the temperatures they are measuring. This simplifies the control system's design, compared to approaches that involve external signal conditioning and an analog-to digital converter (ADC).

- ✓ **Humidity & Temperature Sensors**.
- ✓ **turned parts for Pressure Sensors**
- ✓ **these Pressure sensors**

are widely used in Industrial and hydraulic systems, these are high pressure industrial automation sensors also used in climate control systems.

- ✓ **Vaccum Sensors**

when the Vaccum pressure is below atmospheric pressure levels and it can be difficult to sense through mechanical methods. These sensors generally depend on a heated wire with electrical resistance correlating to temperature. When vaccum pressure increases, convection falls down and wire temperature up rises. Electrical resistance increases proportionally and is calibrated adjacent to pressure in order to give an effective measurement of the vaccum.

- ✓ **MEMS Sensors (Micro-electro-mechanical Systems)**

These MEMS industrial automation sensors convert measured mechanical signals into electrical signals.

- ✓ **motion sensors**

use data processing algorithms designed on a motion interaction platform which integrates numerous low-cost MEMS motion sensors with ZigBee wireless technology to carry personified interactions while working together with machines.

Torque sensors

The torque sensors complete with essential mechanical stops, raise overload capacity and offer additional guard during mounting and operation.

Rotating Torque & Torque Transducers are few important sensors used in industrial automation.

All these above mentioned sensors are increasingly utilised in the automation industry.

4.Actuators:

Actuators are mechanical or electro-mechanical devices that provide controlled and sometimes limited movements or positioning which are operated electrically, manually, or by various fluids such as air, hydraulic, etc.

Two basic motions are linear and rotary.

- **Linear actuators** : convert energy into straight line motions, typically for positioning applications, and usually have a push and pull function. Some linear actuators are unpowered and manually operated by use of a rotating knob or handwheel.
- **Rotary actuators** :convert energy to provide rotary motion. A typical use is the control of various valves such as a ball valves or butterfly valves. Each actuator type has versions for various power configurations and come in many styles and sizes depending on the application. Linear chain actuators provide push and pull motions with rigid chains.

Types of Actuators

- ✓ **Electric Linear:** Electric Linear Actuators are electrically powered, mechanical linear actuators consisting of motors, linear guides, and drive mechanisms, which are used to convert electrical energy into linear displacement through mechanical transmission, electro-magnetism, or thermal expansion to provide straight line push/pull motion.
- ✓ **Electric Rotary:** Electric Rotary Actuators are electrically powered, mechanical devices consisting of motors and output shaft mechanisms with limited rotary travel which are used to convert electrical energy into rotational motion.
- ✓ **Fluid Power Linear:**luid Power Linear Actuators are mechanical devices consisting of cylinder and piston mechanisms that produce linear displacement by means of hydraulic fluid, gas, or differential air pressure. Key specifications include the intended application, fluid power type, mounting configuration, travel length, and force capacities, as well as physical dimensions as required.
- ✓ **Fluid Power Rotary**: Fluid Power Rotary Actuators are fluid powered, mechanical devices consisting of cylinder and piston mechanisms, gearing, and output shafts giving limited rotational travel, which are used to convert hydraulic fluid, gas, or differential air pressure into rotational motion. Key specifications include the intended application, fluid power type, drive mechanism, mounting configuration, output configuration, rotation limits, and force capacities, as well as physical dimensions as required.
- ✓ **Linear Chain Actuators**: Linear Chain Actuators are mechanical devices consisting of sprockets and sections of chain which are used for providing linear motion via the free ends of the specially designed chains. Key specifications include the intended application, drive method and mechanism, actuation length, chain size, and the mounting configuration. Linear chain actuators are used primarily in motion control applications for providing a straight line push or pull motion.
- ✓ **Manual Linear**: Manual Linear Actuators are mechanical devices providing linear displacement through the translation of manually rotated screws or gears and consist of hand-operated knobs or wheels, gearboxes, and guided linear motion mechanisms. Key specifications include the intended application, actuator type, drive mechanism, travel length, and other physical dimensions as required.

- ✓ **Manual Rotary**:anual Rotary Actuators are mechanical devices providing rotary output through the translation of manually rotated screws, levers, or gears, and are usually composed of hand-operated knobs, levers, or handwheels, gearboxes or threaded nut mechanisms, and output shafts. Key specifications include the intended application, drive method and mechanism, mounting configuration, valve type, if applicable, as well as physical dimensions as required.

5.Computer controllers:

The computer uses sensors on the engine and transmission to detect such things as throttle position, vehicle speed, engine speed, engine load, stop light switch position, etc. to control exact shift points as well as how soft or firm the shift should be. Some computerized transmissions even learn your driving style and constantly adapt to it so that every shift is timed precisely when you would need it. The computer monitors this activity to make sure that the driver does not select a gear that could over speed the engine and damage it.

Another advantage to these “smart” transmissions is that they have a self-diagnostic mode which can detect a problem early on and warn you with an indicator light on the dash. A technician can then plug test equipment in and retrieve a list of trouble codes that will help pinpoint where the problem is.

In mechanical engineering as production we have used Computer Numerical Control (CNC) and Numerical control (NC). Fortunately, machine tools were designed so that the operator was standing in front of the machine while operating the controls. This design is no longer necessary, since in CNC the operator no longer controls the machine tool movements

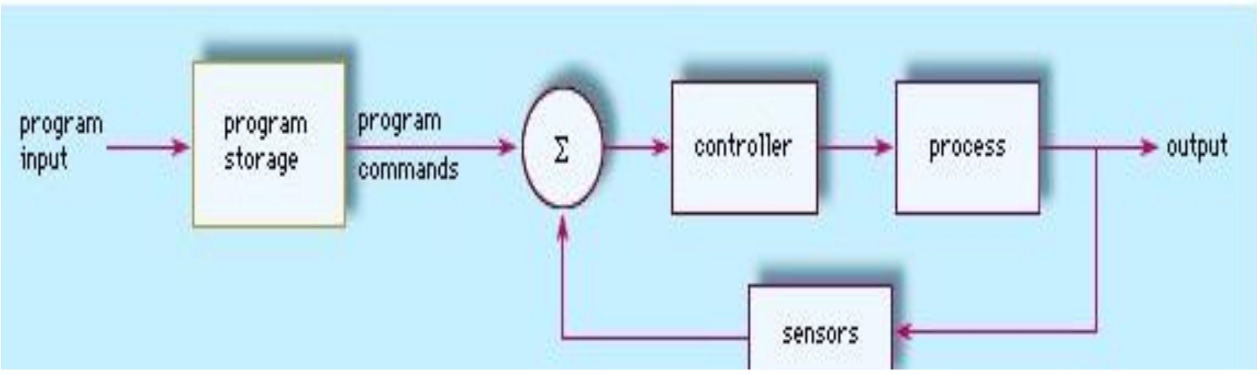


Figure 102.Computer controllers:

LO1.2 -Identify transmission elements

Content/Topic1:Types of mechanical power transmissions

1. Manual transmission elements

1.1. Shafts

Transmission shafts can be found in a manual transmission gearbox. The purpose of a transmission gearbox is to transfer the high output of an automobile's engine to the wheels, and in the process reduce it to a compatible speed. The gearbox does this through a complex arrangement of gears and shafts

1.2. Dog clutch

A dog clutch (also known a dog gears) is a type of clutch that couples two rotating shafts or other rotating components not by friction but by interference or clearance fit. The two parts of the clutch are designed such that one will push the other, causing both to rotate at the same speed and will never slip.

Dog clutches are used where slip is undesirable and/or the clutch is not used to control torque. Without slippage, dog clutches are not affected by wear in the same way that friction clutches are. Dog clutches are used inside manual automotive transmissions to lock different gears to the rotating input and output shafts. simple dog clutch can be found in Sturmey-Archer ,hub gear.

1.3. Synchromesh

In order to provide smooth gearshifts without requiring the driver to manually match the engine revs for each gearshift, most modern passenger car transmissions use 'synchromesh' (also called 'synchronizer rings') on the forward gears. These devices automatically match the speed of the input shaft with that of the gear being selected, thus removing the need for the driver to use techniques such as double clutching. the need for synchromesh in a constant-mesh transmission is because the dog clutches require the input shaft speed to match that of the gear being selected, otherwise, the dog teeth will fail to engage and a loud grinding sound will be heard as they clatter together. Therefore, to speed up or slow down the input shaft as required, cone-shaped brass synchronizer rings are attached to each gear. When the driver moves the gearshift lever towards the next gear, these synchronizer rings press on the cone-shaped sleeve on the dog collar so that the friction forces can reduce the difference in rotational speeds.

1.4. Reverse

Even in modern transmissions where all of the forward gears are in a constant-mesh configuration, often the reverse gear uses the older sliding mesh ('crash box') configuration. This means that moving the gearshift lever into reverse results in gears moving to mesh together. Another unique aspect of the reverse gear is that it consists of two gears— an idler gear on the countershaft and another gear on the output shaft— and both of these are directly fixed to the shaft (i.e. they are always rotating at the same speed as the shaft). These gears are usually spur gears with straight-cut teeth which— unlike the helical teeth used for forward gear— results in a whining sound as the vehicle moves in reverse.

When reverse gear is selected, the idler gear is physically moved to mesh with the corresponding gears on the input and output shafts. To avoid grinding as the gears, begin to the mesh, they need to be stationary. Since the input shaft is often still spinning due to momentum (even after the car has stopped), a mechanism is needed to stop the input shaft, such as using the synchronizer rings for 5th gear. However, some vehicles do employ a synchromesh system for the reverse gear, thus preventing possible crunching if reverse gear is selected while the input shaft is still spinning

1.5. Float shifting

Float shifting (also called "floating gears") is changing gears without disengaging the clutch, usually on a non-synchronized transmission used by large trucks. Since the clutch is not used, it is easy to mismatch speeds of gears, and the driver can quickly cause major (and expensive) damage to the gears and the transmission.

1.6. Floor-mounted shifter

In most vehicles with manual transmission, gears are selected by manipulating a level called a gear stick, shift stick, gear level, gear selector or shifter connected to the transmission via linkages or cable and mounted on the floor. Dashboard, or steering column.

Moving the level forward, backward left, and right into specific position select particular gears.

Sample layout of a four –speed transmission, is show below.

N. mark neutral, the position where no gear are engaged and the engine is decouple from the vehicles driver wheels. The entire horizontal line is a neutral position. Though the shifter is usually spring-loaded so it will return to the center of the N position if not moved to another gears, the **R** mark reverse, the gear position used for moving the vehicles backward

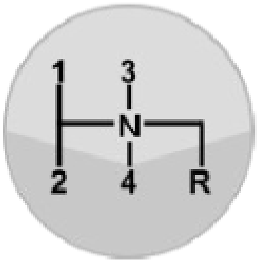


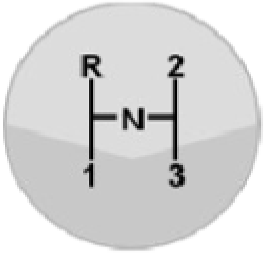
Figure 103four –speed transmission

This layout called the shift pattern. because of the shift quadrant, the basic arrangement is often called an H-pattern. The shift pattern is usually molded or printed on near the gear knob.

Typically, first gear is at the top at top left position with below, third up the right with fourth below and son on. Then only other pattern used in production vehicles manual transmission is known as a dog leg gearbox pattern. This pattern locates first at bottom left position, second up and to the right with third below, four up and to the right and so on, this pattern is found primary in race and race inspire vehicles. placing the selection for second gear above the above the position for third gear is desirable in racing as more frequent shifting occurs from second to third than from first to second.

1.7. Column mounted shifter

Some car has a gear to lever mounted on steering column of car. A 3 SPEED Column shifter, which came to be popular knowns as a three on the tree, began appearing in America in the late 1930 and became common during the 1940 and 190.vehicles by briefly equipped with it was very likely to be belong warranties. backing off the gas when above 28mph (45km/h to enable, and momentarily flooding the gas pedal to return to normal gear, the control simply disable overdrive for such situation as parking on a hill or preventing unwanted shifting into overdrive.



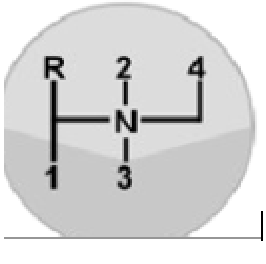


Figure 104. Column mounted shifter

Column shifters are mechanically similar to floor shifter ,although shifting occurs in vertical plane instead of horizontal one .because the shifter is further away from the transmission and the movement at the shifter and the transmission are in different plane, column shifters required more complicated linkage than floor shifter are ability to switch between the two most commonly used .second and third without letting go of the steering wheel, and lack of interference with passenger seating space in vehicles equipped with a bench seat.

2.Automatic transmission elements

2.1Gear box

Most modern automatic gearboxes have a set of gears called a planetary or epicyclic gear train.

A planetary gear set consists of a central gear called the sun gear, an outer ring with internal gear teeth (also known as the annulus, or ring gear), and two or three gears known as planet gears that rotate between the sun and ring gears. The planetary gear train is composed of planetary gear sets as well as clutches and bands. It is the mechanical system that provides the various gear ratios, altering the speed of rotation of the output shaft depending on which planetary gears are locked To effect gear changes, one of two types of clutches or bands are used to hold a particular member of the planetary gear set motionless,

2.2. Torque converter

On automatic transmissions, the torque converter takes the place of the clutch found on standard shift vehicles. It is there to allow the engine to continue running when the vehicle comes to a stop. The principle behind a torque converter is like taking a fan that is plugged into the wall and blowing air into another fan which is unplugged. If you grab the blade on the unplugged fan, you are able to hold it from turning but as soon as you let go, it will begin to speed up until it comes close to the speed of the powered fan. The difference with a torque converter is that instead of using air, it uses oil or transmission fluid, to be more precise.

A torque converter is a large doughnut-shaped fluid coupling (10" to 15" in diameter) that is mounted between the engine and the transmission. It consists of three internal elements that work together to transmit power to the transmission. The three elements of the torque converter are the Pump, the Turbine, and the Stator. The pump is mounted directly to the converter housing which in turn is bolted directly to the engine's crankshaft and turns at engine speed. The turbine is inside the housing and is connected directly to the input shaft of the transmission providing power to move the vehicle.

2.3. Brake bands and clutches

Brake bands explanations

A clutch pack consists of alternating disks that fit inside a clutch drum. Half of the disks are steel and have splines that fit into groves on the inside of the drum. The other half have a friction material bonded to their surface and have splines on the inside edge that fit groves on the outer surface of the adjoining hub. There is a piston inside the drum that is activated by oil pressure at the appropriate time to squeeze the clutch pack together so that the two components become locked and turn as one.

2.5. Bands

A band is a steel strap with friction material bonded to the inside surface. One end of the band is anchored against the transmission case while the other end is connected to a servo. At the

appropriate time hydraulic oil is sent to the servo under pressure to tighten the band around the drum to stop the drum from turning.
The composition of bands includes metal lined with organic friction material. This element locks the ring or sun gear or allows them spinning. Its operation is controlled by a hydraulic unit. The band is installed on the gearbox casing: one end of the band is connected to the casing itself, while its 2-nd end is linked up with a running piston of the servo unit. Changing of the band state has a direct impact on the operation of the drum half of the clutch pack with the planet gear of the other half of the clutch pack. That’s how an automatics shift gears in compliance with the RPM rate.

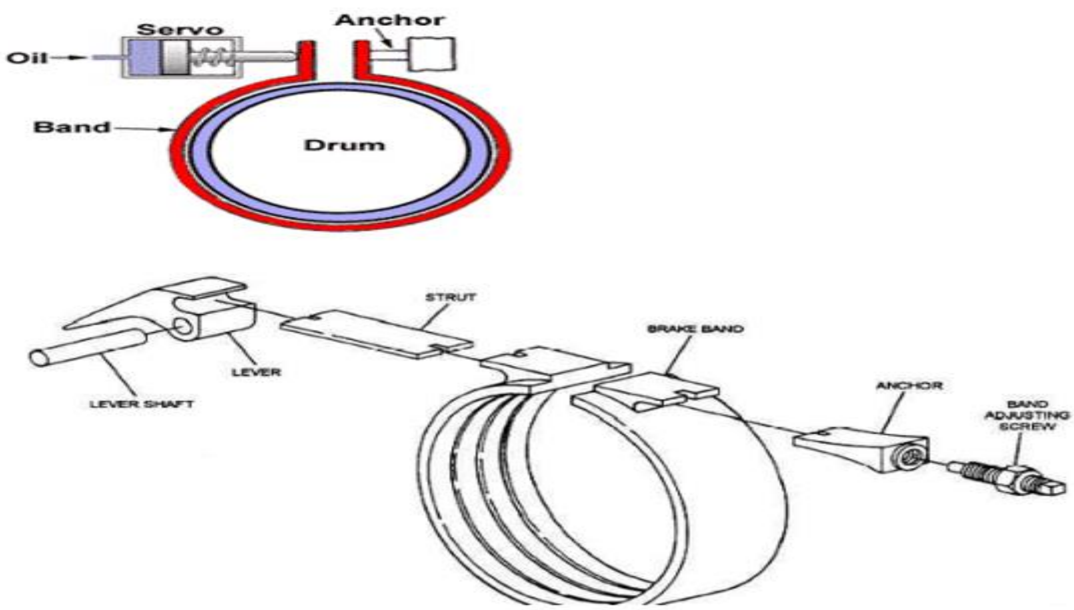


Figure 105.. Bands

3. Semi-Automatic transmission elements

❖ **Clutch slave cylinder**

In semi-hydraulic systems the slave cylinder is typically located at the outside of the transmission housing or serves as an actuation device for the clutch lever. In this case, the slave cylinder comprises the housing, the piston with sealing, a precharge spring and a bleed screw. The precharge spring applies constant preload pressure on the release bearing which therefore rotates even if the release system is free of load. In this manner, no undesirable noise occurs. The bleed screw facilitates the flushing of the system during maintenance. In systems with CSC, the release bearing is directly connected to the piston and tensioned against the diaphragm spring tips of the clutch by the integrated precharge spring.

❖ **Electric motor clutch actuation**

❖ **Piston**

The piston is a reciprocating part of IC engine that performs a number of functions.

The main functions of the piston are as follows:

- It transmits the force due to gas pressure inside the cylinder to the crankshaft through the connecting rod.
- It compresses the gas during the compression stroke.
- It seals the inside portion of the cylinder from the crankcase by means of piston rings.
- It takes the side thrust resulting from obliquity of the connecting rod.
- It dissipates large amount of heat from the combustion chamber to the cylinder wall

❖ **Gear shift mechanism**

A clutch actuator is controlled to work through an electronic device between a handle manipulating assembly and a transmission so as to save part of mechanical mechanism connection, and an installing space is saved; on the premise that a gear transmission main tank is not changed, **only a gear-selecting and gear-shifting actuator** assembly is arranged outside a transmission upper cover, the actuator is controlled to work through a controller, and the goal of gear selection and gear shifting is achieved. Compared with traditional pure manual transmissions, according to the system, a clutch pedal can be trod and is not trod, and automatic separation and connection motion of a clutch is finished through a vehicle-mounted computer when the clutch pedal is not trod; due to the fact that electronic control and pneumatic execution are adopted.

4. Continuously variable transmission elements

A continuously variable transmission is a type of automatic transmission that seamlessly changes through a continuous range of different gear ratios. It is also known as stepless transmission, pulley transmission, single-speed transmission, and in case of motorcycles, a twist and go transmission.

There are several types of continuously variable transmissions,

4.1. Pulley

I there are two V-belt pulleys that are split perpendicular to their axes of rotation, with a V-belt running between them. The drive ratio is changed by moving the two sheaves of one pulley closer together and the two sheaves of the other pulley farther apart. The V-shaped cross section of the belt causes it to ride higher on one pulley and lower on the other. This changes the effective diameters of both pulleys, which changes the overall drive ratio. As the distance between the pulleys and the length of the belt does not change, both pulleys must be adjusted (one bigger, the other smaller) simultaneously in order to maintain the proper amount of tension on the belt.

4.2. Toroidal

are made up of discs and rollers that transmit power between the discs. The discs can be pictured as two almost conical parts, point to point, with the sides dished such that the two parts could fill the central hole of a torus. One disc is the input, and the other is the output. Between the discs are rollers which vary the ratio and which transfer power from one side to the other. When the roller's axis is perpendicular to the axis of the near-conical parts, it contacts the near-conical parts at same-diameter locations and thus gives a 1:1 drive ratio.

4.3. Hydrostatic transmissions

use a variable displacement pump and a hydraulic motor. All power is transmitted by hydraulic fluid. These types can generally transmit more torque, but can be sensitive to contamination. Some designs are also very expensive. However, they have the advantage that the hydraulic motor can be mounted directly to the wheel hub, allowing a more flexible suspension system and eliminating efficiency losses from friction in the drive shaft and differential components. This type of transmission is relatively easy to use because all forward and reverse speeds can be accessed using a single lever.

L O 1.3 -Identify supporting machine elements

- Content/Topic1:Types of supporting machine elements

1.1 Axle

An axle is a rod or shaft that rotates the wheels and supports the weight of your vehicle.

Axles come in three standard types,

1.Rear Axle: This axle is responsible for delivering power to the driving wheels. It comes in two halves, known as half shafts, which are connected by the differential. In most cases, rear axles are live, meaning they rotate with the vehicle's wheels.

2.Front Axle: Located in the front of the vehicle, this axle is responsible for assisting with steering and processing shocks from the uneven surface of the road. They have four main parts, which are the beam, the swivel pin, the track rod, and the stub axle. Front axles must be as sturdy as possible, and that's why they're usually made from carbon steel or nickel steel.

4. **Stub Axle:** Stub axles are attached to the vehicle's front wheels, with kingpins connecting these axles to the front axle.

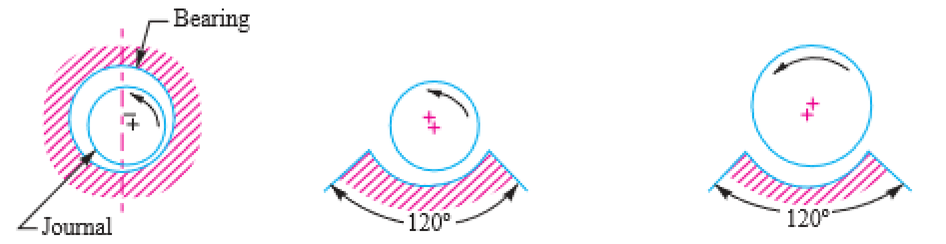
Sr. No.	Shaft	Axle
1	Rotating member	Non-Rotating member
2	Used to transmit the torque and support the transmission elements, like: gears, pulleys	Only used to support the transmission elements, like: wheels, pulleys etc
3	It is subject to torque, bending moment and axial force	It is subjected to bending moment and axial force
4	Example- line shaft, counter shaft, spindle, crankshaft	Example:- front axle of car, wheel axle of motorcycle etc.

Table. Different between shaft and axle

2. Bearings

1.2.1. Sliding Contact Bearings

A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load. Types of Sliding Contact Bearings The sliding contact bearings in which the sliding action is guided in a straight line and carrying radial loads, may be called slipper or guide bearings. Such type of bearings is usually found in cross-head of steam engines.



(*a*) Full journal bearing. (*b*) Partial journal bearing. (*c*) Fitted journal bearing.

Figure 106..Journal or sleeve bearings

The sliding contact bearings in which the sliding action is along the circumference of a circle or an arc of a circle and carrying radial loads are known as journal or sleeve bearings. When the angle of contact of the bearing with the journal is 360° as shown in Fig.98 (a), then the bearing is called a full journal bearing. This type of bearing is commonly used in industrial machinery to accommodate bearing loads in any radial direction.

When the angle of contact of the bearing with the journal is 120°, as shown in Fig. 26.3 (b), then the bearing is said to be partial journal bearing. This type of bearing has less friction than full journal bearing, but it can be used only where the load is always in one direction. The most common application of the partial journal bearings is found in rail road car axles. The full and partial journal bearings may be called as clearance bearings because the diameter of the journal is less than that of bearing.

The sliding contact bearings, according to the thickness of layer of the lubricant between the bearing and the journal, may also be classified as follows:

1. **Thick film bearings**. The thick film bearings are those in which the working surfaces are completely separated from each other by the lubricant. Such type of bearings is also called as hydrodynamic lubricated bearings.

2. **Thin film bearings**. The thin film bearings are those in which, although lubricant is present, the working surfaces partially contact each other at least part of the time. Such type of bearings is also called boundary lubricated bearings.

3. **Zero film bearings**. The zero film bearings are those which operate without any lubricant present.

4. **Hydrostatic or externally pressurized lubricated bearings**. The hydrostatic bearings are those which can support steady loads without any relative motion between the journal and the bearing. This is achieved by forcing externally pressurized lubricant between the members.

Properties of Sliding Contact Bearing Materials

1. **Compressive strength**. The maximum bearing pressure is considerably greater than the average pressure obtained by dividing the load to the projected area. Therefore, the bearing material should have high compressive strength to withstand this maximum pressure so as to prevent extrusion or other permanent deformation of the bearing.

2. Fatigue strength. The bearing material should have sufficient fatigue strength so that it can withstand repeated loads without developing surface fatigue cracks. It is of major importance in aircraft and automotive engines.

3. **Conformability.** It is the ability of the bearing material to accommodate shaft deflections and bearing inaccuracies by plastic deformation (or creep) without excessive wear and heating.

4. **Embed ability.** It is the ability of bearing material to accommodate (or embed) small particles of dust, grit etc., without scoring the material of the journal.

5**. Bendability.** Many high capacity bearings are made by bonding one or more thin layers of a bearing material to a high strength steel shell.

6**. Corrosion resistance**. The bearing material should not corrode away under the action of lubricating oil. This property is of particular importance in internal combustion engines where the same oil is used to lubricate the cylinder walls and bearings.

7. **Thermal conductivity**. The bearing material should be of high thermal conductivity so as to permit the rapid removal of the heat generated by friction.

8. Thermal expansion. The bearing material should be of low coefficient of thermal expansion, so that when the bearing operates over a wide range of temperature, there is no undue change in the clearance.

Lubricants

The lubricants are used in bearings to reduce friction between the rubbing surfaces and to carry away the heat generated by friction. It also protects the bearing against corrosion. All lubricants are classified into the following three groups :

1. Liquid, 2. Semi-liquid, and 3. Solid.

The liquid lubricants usually used in bearings are mineral oils and synthetic oils. The mineral oils are most commonly used because of their cheapness and stability.

grease is a semi-liquid lubricant having higher viscosity than oils. The greases are employed where slow speed and heavy pressure exist and where oil drip from the bearing is undesirable.

The solid lubricants are useful in reducing friction where oil films cannot be maintained because of pressures or temperatures.

Properties of Lubricants

1. **Viscosity:**It is the measure of degree of fluidity of a liquid. It is a physical property by virtue of which an oil is able to form, retain and offer resistance to shearing a buffer film-under heat and pressure.
2. **. Oiliness.** It is a joint property of the lubricant and the bearing surfaces in contact. It is a measure of the lubricating qualities under boundary conditions where base metal to metal is prevented only by absorbed film
3. **Density.** This property has no relation to lubricating value but is useful in changing the kinematic viscosity to absolute viscosity**.**

4. **Viscosity index**. The term viscosity index is used to denote the degree of variation of viscosity with temperature.
5. **Flash point.** It is the lowest temperature at which an oil gives off sufficient vapour to support a momentary flash without actually setting fire to the oil when a flame is brought within 6 mm at the surface of the oil.
6. **Fire point**. It is the temperature at which an oil gives off sufficient vapour to burn it continuously when ignited.
7. **Pour point or freezing point**. It is the temperature at which an oil will cease to flow when cooled.

Terms used in Hydrodynamic Journal Bearing

A hydrodynamic journal bearing is shown in Fig. 26.7, in which O is the centre of the journal and O' is the centre of the bearing.

Let D = Diameter of the bearing,

d = Diameter of the journal, and

l = Length of the bearing

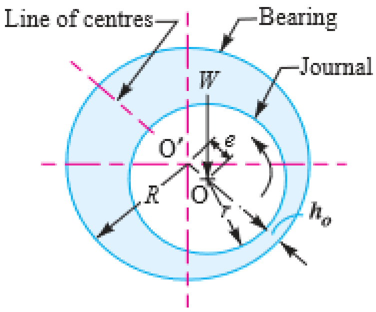


Figure 107.Hydrodynamics journal bearings

The following terms used in hydrodynamic journal bearing are important from the subject point of view:

1. **Diametral clearance**. It the difference between the diameters of the bearing and the journal. Mathematically, diametral clearance,c = D – d

2. **Radial clearance.** It is the difference between the radii of the bearing and the journal. Mathematically, radial clearance,

$$c_1 = R - r = \frac{D-d}{2} = \frac{c}{2}$$

3.Diametral clearance ratio. It is the ratio of the diametral clearance to the diameter of the journal. Mathematically, diametral clearance ratio

$$= \frac{c}{d} = \frac{D-d}{d}$$

4. **Eccentricity.** It is the radial distance between the centre (O) of the bearing and the displaced centre (O') of the bearing under load. It is denoted by e.

5. **Minimum oil film thickness**. It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by h0 and occurs at the line of centres as Its value may be assumed as c / 4.

6. **Attitude or eccentricity ratio**. It is the ratio of the eccentricity to the radial clearance. Mathematically, attitude or eccentricity ratio,

$$\varepsilon = \frac{e}{c_1} = \frac{c_1 - h_0}{c_1} = 1 - \frac{h_0}{c_1} = 1 - \frac{2h_0}{c}$$

Coefficient of Friction for Journal Bearings

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k$$

k = Factor to correct for end leakage. It depends upon the ratio of length to the diameter of the bearing (i.e. l/d). = 0.002 for l/d ratios of 0.75 to 2.8.

Critical pressure or minimum operating pressure

$$p = \frac{ZN}{4.75 \times 10^6}\left(\frac{d}{c}\right)^2\left(\frac{l}{d+l}\right) \text{N/mm}^2$$

Heat Generated in a Journal Bearing The heat generated in a bearing is due to the fluid friction and friction of the parts having relative motion. Mathematically, heat generated in a bearing,

Qg = μ.W.V N-m/s or J/s or watts

μ = Coefficient of friction,

W = Load on the bearing in N,

= Pressure on the bearing in N/mm2 × Projected area of the bearing in mm2 = p (l × d),

V = Rubbing velocity in m/s =

$= \frac{\pi d N}{60}$ d is in metres, and

N = Speed of the journal in r.p.m.

Design Procedure for Journal Bearing

The following procedure may be adopted in designing journal bearings, when the bearing load, the diameter and the speed of the shaft

1. Determine the bearing length by choosing a ratio of l/d
2. Check the bearing pressure, p = W / l.d
3. its operating temperature (t0). This temperature should be between 26.5°C and 60°C with 82°C as a maximum for high temperature installations such as steam turbines.
4. Determine the operating value of ZN / p for the assumed bearing temperature to determine the possibility of maintaining fluid film operation.
5. Assume a clearance ratio c/d
6. Determine the coefficient of friction (μ) by using the relation
7. Determine the heat generated by using the relation
8. Determine the heat dissipated by using the relation
9. Determine the thermal equilibrium to see that the heat dissipated becomes atleast equal to the heat generated

Solid Journal Bearing

A solid bearing, is the simplest form of journal bearing. It is simply a block of cast iron with a hole for a shaft providing running fit. The lower portion of the block is extended to form a base plate or sole with two holes to receive bolts for fastening it to the frame

Bushed Bearing is an improved solid bearing in which a bush of brass or gun metal is provided. The outside of the bush is a driving fit in the hole of the casting whereas the inside is a running fit for the shaft. When the bush gets worn out, it can be easily replaced

Figure 108.Solid journal bearing and Bushed bearing.

Split Bearing or Plummer Block A split-bearing is used for shafts running at high speeds and carrying heavy loads. A splitbearing, as shown in Fig. consists of a cast iron base (also called block or pedestal), gunmetal or phosphor bronze brasses, bushes or steps made in two-halves and a cast iron cap.

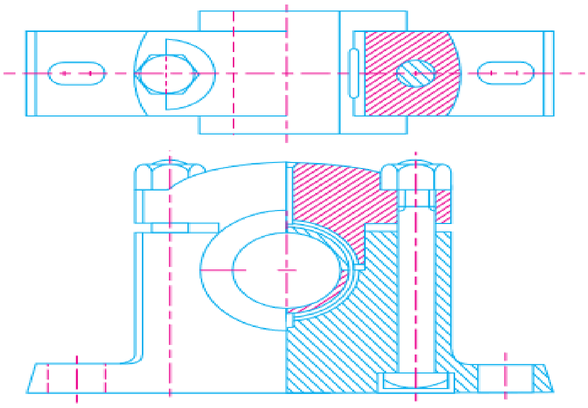


Figure 109.Split bearing or plummer block

Design of Bearing Caps and Bolts

The cap is generally regarded as a simply supported beam, supported by holding down bolts and loaded at the centre.

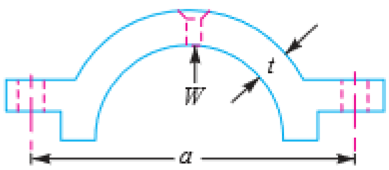


Figure 110.Bearing cap

W = Load supported at the centre,

α = Distance between centres of holding down bolts,

l = Length of the bearing, and

t = Thickness of the cap.

We know that maximum bending moment at the centre, M = W.a /4 and the section modulus of the cap, Z = l.t2/6

Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{W.a}{4} \times \frac{6}{l.t^2} = \frac{3W.a}{2l.t^2}$$

$$t = \sqrt{\frac{3\,W.a}{2\sigma_b.l}}$$

The cap of the bearing should also be investigated for the stiffness. We know that for a simply supported beam loaded at the centre, the deflection,

$$\delta = \frac{W.a^3}{48\,E.I} = \frac{W.a^3}{48\,E \times \frac{l.t^3}{12}} = \frac{W.a^3}{4\,E.l.t^3}$$

$$t = 0.63\,a\left[\frac{W}{E.l.\delta}\right]^{1/3}$$

normal load on each bolt. In other words, load on each bolt is taken $\frac{4\,W}{3n}$, where n is the number of bolts used for holding down the cap.

Let dc = Core diameter of the bolt, and

σt = Tensile stress for the material of the bolt

$$\frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{4}{3} \times \frac{W}{n}$$

Thrust Bearings

A thrust bearing is used to guide or support the shaft which is subjected to a load along the axis of the shaft. Such type of bearings is mainly used in turbines and propeller shafts. The thrust bearings are of the following two types:

1. Foot step or pivot bearings, and 2. Collar bearings.

1.2. Rolling Contact Bearings

In rolling contact bearings, the contact between the bearing surfaces is rolling instead of sliding as in sliding contact bearings**.** Due to this low friction offered by rolling contact bearings, these are called antifriction bearings.

Advantages and Disadvantages of Rolling Contact Bearings Over Sliding Contact Bearings

1. Low starting and running friction except at very high speeds.
2. Ability to withstand momentary shock loads.
3. Accuracy of shaft alignment.
4. Low cost of maintenance, as no lubrication is required while in service.
5. Small overall dimensions.
6. Reliability of service.
7. Easy to mount and erect.
8. Cleanliness.

Disadvantages

1. Noisier at very high speeds.

2. Low resistance to shock loading.
3. More initial cost.
4. Design of bearing housing complicated.

Types of Rolling Contact Bearings Following are the two types of rolling contact bearings:

1. Ball bearings; and 2. Roller bearings.

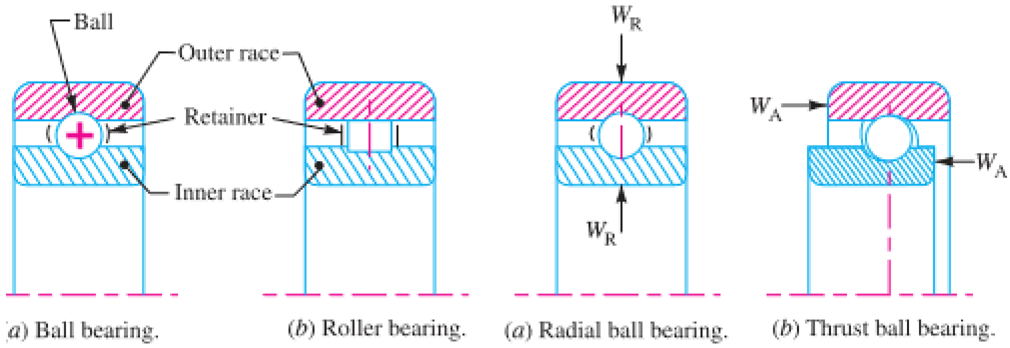


Figure 111.Ball and roller bearings and Radial and thrust ball bearings.

The ball and roller bearings consist of an inner race which is mounted on the shaft or journal and an outer race which is carried by the housing or casing. In between the inner and outer race, there are balls or rollers.

The radial and thrust ball bearings are (a) and (b) respectively. When a ball bearing supports only a radial load (WR), the plane of rotation of the ball is normal to the centre line of the bearing, (a). The action of thrust load (WA) is to shift the plane of rotation of the balls, the radial and thrust loads both may be carried simultaneously

Types of Radial Ball Bearings

Following are the various types of radial ball bearings**:**

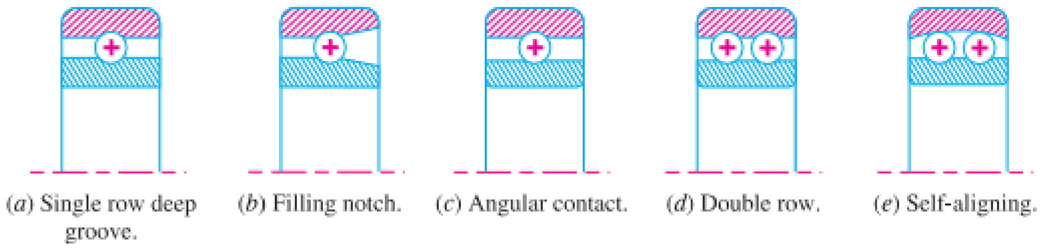
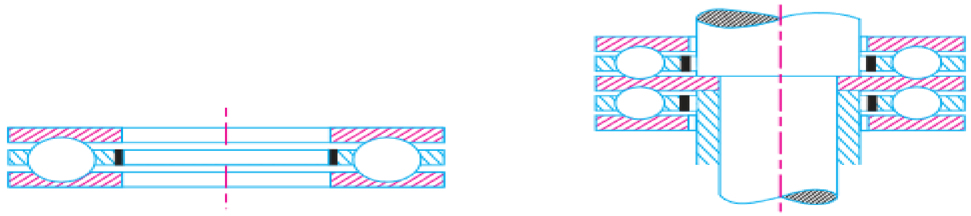


Figure 112.Types of radial ball bearings

Thrust Ball Bearings: are used for carrying thrust loads exclusively and at speeds below 2000 r.p.m. At high speeds, centrifugal force causes the balls to be forced out of the races. Therefore, at high speeds, it is recommended that angular contact ball bearings should be used in place of thrust ball bearings.



(a) Single direction thrust ball bearing. (b) Double direction thrust ball bearing.

Figure 113. Thrust Ball Bearings

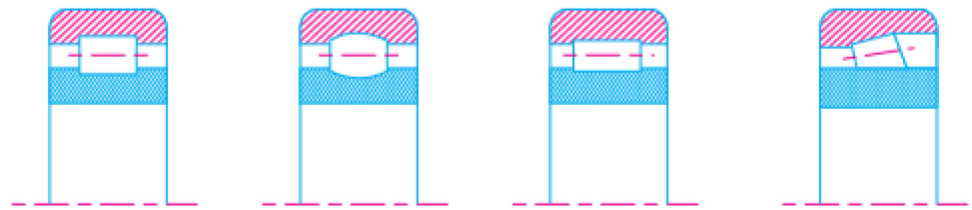
Types roller bearing

1.A cylindrical roller bearing. These bearings have short rollers guided in a cage. These bearings are relatively rigid against radial motions of Roller Bearing sand have the lowest coefficient of friction of any form of heavy duty rolling-contact bearings.

2.**Spherical roller bearings**. These bearings are self-aligning bearings. The self-aligning feature is achieved by grinding one of the races in the form of sphere.

3.**Needle roller bearings.** These bearings are relatively slender and completely fill the space so that neither a cage nor a retainer is needed. These bearings are used when heavy loads are to be carried with an oscillatory motion.

4.**Tapered roller bearings**. The rollers and race ways of these bearings are truncated cones whose elements intersect at a common point. Such type of bearings can carry both radial and thrust loads



(a) Cylindrical roller. (b) Spherical roller. (c) Needle roller. (d) Tapered roller.

Figure 114.Types of roller bearings.

S.No.	Type of bearing	Single row bearing		Double row bearing	
		X_0	Y_0	X_0	Y_0
1.	Radial contact groove ball bearings	0.60	0.50	0.60	0.50
2.	Self aligning ball or roller bearings and tapered roller bearing	0.50	0.22 cot θ	1	0.44 cot θ
3.	Angular contact groove bearings :				
	$\alpha = 15°$	0.50	0.46	1	0.92
	$\alpha = 20°$	0.50	0.42	1	0.84
	$\alpha = 25°$	0.50	0.38	1	0.76
	$\alpha = 30°$	0.50	0.33	1	0.66
	$\alpha = 35°$	0.50	0.29	1	0.58
	$\alpha = 40°$	0.50	0.26	1	0.52
	$\alpha = 45°$	0.50	0.22	1	0.44

Dynamic Load Rating for Rolling Contact Bearings under Variable Loads

The approximate rating (or service) life of ball or roller bearings is based on the fundamental equation**.**

$$L = \left(\frac{C}{W}\right)^k \times 10^6 \text{ revolutions}$$

$$C = W\left(\frac{L}{10^6}\right)^{1/k}$$

L = Rating life,

C = Basic dynamic load rating,

W = Equivalent dynamic load, and

k = 3, for ball bearings,

= 10/3, for roller bearings.

The relationship between the life in revolutions (L) and the life in working hours (LH) is given by

L = 60 N. LH revolutions

Where N is the speed in r.p.m, now consider a rolling contact bearing subject to variable load. let w1. w2, w3 etc.be the load on the bearing for successive n1 n2, n3 etc. number of revolution respectively.

If the bearing is operated exclusively at the constant load W1, then its life given by

$$L_1 = \left(\frac{C}{W_1}\right)^k \times 10^6 \text{ revolutions}$$

$\therefore$ Fraction of life consumed with load W_1 acting for n_1 number of revolutions is

$$\frac{n_1}{L_1} = n_1 \left(\frac{W_1}{C}\right)^k \times \frac{1}{10^6}$$

Similarly, fraction of life consumed with load W_2 acting for n_2 number of revolutions is

$$\frac{n_2}{L_2} = n_2 \left(\frac{W_2}{C}\right)^k \times \frac{1}{10^6}$$

and fraction of life consumed with load W_3 acting for n_3 number of revolutions is

$$\frac{n_3}{L_3} = n_3 \left(\frac{W_3}{C}\right)^k \times \frac{1}{10^6}$$

But $\quad \dfrac{n_1}{L_1} + \dfrac{n_2}{L_2} + \dfrac{n_3}{L_3} + \ldots = 1$

or $\quad n_1\left(\dfrac{W_1}{C}\right)^k \times \dfrac{1}{10^6} + n_2\left(\dfrac{W_2}{C}\right)^k \times \dfrac{1}{10^6} + n_3\left(\dfrac{W_3}{C}\right)^k \times \dfrac{1}{10^6} + \ldots\ldots = 1$

$\therefore n_1(W_1)^k + n_2(W_2)^k + n_3(W_3)^k + \ldots\ldots\ldots = C^k \times 10^6$

If an equivalent constant load (W) is acting for n number of revolutions, then

$$n = \left(\frac{C}{W}\right)^k \times 10^6$$

$$\therefore \qquad n(W)^k = C^k \times 10^6$$

where $n = n_1 + n_2 + n_3 + ...$

From equations (*i*) and (*ii*), we have

$$n_1 (W_1)^k + n_2 (W_2)^k + n_3 (W_3)^k + ... = n (W)^k$$

$$\therefore \qquad W = \left[\frac{n_1 (W_1)^k + n_2 (W_2)^k + n_3 (W_3)^k + ...}{n}\right]^{1/k}$$

Substituting $n = n_1 + n_2 + n_3 + ...$, and $k = 3$ for ball bearings, we have

$$W = \left[\frac{n_1 (W_1)^3 + n_2 (W_2)^3 + n_3 (W_3)^3 + ...}{n_1 + n_2 + n_3 + ...}\right]^{1/3}$$

Note : The above expression may also be written as

$$W = \left[\frac{L_1 (W_1)^3 + L_2 (W_2)^3 + L_3 (W_3)^3 + ...}{L_1 + L_2 + L_3 + ...}\right]^{1/3}$$

Reliability of a Bearing

the relation between the bearing life and the reliability is given as

$$\log_e \left(\frac{1}{R}\right) = \left(\frac{L}{a}\right)^b \qquad \text{or} \qquad \frac{L}{a} = \left[\log_e \left(\frac{1}{R}\right)\right]^{1/b} \qquad ...(i)$$

where L is the life of the bearing corresponding to the desired reliability R and a and b are constants whose values are

$$a = 6.84, \quad \text{and} \quad b = 1.17$$

If L_{90} is the life of a bearing corresponding to a reliability of 90% (*i.e.* R_{90}), then

$$\frac{L_{90}}{a} = \left[\log_e \left(\frac{1}{R_{90}}\right)\right]^{1/b} \qquad ...(ii)$$

Dividing equation (*i*) by equation (*ii*), we have

$$\frac{L}{L_{90}} = \left[\frac{\log_e (1/R)}{\log_e (1/R_{90})}\right]^{1/b} = {}^*6.85 \left[\log_e (1/R)\right]^{1/1.17} \qquad ... (\because b = 1.17)$$

This expression is used for selecting the bearing when the reliability is other than 90%.

Note : If there are n number of bearings in the system each having the same reliability R, then the reliability of the complete system will be

$$R_S = R_p$$

where R_S indicates the probability of one out of p number of bearings failing during its life time.

---

* $[\log_e (1 / R_{90})]^{1/b} = [\log_e (1/0.90)]^{1/1.17} = (0.10536)^{0.8547} = 0.146$

$$\therefore \qquad \frac{L}{L_{90}} = \frac{[\log_e (1/R)]^{1/b}}{0.146} = 6.85 \left[\log_e (1/R)\right]^{1/1.17}$$

Materials and Manufacture of Ball and Roller Bearings

The balls are generally made of high carbon chromium steel. The material of both the balls and races are heat treated to give extra hardness and toughness

Life of a Bearing: The life of an individual ball (or roller) bearing may be defined as the number of revolutions (or hours at some given constant speed) which the bearing runs before the first evidence of fatigue develops in the material of one of the rings or any of the rolling elements

Lubrication of Ball and Roller Bearings

The ball and roller bearings are lubricated for the following purposes:

- To reduce friction and wear between the sliding parts of the bearing
- To prevent rusting or corrosion of the bearing surface
- To protect the bearing surfaces from water, dirt etc., and
- To dissipate the heat

2. Brackets

Brackets are used to support beams, conduits, pipes etc. When the roofing work is finished for a portal structure, the overhang of the sheets is supported by brackets, the louvres which are essential for ventilation in a shed system are supported by brackets. The railings provided around a Walkway are supported by brackets. The typical cross-section of a bracket is channel. The best example of a brackets is the catenary support system used by railways.

Choice of frames/brackets

- Lightweight
- Easy to process
- Easy to assembly
- Cost effective

Metal brackets:

- stainless steel bracket
- gold coated bracket
- titanium bracket

plastic bracket:

- polycarbonate bracket
- polyurethane composite bracket
- thermoplastic polyurethane bracket

ceramic bracket

- monocrystalline alumina
- polycrystalline alumina
- polycrystalline zirconia

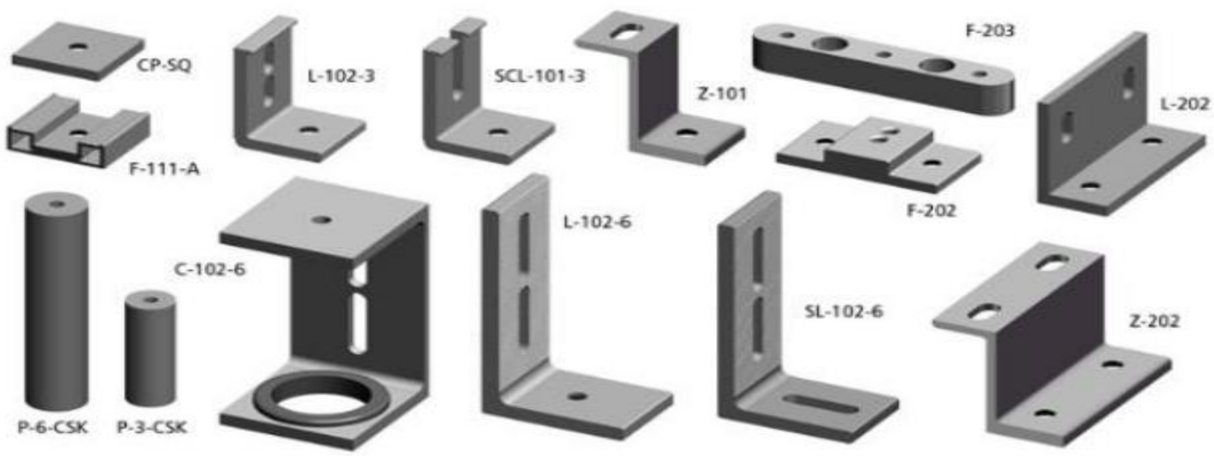
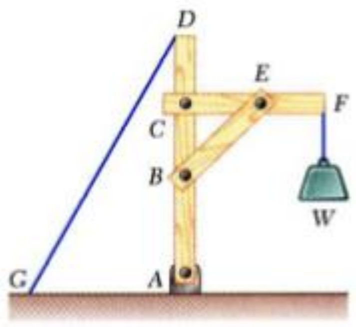


Figure 115.Brackets

3. Frames

Frames are structures with at least one multi-force membered. at least one member that has 3 or more forces acting on it at different points.

Frame analysis involves determining



i)External Reactions (ii) Internal forces at the joints

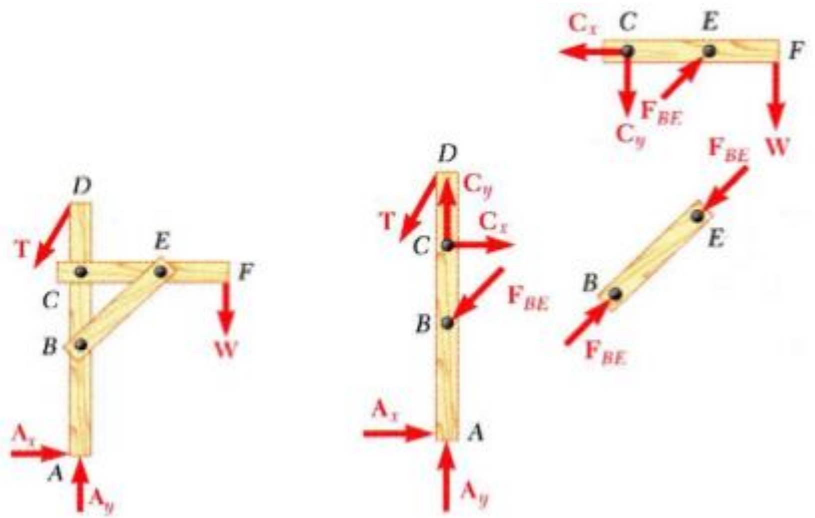
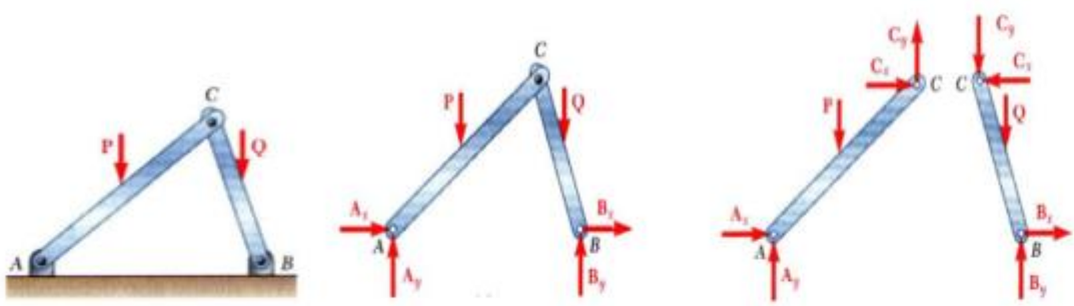


Figure 116.Frame analysis involves determining

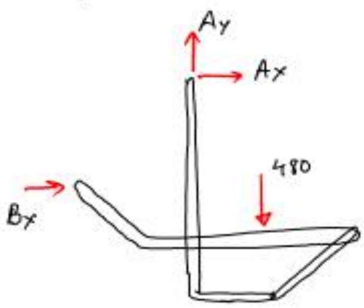
Note: Follow Newton's 3rd Law

Frames that are not internally Rigid

When a frame is not internally rigid, it has to be provided with additional external supports to make it rigid. The support reactions for such frames cannot be simply determined by external equilibrium. One has to draw the FBD of all the component parts to find out whether the frame is determinate or indeterminate.



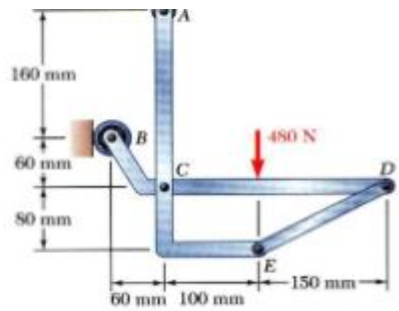
Examples:

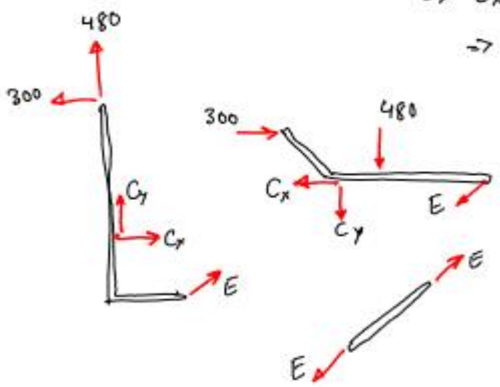


$\Sigma F_x = 0$
$\Rightarrow A_x + B_x = 0$

$\Sigma F_y = 0$
$A_y = 480$

$\Sigma M_A = 0$
$\Rightarrow -480 \times 0.1$
$+ B_x \times 0.16 = 0$
$\Rightarrow B_x = \frac{48}{0.16} = 300N$
$\Rightarrow A_x = -300N$





(R.G. Kirk, R. Gao, 2012)(Sir Robert Stawell Ball ,Leonard Euler,James Watt, Alexander , 2008)(wiley,WeiJiang, 2019)

Learning Unit 2 -analyses machine elements arrangement

L O 2.1 -Identify types of machine structure components

- Content/Topic1: Machine components structures

1.1. Frame

Machines are structures designed to transmit and modify forces. Their main purpose is to transform input forces into output forces.

Machines are usually non-rigid internally. So we use the components of the machine as a free-body. Given the magnitude of P, determine the magnitude of Q.

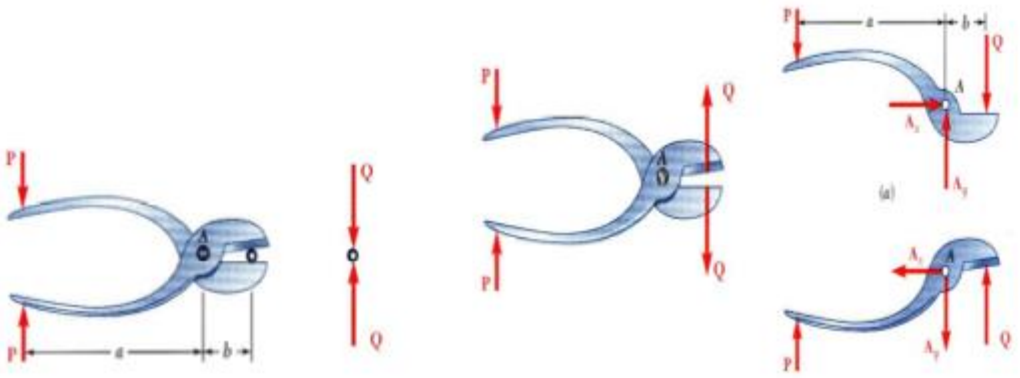


Figure 117.Frame

Example: The tongs show is used to apply a total upward force of 45 KN on a pipe cap. determine the force exerted at D and F on tong ADF

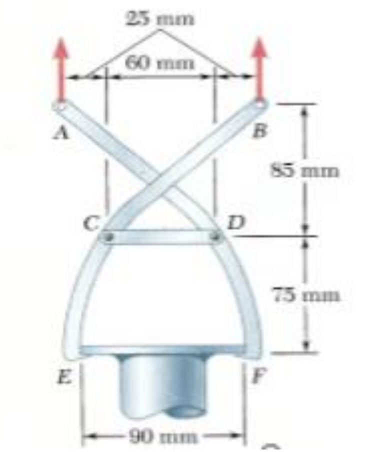
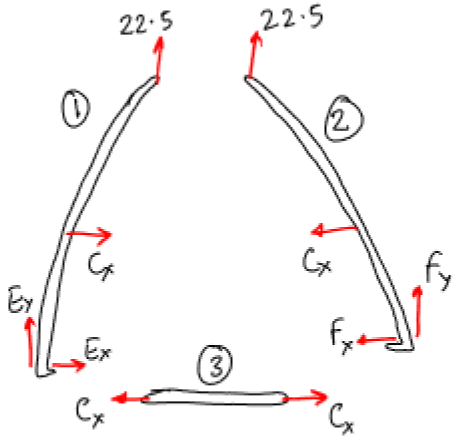


Figure 118.A tongs



(2)

$\sum M_F = 0$

$\Rightarrow -22.5 \times 100 + C_x \times 75 = 0$

$\Rightarrow C_x = 22.5 \times \frac{4}{3} = \boxed{30\ \text{kN}}$ $\Rightarrow \sum F_x = 0 \Rightarrow -C_x - F_x \Rightarrow \boxed{F_x = -30\ \text{kN}}$

$\sum F_y \Rightarrow \boxed{F_y = -22.5\ \text{kN}}$

Determinate vs. Indeterminate Structures

Structures such as Trusses and Frames can be broadly classified as:

Determinate

When all the unknowns (external reactions and internal forces) can be found using "Statics" i.e. Drawing FBDs and writing equilibrium equations.

Indeterminate

When, not all the unknowns can be found using Statics.

Note: Some/most unknowns can still be found.

Structures can also be classified as:

- Completely restrained
- Partially restrained
- Improperly restrained

For trusses, we have been using "formulas" such as (2n = m+r) for planar trusses, and (3n = m+r) for space trusses to judge the type of structure. For frames, this can be much more complicated. We need to write and solve the equilibrium equations and only if a solution exists, we can conclude that the structure is determinate. Otherwise the structure may be partially constrained or indeterminate or both.

1.2. Shell

Shells and space structures are very attractive and have been constructed to solve a large variety of functional problems (roofs, industrial building, aqueducts, reservoirs, footings etc). In this type of stmctures aesthetics, structural efficiency and concept play a very important role. This class of structures can be divided into three main groups, namely continuous (concrete) shells, space frames and tension (fabric, pneumatic, cable etc) structures.

- Content/Topic2:Mechanical linkage systems

1. **External linkage**

External Linkage: An identifier implementing external linkage is visible to every translation unit. Externally linked identifiers are *shared* between translation units and are considered to be located at the outermost level of the program. In practice, this means that you must define an identifier in a place which is visible to all, such that it has only one visible definition. It is the default linkage for globally scoped variables and functions. Thus, all instances of a particular identifier with external linkage refer to the same identifier in the program. The keyword extern implements external linkage. When we use the keyword extern, we tell the linker to look for the definition elsewhere. Thus, the declaration of an externally linked identifier does not take up any space. Extern identifiers are generally stored in initialized/uninitialized or text segment of RAM.

2. **Internal linkage**

Internal Linkage: An identifier implementing internal linkage is not accessible outside the translation unit it is declared in. Any identifier within the unit can access an identifier having internal linkage. It is implemented by the keyword static. An internally linked identifier is stored in initialized or uninitialized segment of RAM. (**note:** static also has a meaning in reference to scope,

L O 2.2 - Calculate power transmission of working mechanisms

- Content/Topic1: Power transmission calculations

1. **Gear drives calculations**

Gears are defined as toothed wheels or multilobed cams, which transmit power and motion from one shaft to another by means of successive engagement of teeth. Gear drives offer the following advantages compared with chain or belt drives:

Gear terminology consists of:

(i) ***Pinion*** A pinion is the smaller of the two mating gears

(ii) ***Gear*** A gear is the larger of the two mating gears.

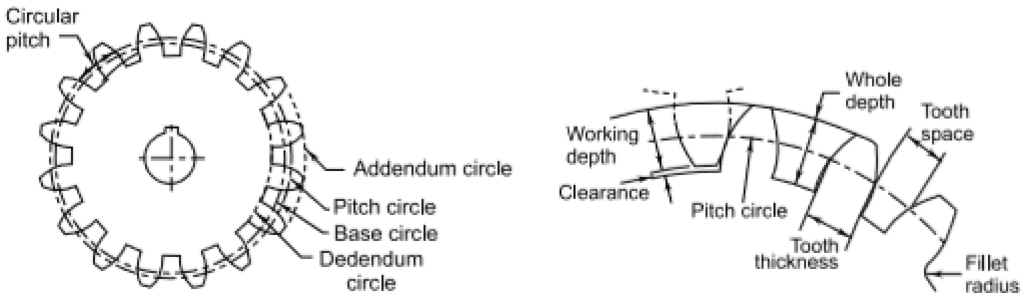


Figure 119.terminology of gears

(ii) ***Velocity Ratio (i)*** Velocity ratio is the ratio of angular velocity of the driving gear to the angular velocity of the driven gear. It is also called the speed ratio.

(iii) ***Transmission Ratio (i')*** The transmission ratio (i') is the ratio of the angular speed of the fi rst driving gear to the angular speed of the last driven gear in a gear train.

(iv) ***Pitch Surface*** The pitch surfaces of the gears are imaginary planes, cylinders or cones that roll together without slipping.

Pitch Circle The pitch circle is the curve of intersection of the pitch surface of revolution

(v) ***Pitch Circle Diameter*** The pitch circle diameter is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also called *pitch diameter*. The pitch circle diameter is denoted by d ⍰.

(vi) ***Pitch Point*** The pitch point is a point on the line of centres of two gears at which two pitch circles of mating gears are tangent to each other.

(vii) ***Top Pand***The top land is the surface of the top of the gear tooth.

(viii) ***Bottom Pand***The bottom land is the surface of the gear between the flanks of adjacent teeth.

(ix) ***Involute*** An involute is a curve traced by a point on a line as the line rolls without slipping on a circle.

(x) ***Base Circle*** The base circle is an imaginary circle from which the involute curve of the tooth profi le is generated. The base circles of two mating gears are tangent to the pressure line.

(xi) ***Addendum Circle*** The addendum circle is an imaginary circle that borders the tops of gear teeth in the cross section.

(xii) ***Addendum (h_a)*** The addendum (h_a) is the radial distance between the pitch and the addendum circles. Addendum indicates the height of the tooth above the pitch circle.

(xiii) ***Dedendum Circle*** The dedendum circle is an imaginary circle that borders the bottom of spaces between teeth in the cross section. It is also called *root* circle.

(xiv) ***Dedendum (h_f)*** The dedendum (h_f) is the radial distance between pitch and the dedendum circles. The dedendum indicates the depth of the tooth below the pitch circle.

(xv) ***Clearance (c)*** The clearance is the amount by which the dedendum of a given gear exceeds the addendum of its mating tooth.

(xvi) ***Face of Tooth*** The surface of the gear tooth between the pitch cylinder and the addendum cylinder is called the face of tooth.

(xvii) ***Flank of Tooth*** The surface of the gear tooth between the pitch cylinder and the root cylinder is called flank of the tooth.

(xviii) ***Face Width (b)*** Face width is the width of the tooth measured parallel to the axis.

(xix) ***Fillet Radius*** The radius that connects the root circle to the profi le of the tooth is called fi llet radius.

(xx) ***Circular Tooth Thickness*** The length of the arc on the pitch circle subtending a single gear tooth is called circular tooth thickness. Theoretically, circular tooth thickness is half of the circular pitch.

(xxi) ***Tooth Space*** The width of the space between two adjacent teeth measured along the pitch circle is called the tooth space. Theoretically, tooth space is equal to circular tooth thickness or half the circular pitch.

(xxii) ***Working Depth (h_k)*** The working depth is the depth of engagement of two gear teeth, that is, the sum of their addendums.

(xxiii) ***Whole Depth (h)*** The whole depth is the total depth of the tooth space, that is, the sum of the addendum and dedendum. Whole depth is also equal to working depth plus clearance.

(xxiv) ***Centre Distance*** The centre distance is the distance between centres of pitch circles of mating gears. It is also the distance between centres of base circles of mating gears.

(xxv) ***Pressure Angle*** The pressure angle is the angle which the line of action makes with the common tangent to the pitch circles. The pressure angle is also called the angle of obliquity.

(xxvi) ***Line of Action*** The line of action is the common tangent to the base circles of mating gears. The contact between the involute surfaces of mating teeth must be on this line to give a smooth operation. The force is transmitted from the driving gear to the driven gear on this line.

(xxvii) ***Arc of Contact*** The arc of contact is the arc

of the pitch circle through which a tooth moves from the beginning to the end of contact with mating tooth.

(xxviii) ***Arc of Approach*** The arc of approach is the arc of the pitch circle through which a tooth moves from its beginning of contact until the point of contact arrives at the pitch point.

(xxix) ***Arc of Recess*** The arc of recess is the arc of the pitch circle through which a tooth moves from the contact at the pitch point until the contact ends.

(xxx) ***Contact Ratio (m_p)*** The number of pairs of teeth that are simultaneously engaged is called contact ratio. If there are two pairs of teeth in contact all the time, the contact ratio is 2. As the two gears rotate, smooth and continuous transfer of power from one pair of meshing teeth to the following pair is achieved when the contact of the fi rst pair continues until the following pair has established contact. Some overlapping is essential for this purpose. Therefore, the contact ratio is always more than 1. Other things being, the greater the contact ratio, the smoother the action of gears. The contact ratio for smooth transfer of motion is usually taken as 1.2. In industrial gearboxes for power transmission, the contact ratio is usually more than 1.4 (1.6 to 1.7).

Circular Pitch The circular pitch (p) is the distance measured along the pitch circle between two similar points on adjacent teeth.

(xxxi) ***Diametral Pitch*** The diametral pitch (P) is the ratio of the number of teeth to the pitch circle diameter. Therefore,

$$P = \frac{z}{d'}$$

$$P \times p = \pi$$

Module The module (m) is defined as the inverse of the diametral pitch. Therefore

$$m = \frac{1}{P} = \frac{d'}{z}$$

$$d' = mz$$

The centre to centre distance between two gears having $z_p$and z_g

$$a = \frac{1}{2}(d'_p + d'_g) = \frac{1}{2}(mz_p + mz_g)$$

$$a = \frac{m(z_p + z_g)}{2} \qquad (17$$

where,

a = centre to centre distance (mm)

z_p = number of teeth on pinion

z_g = number of teeth on gear

The gear ratio (i) that is, the ratio of the number of teeth on gear to that on pinion is given by,

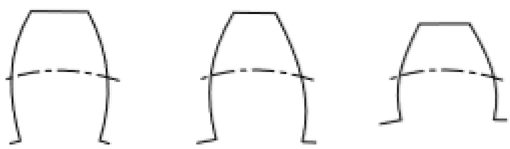
$$i = \frac{n_p}{n_g} = \frac{z_g}{z_p}$$

where n_p = speed of pinion (rpm)

n_g = speed of gear (rpm)

There are three standard systems for the shape of gear teeth. They are as follows:

- 14.5° full depth involute system
- 20° full depth involute systems
- 20° stub involute system



(a) 14.5° full depth (b) 20° full depth (c) 20° stub

Figure 120.standard tooth profile

	14.5° full depth system	*20° full depth system*	*20° stub system*
Pressure angle	14.5°	20°	20°
Addendum	m	m	0.8 m
Dedendum	1.157 m	1.25 m	m
Clearance	0.157 m	0.25 m	0.2 m
Working depth	2 m	2 m	1.6 m
Whole depth	2.157 m	2.25 m	1.8 m
Tooth thickness	1.5708 m	1.5708 m	1.5708 m

Table*. Proportions of standard involute teeth (in terms of module m)*

The standard proportions of the gear tooth in terms of module *m*, for 20° full depth system are rewritten here.

Addendum (h_a) = (m)

Dedendum (h_f) = (1.25 m)

Clearance (c) = (0.25 m)

Working depth (h_k) = (2 m)

Whole depth (h) = (2.25 m)

Tooth thickness (s) = (1.5708 m)

Tooth space = (1.5708 m)

Fillet radius = (0.4 m)

Example Question:

A pair of spur gears consists of a 20 teeth pinion meshing with a 120 teeth gear. The module is 4 mm. Calculate

(i) the centre distance;

(ii) the pitch circle diameters of the pinion and the gear;

(iii) the addendum and dedendum;

(iv) the tooth thickness;

(v) (v) the bottom clearance; and

(vi) (vi) the gear ratio.

Solution: Given z_p = 20 z_g = 120 m = 4 mm

Step I *Centre distance*

Step I *Centre distance*

From Eq. (17.5),

$$a = \frac{m(z_p + z_g)}{2} = \frac{4(20+120)}{2} = 280 \text{ mm}$$

Step II *Pitch circle diameters of pinion and gear*

$$d'_p = m\, z_p = 4(20) = 80 \text{ mm}$$

$$d'_g = m\, z_g = 4(120) = 480 \text{ mm}$$

Step III *Addendum and dedendum*

$$\text{addendum } (h_a) = \text{m} = 4 \text{ mm}$$

$$\text{dedendum } (h_f) = 1.25 \text{ m} = 1.25(4) = 5 \text{ mm}$$

Step IV *Tooth thickness*

$$\text{tooth thickness} = 1.5708 \text{ m} = 1.5708(4) = 6.2832 \text{ mm}$$

Step V *Bottom clearance*

$$\text{clearance } (c) = 0.25 \text{ m} = 0.25(4) = 1 \text{ mm}$$

Step VI *Gear ratio*

$$i = \frac{z_g}{z_p} = \frac{120}{20} = 6$$

2. Pulley-Belt drives calculations

PULLEYS FOR FLAT BELTS

The pulleys for flat belts consist of three parts, viz., rim, hub and arms or web. The rim carries the belt. The hub connects the pulley to the shaft. The arms or web join the hub with the rim. There are two types of pulleys that are used for flat belts, cast iron pulleys and mild steel pulleys.

No. of plies	*Maximum belt speed (m/s)*				
	10	*15*	*20*	*25*	*30*
3	90	100	112	140	180
4	140	160	180	200	250
5	200	224	250	315	355
6	250	315	355	400	450
7	355	400	450	500	560
8	450	500	560	630	710
9	560	630	710	800	900
10	630	710	800	900	1000

Table: Minimum pulley diameters for given belt speeds and belt plies (mm)

Nominal diameter (mm)	Tolerance on diameter (mm)
40	± 0.5
45, 50	± 0.6
56, 63	± 0.8
71, 80	± 1.0
90, 100, 112	± 1.2
125, 140	± 1.6
160, 180, 200	± 2.0
224, 250	± 2.5
280, 315, 355	± 3.2
400, 450, 500	± 4.0
560, 630, 710	± 5.0
800, 900, 1000	± 6.3
1120, 1250, 1400	± 8.0
1600, 1800, 2000	± 10.0

Table13: Recommended diameters of cast iron

cast iron *and mild steel flat pulleys*

There is a specific term, '*crowning*' of pulley in flat belt drive. The thickness of the rim is slightly increased in the center to give it a convex or conical shape This is called '*crown*' of the pulley. The crown is provided only on one of the two pulleys.

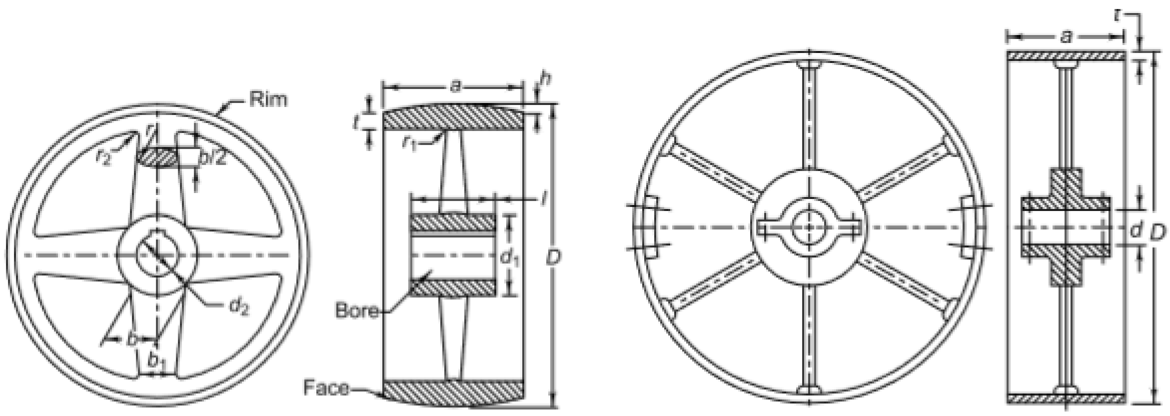


Figure 121.cast iron and mild steel pulleys

(b) *Cross-section of arms-elliptical*

(c) *Thickness of arm b near boss*

$$b = 2.94\sqrt[3]{\frac{aD}{4n}} \text{ for single belt}$$

$$b = 2.94\sqrt[3]{\frac{aD}{2n}} \text{ for double belt}$$

where, a = width of pulley

D = diameter of pulley

n = number of arms in the pulley

(d) *Thickness of arm b_1 near rim* = use taper 4 mm per 100 mm (from boss to rim)

(e) *Radius of the cross-section of arms*

$$r = \frac{3}{4}b$$

(f) *Minimum length l of the bore*

$$l = \frac{2}{3}a$$

It may be more for loose pulleys, but in no case it should exceed a.

(g) $$\frac{d_1 - d_2}{2} = 0.412 \times \sqrt[3]{aD} + 6 \text{ mm for single belt}$$

$$\frac{d_1 - d_2}{2} = 0.529 \times \sqrt[3]{aD} + 6 \text{ mm for double belt}$$

(h) *Radius r_1 and r_2*

$$r_1 = \frac{b}{2} \text{ (near rim)}$$

$$r_2 = \frac{b}{2} \text{ (near rim)}$$

(i) Thickness of rim

$$\text{Rim thickness} = \left(\frac{D}{200} + 3\right)\text{mm (for single belt)}$$

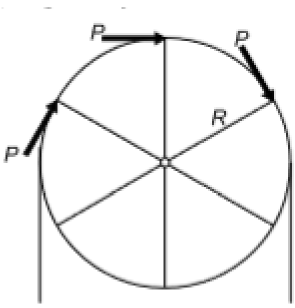
$$\text{Rim thickness} = \left(\frac{D}{200} + 6\right)\text{mm (for double belt)}$$

ARMS OF CAST IRON PULLEY

There are three important things about the arms of the pulley. They are as follows:

- The arms of pulley have elliptical cross section.
- The major axis of elliptical cross-section is in the plane of rotation.
- The major axis of elliptical cross-section is usually twice the minor axis.

The design of these arms illustrates the application of simple formula for bending stresses. It is assumed that the belt wraps around the rim of the pulley through approximately 180° and onehalf of the arms carry the load at any moment. The torque transmitted by the pulley is given by,



cast iron and mild steel pulleys ..

$$M_t = PR\left(\frac{N}{2}\right) \quad \text{or,} \quad P = \frac{2M_t}{RN}$$

M_t= torque transmitted by the pulley (N-mm)

P= tangential force at the end of each arm (N)

R= radius of the rim (mm)

N= number of arms

As shown in Fig. (a), the bending moment acting on the arm is given by

$$M_b = PR \quad M_b = \frac{2M_t}{N}$$

$$I = \frac{\pi ab^3}{64} \qquad \text{(d)}$$

Since the major axis is twice of the minor axis,

$$b = 2a \qquad \text{(e)}$$

Substituting (e) in (d),

$$I = \frac{\pi a^4}{8} \quad \text{and} \quad y = \frac{b}{2} = a$$

The bending stress in the arm is given by,

$$\sigma_b = \frac{M_b y}{I} = \frac{(2M_t/N)(a)}{(\pi a^4/8)}$$

or,

$$a^3 = \frac{16\, M_t}{\pi\, N\, \sigma_b}$$

$$\therefore \qquad a = 1.72\, \sqrt[3]{\frac{M_t}{N\,\sigma_b}} \qquad \text{(f)}$$

where σ_b is the permissible bending stress.

If we consider the minor axis in the plane of rotation as illustrated in Fig. 13.21(b),

$$I = \frac{\pi\, ba^3}{64}$$

Since the major axis is twice of the minor axis,

$$b = 2a$$

$$I = \frac{\pi a^4}{32} \quad \text{and} \quad y = \frac{a}{2}$$

The bending stresses in the arm is given by,

$$\sigma_b = \frac{M_b y}{I} = \frac{(2M_t/N)(a/2)}{(\pi a^4/32)} \quad \text{or,} \quad a^3 = \frac{32M_t}{\pi N\sigma_b}$$

$$\therefore \qquad a = 2.17\sqrt[3]{\frac{M_t}{N\sigma_b}} \qquad ($$

Example Calculation:

A belt pulley, made of greycast iron FG150, has four arms of elliptical crosssection, in which the major axis is twice of the minor axis. The tensions on tight and slack sides of the belt are 750 and 250 N respectively. The mean diameter of the pulley is 300 mm, while the hub diameter 60 is mm. Assume that half the number of arms transmit torque at any time. The factor of safety is 5. Determine the dimensions of the crosssection of the pulley arm near the hub.

Solution

Given $P_1 = 750$ N $P_2 = 250$ N $D = 300$ mm
hub diameter = 60 mm $S_{ut} = 150$ N/mm² $(fs) = 5$
$N = 4$ $b = 2a$

Step I *Permissible stress*

$$\sigma_t = \frac{S_{ut}}{(fs)} = \frac{150}{5} = 30 \text{ N/mm}^2$$

Step II *Bending moment on arm*

As shown in Fig. 13.22,

$$M_t = (P_1 - P_2)(D/2) = (750 - 250)(300/2) = 75000 \text{ N-mm}$$

Assuming that one-half of the arms carry the load at any time, the tangential force P at the end of the arm is given by,

$$M_t = PR\left(\frac{N}{2}\right) \quad \text{or,}$$

$$P = \frac{2M_t}{RN} = \frac{2(75000)}{(150)(4)} = 250 \text{ N}$$

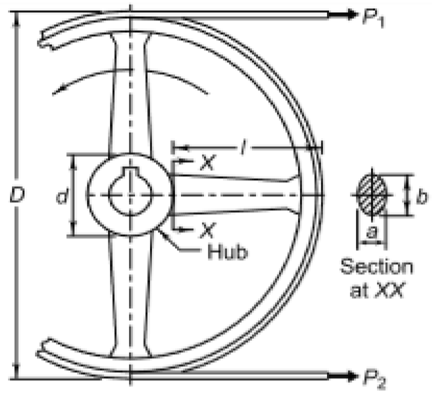


Fig. 13.22

The arm l for the bending moment at the section XX near the hub is given by,

$$l = \frac{D}{2} - \frac{d}{2} = \frac{300}{2} - \frac{60}{2} = 120 \text{ mm}$$

At the section XX,

$$M_b = P \times l = 250 \times 120 = 30000 \text{ N-mm}$$

Step III *Cross-section of arm*

$$b = 2a$$

$$I = \frac{\pi ab^3}{64} = \frac{\pi a(2a)^3}{64} = \frac{\pi a^4}{8} \text{ mm}^4$$

$$y = \frac{b}{2} = \frac{(2a)}{2} = a \text{ mm}$$

$$\sigma_b = \frac{M_b y}{I} \quad \text{or} \quad 30 = \frac{(30000)(a)}{\left(\frac{\pi a^4}{8}\right)}$$

$$a = 13.66 \text{ or } 15 \text{ mm}$$
$$b = 2a = 30 \text{ mm}$$

Question Two

A pulley, made of grey cast iron FG 150, transmits 10 kW of power at 720 rpm. The diameter of the pulley is 500 mm. The pulley has four arms of elliptical cross-section, in which the major axis is twice of the minor axis. Determine the dimensions of the cross-section of the arm, if the factor of safety is

Solution

Given kW = 10 $n = 720$ rpm $D = 500$ mm
For pulley, $S_{ut} = 150\ \text{N/mm}^2$ $(fs) = 5$ $N = 4$
$b = 2a$

Step I *Permissible stress*

$$\sigma_t = \frac{S_{ut}}{(fs)} = \frac{150}{5} = 30\ \text{N/mm}^2$$

Step II *Bending moment on arm*

The torque transmitted by the pulley is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (10)}{2\pi (720)}$$
$$= 132\ 629.12\ \text{N-mm}$$

Assuming that one-half of the arms carry the load at any instant, the tangential force at the end of each arm is given by

$$P = \frac{M_t}{R\left(\frac{N}{2}\right)} = \frac{132\ 629.12}{(250)\left(\frac{4}{2}\right)} = 265.26\ \text{N}$$

$$\therefore\ M_b = P \times R = 265.26 \times 250 = 66\ 314.56\ \text{N-mm}$$

Step III *Cross-section of arm*

For an elliptical cross-section with a and b as minor and major axes respectively,

$$I = \frac{\pi a b^3}{64} \quad \text{and} \quad b = 2a$$

$$I = \frac{\pi a (2a)^3}{64} = \frac{\pi a^4}{8}\ \text{mm}^2 \quad \text{and} \quad y = \frac{b}{2} = a\ \text{mm}$$

The bending stresses in the arm are given by,

$$\sigma_b = \frac{M_b\, y}{I} \quad \text{or} \quad 30 = \frac{(66\ 314.56)(a)}{\left(\frac{\pi a^4}{8}\right)}$$

$$\therefore \quad a = 17.78 \text{ or } 20\ \text{mm}$$
$$b = 2a = 40\ \text{mm}$$

Alternatively, using Eq. (f),

$$a = 1.72 \sqrt[3]{\frac{M_t}{N \sigma_b}} = 1.72 \sqrt[3]{\frac{132\ 629.12}{4(30)}}$$
$$= 17.78 \text{ or } 20\ \text{mm}$$

Question Three

The following data is given for an
open-type V-belt drive: diameter of driving pulley = 200 mm diameter of driven pulley = 600 mm groove angle for sheaves = 34° mass of belt = 0.5 kg/m maximum permissible tension in belt = 500 N coefficient of friction = 0.2 contact angle for smaller pulley = 157° speed of smaller pulley = 1440 rpm power to be transmitted = 10 kW
How many V-belts should be used, assuming each belt takes its proportional part of the load?

Solution

Given $kW = 10$ $n = 1440$ rpm $D = 600$ mm

$d = 200$ mm $\theta = 34°$ $m = 0.5$ kg/m $f = 0.2$

$\alpha_s = 157°$ allowable belt tension = 500 N

Step I *Belt tensions*

$$\frac{f\alpha}{\sin\left(\frac{\theta}{2}\right)} = \frac{0.2\left(\frac{157}{180}\right)\pi}{\sin\left(\frac{34}{2}\right)} = 1.874$$

$$e^{f\alpha/\sin(\theta/2)} = e^{1.874} = 6.52$$

$$v = \frac{\pi d n}{60 \times 10^3} = \frac{\pi(200)(1440)}{60 \times 10^3} = 15.08 \text{ m/s}$$

$$mv^2 = 0.5(15.08)^2 = 113.70$$

From Eq. (13.7),

$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha/\sin(\theta/2)}$$

$$\therefore \quad \frac{P_1 - 113.70}{P_2 - 113.70} = 6.52$$

$$P_1 - 6.52P_2 + 627.61 = 0 \qquad \text{(i)}$$

From Eq. (13.8),

$$\text{kW} = \frac{(P_1 - P_2)v}{1000} \quad \text{or} \quad 10 = \frac{(P_1 - P_2)(15.08)}{1000}$$

$$P_1 - P_2 - 663.13 = 0$$

Solving Eq. (i) and (ii),

$$P_1 = 896.96 \text{ N} \quad \text{and} \quad P_2 = 233.83 \text{ N}$$

Step II *Number of belts*

$$\text{Number of belts} = \frac{\text{Maximum tension in belt}}{\text{Allowable belt load}} = \frac{896.96}{500} = 1.79 \text{ or } 2 \text{ belts}$$

3. Sprocket-chain drives calculations

Simple calculation:

Question one: single-strand chain No. 12A is used in a mechanical drive. The driving sprocket has 17 teeth and rotates at 1000 rpm. What is the factor of safety used for standard power rating? Neglect centrifugal force acting on the chain.

Solution

Given $z_1 = 17$ $n_1 = 1000$ rpm Chain-12A

Step I Chain tension

The pitch of the chain is given as 19.05 mm

$$v = \frac{z_1 p n_1}{60 \times 10^3} = \frac{17(19.05)(1000)}{60 \times 10^3} = 5.4 \text{ m/s}$$

The kW rating of chain 12A at 1000 rpm is given as 14.32 kW

Therefore, the chain tension P_1 at the rated power is given by,

$$P_1 = \frac{1000(\text{kW})}{v} = \frac{1000(14.32)}{5.4} = 2651.85\text{ N}$$

Step II Factor of safety

the breaking load for the above chain is given as 31 100 N

$$(fs) = \frac{31\,100}{2651.81} = 11.73$$

QUESTION Two: A chain drive is used in a specialpurpose vehicle. The vehicle is run by a 30 kW rotary engine. There is a separate mechanical drive from the engine shaft to the intermediate shaft. The driving sprocket is fixed to this intermediate shaft. The efficiency of the drive between the engine and the intermediate shafts is 90%. The driving sprocket has 17 teeth and it rotates at 300 rpm. The driven sprocket rotates at 100 rpm. Assume moderate shock conditions and select a suitable four-strand chain for this drive.

Given Engine power = 30 kW $\eta = 0.9$ $z_1 = 17$ $n_1 = 300$ rpm $n_2 = 100$ rpm

Step I*kW rating of chain*

The power transmitted to the driving sprocket is given by, $\text{kW} = \eta\,(30) = 0.9\,(30) = 27\text{ kW}$

$K_s = 1.4$

$K_1 = 3.3$

$K_2 = 1.0$

$$\text{kW rating of chain} = \frac{(\text{kW to be transmitted}) \times K_s}{K_1 \times K_2} = \frac{27 \times 1.4}{3.3 \times 1} = 11.45\text{ kW}$$

Step II*Selection of chain*

The required kW rating is 11.45 kW at 300 rpm speed of driving sprocket. Chain No.16A (kW rating = 11.63) is suitable for the above application.

Question 3: It is required to design a chain drive to connect 5 kW, 1400 rpm electric motor to a drilling machine. The speed reduction is 3: 1. The centre distance should be approximately 500 mm.

(i) Select a proper roller chain for the drive.

(ii) Determine the number of chain links.

(iii) Specify the correct centre distance between the axes of sprockets.

Solution

Given kW = 5 $a = 500$ mm $n_1 = 1400$ rpm $i = 3$

Step IkW rating of chain

The number of teeth on the driving sprocket is selected as 21. It is further assumed that the chain is a simple roller chain with only one strand.

the service factor is taken as 1.3 assuming moderate shock conditions. For single strand chain,

$K_1 = 1$

For 21 teeth,

$K_2 = 1.26$

$$\text{kW rating of chain} = \frac{(\text{kW to be transmitted}) \times K_s}{K_1 \times K_2}$$
$$= \frac{5 \times 1.3}{1 \times 1.26} = 5.16 \text{ kW}$$

Step IISelection of chain

the power rating of the chain 8A at 1400 rpm is 5.28 kW. Therefore, the chain number 8A is selected.

Step III Number of chain links

The pitch dimension (*p*) of this chain is 12.70 mm.

$$z_1 = 21 \text{ teeth} \qquad z_2 = iz_1 = 3(21) = 63 \text{ teeth}$$
$$p = 12.70 \text{ mm} \qquad a = 500 \text{ mm}$$

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right)$$
$$= 2\left(\frac{500}{12.70}\right) + \left(\frac{21 + 63}{2}\right)$$
$$+ \left(\frac{63 - 21}{2\pi}\right)^2 \times \left(\frac{12.70}{500}\right)$$

= 121.87 or 122 links

Step IV Correct centre distance

$$\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right] = \left[122 - \left(\frac{21 + 63}{2}\right)\right] = 80$$

$$a = \frac{p}{4}\left\{\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right] + \sqrt{\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right]^2 - 8\left[\frac{z_2 - z_1}{2\pi}\right]^2}\right\}$$

$$= \frac{12.70}{4}\left\{80 + \sqrt{(80)^2 - 8\left[\frac{63 - 21}{2\pi}\right]^2}\right\}$$
$$= 500.81 \text{ mm}$$

4. Friction wheels' calculations

Friction Wheels The motion and power transmitted by gears is kinematically equivalent to that transmitted by frictional wheels or discs. In order to understand how the motion can be transmitted by two toothed wheels, consider two plain circular wheels A and B mounted on shafts. The wheels have sufficient rough surfaces and press against each other as shown in Fig.

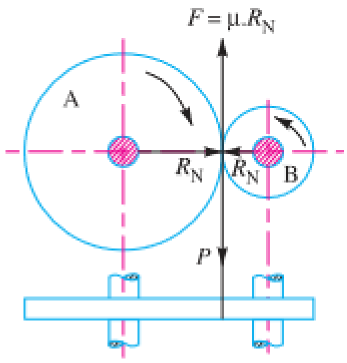


Figure 122.Friction wheels

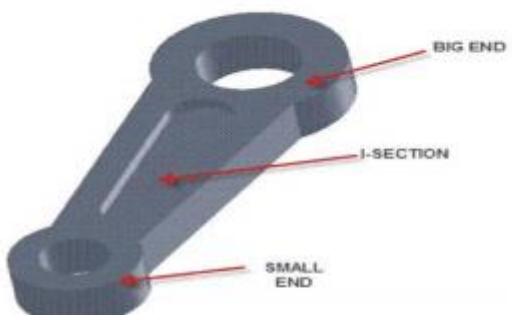
5. connecting rod calculations

A connecting rod, also called a con rod, is the part of a piston engine which connects the piston to the crankshaft. Together with the crank, the connecting rod converts the reciprocating motion of the piston into the rotation of the crankshaft.

DESIGN OF CONNECTING ROD

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile force, therefore the cross section of the connecting rod is designed as a strut and the Rankine formula.

Sno	Parameters (mm)
1	Thickness of the connecting rod (t) = 3.2
2	Width of the section (B = 4t) = 12.8
3	Height of the section(H = 5t) = 16
4	Height at the big end = (1.1 to 1.125)H = 17.6
5	Height at the small end = 0.9H to 0.75H= 14.4
6	Inner diameter of the small end = 17.94
7	Outer diameter of the small end = 31.94
8	Inner diameter of the big end = 23.88
9	Outer diameter of the big end = 47.72



According to rankine formulae Wcr about x-axis

$$= \frac{[\sigma_c \times A]}{1+a[\frac{L}{Kxx}]^2} = \frac{[\sigma_c \times A]}{1+a[\frac{l}{Kxx}]^2}$$

$$[\therefore \, for\ both\ ends\ hinged\ L = l]$$

W_{cr} about y-axis

$$= \frac{[\sigma_c \times A]}{1+a[\frac{L}{Kyy}]^2} = \frac{[\sigma_c \times A]}{1+a[\frac{l}{2Kyy}]^2} \qquad [\therefore \, for\ both\ ends\ fixed\ L = l/2]$$

In order to have a connecting rod equally strong in buckling about both the axis, the buckling loads must be equal. i.e.

$$= \frac{[\sigma_c \times A]}{1+a[\frac{l}{Kxx}]^2} = \frac{[\sigma_c \times A]}{1+a[\frac{l}{2Kyy}]^2} \ [or]$$

$$[\frac{l}{Kxx}]^2 = [\frac{l}{2Kyy}]^2$$

$$K^2_{xx} = 4K^2_{yy} \quad [or] \quad I_{xx} = 4I_{yy} \quad [\therefore I = A \times K^2]$$

This shows that the connecting rod is four times strong in buckling about y-axis than about-axis. If I xx > 4Iyy, then buckling will occur about y-axis and if I xx < 4Iyy, then buckling will occur about x-axis. In Actual practice I xx is kept slightly less than 4Iyy. It is usually taken between 3 and 3.5 and the Connecting rod is designed for buckling about x-axis. The design will always be satisfactory for buckling about y-axis. The most suitable section for the connecting rod is I-section with the proportions shown mfg.

Area of the cross section = 2[4t x t] + 3t x t=11t2

Moment of inertia about x-axis = 2[4txt] +3txt=11t2 Moment of inertia about x-axis

$$I_{xx} = \frac{1}{12}[4t\{5t\}^3 - 3t\{3t\}^3] = \frac{419}{12}[t^4]$$

And moment of inertia about y-axis

$$I_{yy} = \frac{2\times1}{12} \times t \times \{4t\}^3 + \frac{1}{12}\{3t\}t^3 = \frac{131}{12}[t^4]$$

$$I_{xx}/I_{yy} = [419/12]x[12/131] = 3.2$$

Since the value of I xx/I yy lies between 3 and 3.5 m therefore I-section chosen is quite satisfactory.

Design Calculations for Existing Connecting Rod

Thickness of flange & web of the section = t

Width of section B= 4t

The standard dimension of I - SECTION.

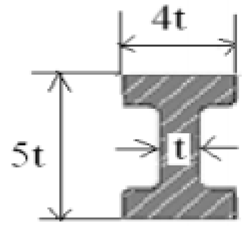


Figure 123.Standard Dimension of I – Section

Height of section H = 5t

Area of section A= 2(4t×t) +3t×t

$A = 11t^2$

M.O.I of section about x axis:

$I_{xx} = \frac{1}{12}[4t\{5t\}^3 - 3t\{3t\}^3]$

$= \frac{419}{12}[t^4]$

MI of section about y axis:

$I_{yy} = \frac{2\times1}{12} \times t \times \{4t\}^3 + \frac{1}{12}\{3t\}t^3$

$= \frac{131}{12}[t^4]$

$\frac{Ixx}{Iyy}$ = **3.2**

Length of connecting rod (L) = 2 times the stroke

L = **117.2 mm**

Buckling load W_B = maximum gas force × F.O.S

$W_B = \frac{(\sigma_c \times A)}{(1+a\,(L/Kxx)2}$

= **37663N**

σ_c = compressive yield stress = **415MPa**

$K_{xx} = \frac{I\,xx}{A}$

Width of section B = 4t

= 4×3.2

= **12.8mm**

Height of section H = 5t

= 5×3.2

= **16mm**

Area A = $11t^2$

=11×3.2×3.2

K_{xx} = 1.78t

$a = \frac{\sigma_c}{\pi^2 E}$

a = **0.0002**

By substituting σ_c, A, a, L, K_{xx} on W_B then

$= 4565t^4 - 37663t^2 - 81639.46 = 0$

$t^2 = 10.03$

t = 3.167mm

t = **3.2mm**

= 112.64mm²

Height at the big end (crank end) = H_2

= 1.1H to 1.25H

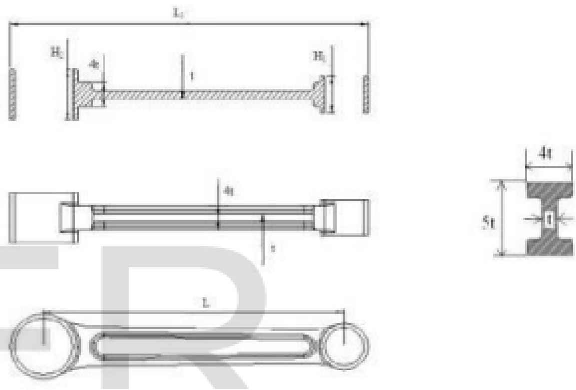
= 1.1×16

H_2 **=17.6mm**

Height at the small end (piston end) = 0.9H to 0.75H

= 0.9×16

H_1 **=12mm**



6. Bearing calculations

Example question one: Design a journal bearing for a centrifugal pump from the following data : Load on the journal = 20 000 N; Speed of the journal = 900 r.p.m.; Type of oil is SAE 10, for which the absolute viscosity at 55°C = 0.017 kg / m-s; Ambient temperature of oil = 15.5°C ; Maximum bearing pressure for the pump = 1.5 N / mm2. Calculate also mass of the lubricating oil required for artificial cooling, if rise of temperature of oil be limited to 10°C. Heat dissipation coefficient = 1232 W/m2/°C.

Solution.

Given: W = 20 000 N ; N = 900 r.p.m. ; t0 = 55°C ; Z = 0.017 kg/m-s ; ta = 15.5°C ; p = 1.5 N/mm2; t = 10°C ; C = 1232 W/m2/°C

The journal bearing is designed as discussed in the following steps: 1. First of all, let us find the length of the journal (l). Assume the diameter of the journal (d) as 100 mm. we find that the ratio of l / d for centrifugal pumps varies from 1 to 2. Let us take l/d = 1.6

l = 1.6 d = 1.6 × 100 = 160 mm Ans.

$$p = \frac{W}{l.d} = \frac{20\ 000}{160 \times 100} = 1.25$$

Since the given bearing pressure for the pump is 1.5 N/mm^2, therefore the above value of p is safe and hence the dimensions of l and d are safe.

3. $$\frac{Z.N}{p} = \frac{0.017 \times 900}{1.25} = 12.24$$

From Table 26.3, we find that the operating value of

$$\frac{Z.N}{p} = 28$$

We have discussed in Art. 26.14, that the minimum value of the bearing modulus at which the oil film will break is given by

$$3\ K = \frac{ZN}{p}$$

∴ Bearing modulus at the minimum point of friction,

$$K = \frac{1}{3}\left(\frac{Z.N}{p}\right) = \frac{1}{3} \times 28 = 9.33$$

Since the calculated value of bearing characteristic number $\left(\frac{Z.N}{p} = 12.24\right)$ is more than 9.33, therefore the bearing will operate under hydrodynamic conditions.

4. From Table 26.3, we find that for centrifugal pumps, the clearance ratio (c/d)

$$= 0.0013$$

5. We know that coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8} \times 12.24 \times \frac{1}{0.0013} + 0.002$$

$$= 0.0031 + 0.002 = 0.0051 \qquad \text{... [From Art. 26.13, } k = 0.002]$$

6. Heat generated,

$$Q_g = \mu\, W\, V = \mu\, W\left(\frac{\pi d.N}{60}\right) \text{W} \qquad \text{...}\left(\because V = \frac{\pi d.N}{60}\right)$$

$$= 0.0051 \times 20000\left(\frac{\pi \times 0.1 \times 900}{60}\right) = 480.7 \text{ W}$$

... (d is taken in metres)

7. Heat dissipated,

$$Q_d = C.A\,(t_b - t_a) = C.l.d\,(t_b - t_a) \text{ W} \qquad \text{... } (\because A = l \times d)$$

We know that

$$(t_b - t_a) = \tfrac{1}{2}\,(t_0 - t_a) = \tfrac{1}{2}\,(55° - 15.5°) = 19.75°\text{C}$$

$$\therefore \qquad Q_d = 1232 \times 0.16 \times 0.1 \times 19.75 = 389.3 \text{ W}$$

We see that the heat generated is greater than the heat dissipated which indicates that the bearing is warming up. Therefore, either the bearing should be redesigned by taking $t_0 = 63°C$ or the bearing should be cooled artificially.

We know that the amount of artificial cooling required

$$= \text{Heat generated} - \text{Heat dissipated} = Q_g - Q_d$$
$$= 480.7 - 389.3 = 91.4 \text{ W}$$

Mass of lubricating oil required for artificial cooling

Let m = Mass of the lubricating oil required for artificial cooling in kg / s.

We know that the heat taken away by the oil,

$$Q_t = m.S.t = m \times 1900 \times 10 = 19\,000\, m \text{ W}$$

... [$\because$ Specific heat of oil (S) = 1840 to 2100 J/kg/°C]

Equating this to the amount of artificial cooling required, we have

$$19\,000\, m = 91.4$$

$\therefore$ $m = 91.4 / 19\,000 = 0.0048$ kg / s = 0.288 kg / min **Ans.**

Example question two: The load on the journal bearing is 150 kN due to turbine shaft of 300 mm diameter running at 1800 r.p.m.

Determine the following :

Length of the bearing if the allowable bearing pressure is 1.6 N/mm2, and 2. Amount of heat to be removed by the lubricant per minute if the bearing temperature is 60°C and viscosity of the oil at 60°C is 0.02 kg/m-s and the bearing clearance is 0.25 mm.

Solution. Given : W = 150 kN = 150 × 103 N; d = 300 mm = 0.3 m ; N = 1800 r.p.m. ; p = 1.6 N/mm2; Z = 0.02 kg / m-s ; c = 0.25 mm

Length of the bearing

Let l = Length of the bearing in mm. We know that projected bearing area,

A = l × d = l × 300 = 300 l mm2 and allowable bearing pressure (p),

$$1.6 = \frac{W}{A} = \frac{150 \times 10^3}{300\, l} = \frac{500}{l}$$

$l = 500 / 1.6 = 312.5$ mm **Ans.**

2. Amount of heat to be removed by the lubricant

We know that coefficient of friction for the bearing,

$$\mu = \frac{33}{10^8}\left(\frac{Z.N}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.02 \times 1800}{1.6}\right)\left(\frac{300}{0.25}\right) + 0.002$$

$$= 0.009 + 0.002 = 0.011$$

Rubbing velocity,

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.3 \times 1800}{60} = 28.3 \text{ m/s}$$

$\therefore$ Amount of heat to be removed by the lubricant,

$$Q_g = \mu.W.V = 0.011 \times 150 \times 10^3 \times 28.3 = 46\,695 \text{ J/s or W}$$
$$= 46.695 \text{ kW } \textbf{Ans.}$$

Example question three: A wall bracket supports a plummer block for 80 mm diameter shaft. The length of bearing is 120 mm. The cap of bearing is fastened by means of four bolts, two on each side of the shaft. The cap is to withstand a load of 16.5 kN. The distance between the centre lines of the bolts is50 mm. Determine the thickness of the bearing cap and the diameter of the bolts. Assume safe stresses in tension for the material of the cap, which is cast iron, as 15 MPa and for bolts as 35 MPa. Also check the deflection of the bearing cap taking E = 110 kN / mm2.

Solution : Given : d = 80 mm ; l = 120 mm ; n = 4 ; W = 16.5 kN = 16.5 × 103 N ; a = 150 mm ; σb = 15 MPa = 15 N/mm2; σt = 35 MPa = 35 N/mm2; E = 110 kN/mm2 = 110 × 103 N/mm2 Thickness of the bearing cap We know that thickness of the bearing cap,

$$t = \sqrt{\frac{3\,W.a}{2\sigma_b.l}} = \sqrt{\frac{3 \times 16.5 \times 10^3 \times 150}{2 \times 15 \times 120}} = \sqrt{2062.5}$$

$$= 45.4 \text{ say } 46 \text{ mm } \textbf{Ans.}$$

Diameter of the bolts

Let d_c = Core diameter of the bolts.

We know that

$$\frac{\pi}{4}(d_c)^2\ \sigma_t = \frac{4}{3} \times \frac{W}{n}$$

or

$$\frac{\pi}{4}(d_c)^2\ 35 = \frac{4}{3} \times \frac{16.5 \times 10^3}{4} = 5.5 \times 10^3$$

$$\therefore \qquad (d_c)^2 = \frac{5.5 \times 10^3 \times 4}{\pi \times 35} = 200 \quad \text{or} \quad d_c = 14.2 \text{ mm } \textbf{Ans.}$$

Deflection of the cap

We know that deflection of the cap,

$$\delta = \frac{W.a^3}{4\,E.l.t^3} = \frac{16.5 \times 10^3\ (150)^3}{4 \times 110 \times 10^3 \times 120(46)^3} = 0.0108 \text{ mm } \textbf{Ans.}$$

Since the limited value of the deflection is 0.025 mm, therefore the above value of deflection is within limits.

Example question four: A shaft rotating at constant speed is subjected to variable load. The bearings supporting the shaft are subjected to stationary equivalent radial load of 3 kN for 10 per cent of time, 2 kN for 20 per cent of time, 1 kN for 30 per cent of time and no load for remaining time of cycle. If the total life expected for the bearing is 20 × 106 revolutions at 95 per cent reliability, calculate dynamic load rating of the ball bearing.

Solution. Given : W1 = 3 kN ; n1 = 0.1 n ; W2 = 2 kN ; n2 = 0.2 n ; W3 = 1 kN ; n3 = 0.3 n ; W4 = 0 ; n4 = (1 – 0.1 – 0.2 – 0.3) n = 0.4 n ; L95 = 20 × 106 rev

L90 = Life of the bearing corresponding to reliability of 90 per cent,

L95 = Life of the bearing corresponding to reliability of 95 per cent

= 20 × 106 revolutions ...

(Given) We know that

$$\frac{L_{95}}{L_{90}} = \left[\frac{\log_e (1/R_{95})}{\log_e (1/R_{90})}\right]^{1/b} = \left[\frac{\log_e (1/0.95)}{\log_e (1/0.90)}\right]^{1/1.17} \quad \text{... (}$$

$$= \left(\frac{0.0513}{0.1054}\right)^{0.8547} = 0.54$$

$$\therefore \quad L_{90} = L_{95} / 0.54 = 20 \times 10^6 / 0.54 = 37 \times 10^6 \text{ rev}$$

We know that equivalent radial load,

$$W = \left[\frac{n_1 (W_1)^3 + n_2 (W_2)^3 + n_3 (W_3)^3 + n_4 (W_4)^3}{n_1 + n_2 + n_3 + n_4}\right]^{1/3}$$

$$= \left[\frac{0.1n \times 3^3 + 0.2\, n \times 2^3 + 0.3n \times 1^3 + 0.4\, n \times 0^3}{0.1\, n + 0.2n + 0.3n + 0.4n}\right]^{1/3}$$

$$= (2.7 + 1.6 + 0.3 + 0)^{1/3} = 1.663 \text{ kN}$$

We also know that dynamic load rating,

$$C = W \left(\frac{L_{90}}{10^6}\right)^{1/k} = 1.663 \left(\frac{37 \times 10^6}{10^6}\right)^{1/3} = 5.54 \text{ kN } \textbf{Ans.}$$

7. Spring calculations

A spring is defined as an elastic machine element, which deflects under the action of the load and returns to its original shape when the load is removed. It can take any shape and form depending upon the application.

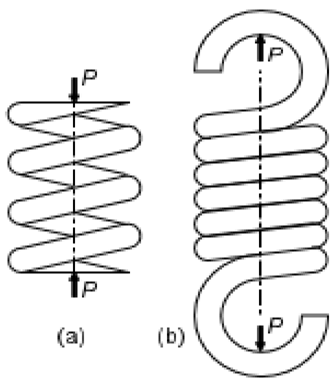
The important functions and applications of springs are as follows:

- Springs are used to absorb shocks and vibrations, e.g., vehicle suspension springs, railway buffer springs, buffer springs in elevators and vibration mounts for machinery.
- Springs are used to measure force, e.g., springs used in weighing balances and scales.
- Springs are used to apply force and control motion

TYPES OF SPRINGS

Springs are classified according to their shape. The shape can be a helical coil of a wire, a piece of stamping or a flat wound-up strip. The most popular type of spring is the helical spring.

The helical spring is made from a wire, usually of circular cross section, which is bent in the form of a helix. There are two basic types of helical springs—***compression spring*** and ***extension spring*** as shown



Helical Springs: (a) Compression Spring (b) Extension Spring

Figure 124.compression spring and extension spring

Helical springs, compression as well as extension, have the following advantages:

- They are easy to manufacture.
- They are cheaper than other types of springs.
- Their reliability is high.
- The deflection of the spring is linearly proportional to the force acting on the spring.

TERMINOLOGY OF HELICAL SPRINGS

The main dimensions of a helical spring subjected to compressive force are shown fig They are as follows:

d = wire diameter of spring (mm)

D_i = inside diameter of spring coil (mm)

D_o = outside diameter of spring coil (mm)

D = mean coil diameter (mm) Therefore

$$D = \frac{D_i + D_o}{2}$$

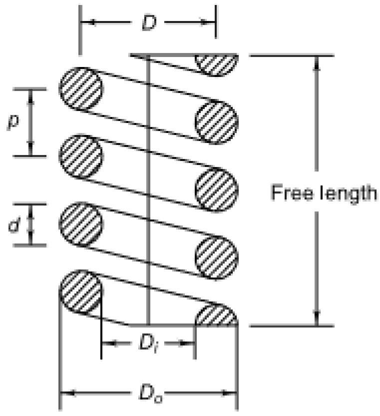


Figure 125

The spring index is defined as the ratio of mean coil diameter to wire diameter. Or,

$$C = \frac{D}{d}$$

The spring index indicates the relative sharpness of the curvature of the coil, There are three terms

Solid Length **Solid** length is defined as the axial length of the spring which is so compressed that the adjacent coils touch each other.

$$\text{Solid length} = N_t d$$

where,

N_t = total number of coils

Compressed Length **Compressed** length is defined as the axial length of the spring, which is subjected to maximum compressive force.

Total gap = $(N_t - 1)$ × Gap between adjacent coils

Free Length Free length is defined as the axial length of an unloaded helical compression spring. In this case, no external force acts on the spring.

free length = compressed length + δ

= solid length + total axial gap + δ

The *pitch of the coil* is defined as the axial distance between adjacent coils in uncompressed state of spring.

$$p = \frac{\text{free length}}{(N_t - 1)}$$

The *stiffness of the spring* (*k*) is defined as the force required to produce unit deflection.

$$k = \frac{P}{\delta}$$

K=stiffness of the spring(n/m)
P=axial spring force(n)

δ = axial deflection of the spring corresponding to the force P (mm)

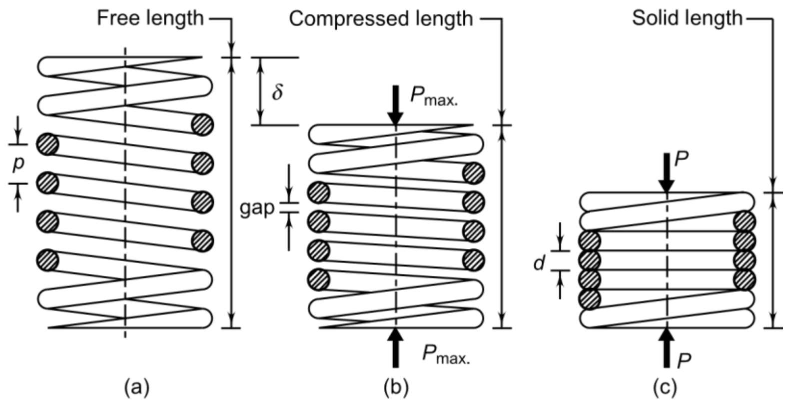


Figure 126

There are two terms related to the spring coils, viz., active coils and inactive coils. *Active coils* are the coils in the spring which contribute to spring action, support the external force and deflect under the action of force. A portion of the end coils, which is in contact with the seat, does not contribute to spring action and are called *inactive coils*.

inactive coils = $N_t - N$

where,

N = number of active coils.

There are four common methods which are used in forming the ends of the helical compression spring as shown in Fig. —plain ends, plain and ground ends, square ends and square and ground ends.

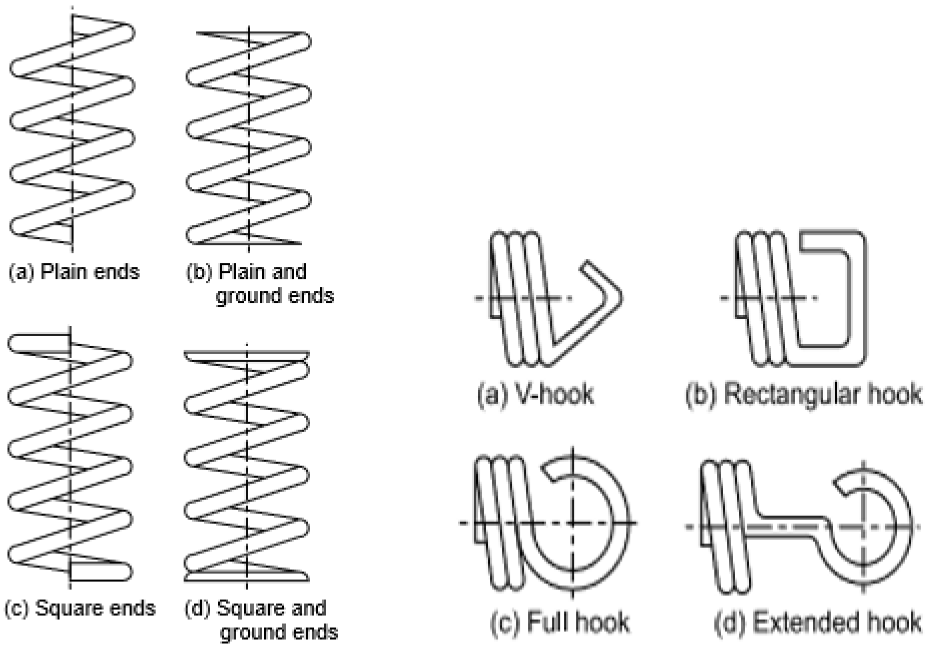


Figure 127.End Styles of Helical Compression Springs AND End Styles of Helical Extension Springs

The dimensions of equivalent bar are as follows:

(i) The diameter of the bar is equal to the wire diameter of the spring (*d*).

(i) The length of one coil in the spring is (*D*). There are *N* such active coils. Therefore, the length of equivalent bar is (*DN*).

(ii) The bar is fi tted with a bracket at each end. The length of this bracket is equal to mean coil radius of the spring (*D*/2).The force *P* acting at the end of the bracket induces torsional shear stress in the bar.

The torsional moment M_t is given by

$$M_t = \frac{PD}{2}$$

The torsional shear stress in the bar is given b

$$\tau_1 = \frac{16M_t}{\pi d^3} = \frac{16(PD/2)}{\pi d^3}$$

or

$$\tau_1 = \frac{8PD}{\pi d^3}$$

The combined effect of these two factors is given by,

$K = K_S\, K_C$

K_S = factor to account for direct shear stress

K_c = factor to account for stress concentration due to curvature effect

the direct shear stress in the bar is given by,

$$\tau_2 = \frac{P}{\left(\frac{\pi}{4}d^2\right)} = \frac{4P}{\pi d^2} = \frac{8PD}{\pi d^3}\left(\frac{0.5d}{D}\right)$$

The shear stress correction factor (K_s) is defined

$$K_s = \left(1 + \frac{0.5d}{D}\right)$$

$$K_s = \left(1 + \frac{0.5}{C}\right)$$

where *K* is called the *stress factor* or *Wahl factor*. The Wahl factor is given by

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

When the spring is subjected to fluctuating stresses, two factors K_s and K_c are separately used. The angle of twist for the equivalent bar

$$\theta = \frac{M_t\, l}{JG} \qquad \text{(f)}$$

where,

θ = angle of twist (radians)
M_t = torsional moment ($PD/2$)
l = length of bar (πDN)
J = polar moment of inertia of bar ($\pi d^4/32$)
G = modulus of rigidity

$$\theta = \frac{(PD/2)(\pi DN)}{(\pi d^4/32)G}$$

$$\theta = \frac{16PD^2N}{Gd^4}$$

(length of bracket)

$$\delta = \frac{8PD^3N}{Gd^4}$$

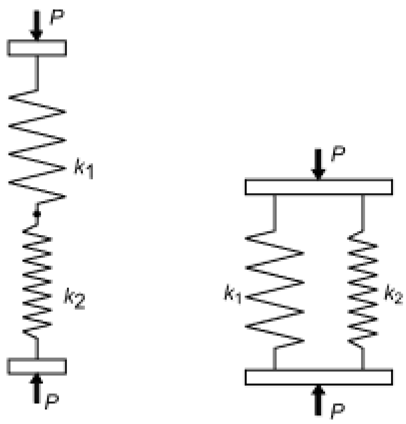
The rate of spring (k) is given by

$$k = \frac{P}{\delta} \qquad k = \frac{Gd^4}{8D^3N}$$

SERIES AND PARALLEL CONNECTIONS

There are two types of spring connections—series and parallel. The objectives of series and parallel combinations are as follows:

(i) to save the space;
(ii) to change the rate of the spring at a certain deflection; and
(iii) to provide a fail-safe system.



Springs in Series Springs in Parallel

(a)

The deflections of the two springs.

$$\delta = \frac{P}{k}$$

$$\delta_1 = \frac{P}{k_1} \text{ and } \delta_2 = \frac{P}{k_2}$$

where k is the combined stiffness of all springs in the connection

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2} \text{ or } k = \frac{k_1 k_2}{k_1 + k_2}$$

Therefore,

$P = P_1 + P_2$

$$P = k\delta$$

$$P_1 = k_1\delta \quad \text{and} \quad P_2 = k_2\delta$$

From Eqs (c) and (d),

$$k\delta = k_1\delta + k_2\delta$$

DESIGN OF HELICAL SPRINGS

There are three objectives for the design of the helical spring. They are as follows:

(i) It should possess sufficient strength to withstand the external load.

(ii) It should have the required load-deflection characteristic.

(iii) It should not buckle under the external load.

$$\tau = K\left(\frac{8PD}{\pi d^3}\right)$$

Substituting $\left(\frac{D}{d} = C\right)$ in the above equation,

$$\tau = K\left(\frac{8PC}{\pi d^2}\right)$$

Therefore, the factor of safety based on torsional yield strength (S_{sy})

$$\tau = \frac{S_{sy}}{1.5}$$

Assuming, $S_{yt} = 0.75 S_{ut}$ and $S_{sy} = 0.577 S_{yt}$

Expression (a) is written as,

$$\tau = \frac{(0.577)(0.75) S_{ut}}{1.5}$$

$$\tau \cong 0.3\, S_{ut}$$

(i) Calculate the Wahl factor by the following equation:

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

Determine wire diameter (*d*)

$$\tau = K\left(\frac{8\,PC}{\pi d^2}\right)$$

Determine mean coil diameter (*D*) by the following relationship:

D =Cd

Determine the number of active coils

$$\delta = \frac{8PD^3N}{Gd^4}$$

Solid length = $N_t d$

Determine the actual deflection of the spring by Eq.

$$\delta = \frac{8PD^3N}{Gd^4}$$

The total axial gap between coils is given by,

total gap = $(N_t - 1) \times$ gap between two adjacent coils

Determine the free length of the spring by the following relationship:

free length = solid length + total gap + ⍰

Determine the pitch of the coil by the following relationship:

$$p = \frac{\text{free length}}{(N_t - 1)}$$

Determine the rate of spring by Eq

$$k = \frac{Gd^4}{8D^3N}$$

calculation

Simple question: It is required to design a helical compression spring subjected to a maximum force of 1250 N. The deflection of the spring corresponding to the maximum force should be approximately 30 mm. The spring index can be taken as 6. The spring is made of patented and cold-drawn steel wire. The ultimate tensile strength and modulus of rigidity of the spring material are 1090 and 81 370 N/mm^2 respectively. The permissible shear stress for the spring wire should be taken as 50% of the ultimate tensile strength. Design the spring and calculate: wire diameter;

(i) mean coil diameter;

(ii) number of active coils;

(iii) total number of coils;

(iv) free length of the spring; and (vi) pitch of the coil.

Draw a neat sketch of the spring showing various dimensions.

Solution

Given $P = 1250$ N $\delta = 30$ mm $C = 6$

$S_{ut} = 1090$ N/mm^2 $G = 81\ 370$ N/mm^2

$\tau = 0.5\ S_{ut}$

Step I *Wire diameter*

The permissible shear stress is given by,

$\tau = 0.5\ S_{ut} = 0.5(1090) = 545$ N/mm^2

From Eq. (10.7),

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4(6)-1}{4(6)-4} + \frac{0.615}{6} = 1.2525$$

From Eq. (10.13),

$$\tau = K\left(\frac{8PC}{\pi d^2}\right) \quad \text{or} \quad 545 = (1.2525)\left\{\frac{8(1250)(6)}{\pi d^2}\right\}$$

$\therefore$ $d = 6.63$ or 7 mm (i)

Step II *Mean coil diameter*

$D = Cd = 6(7) = 42$ mm (ii)

Step III *Number of active coils*

From Eq. (10.8),

$$\delta = \frac{8PD^3N}{Gd^4} \quad \text{or} \quad 30 = \frac{8(1250)(42)^3N}{(81\ 370)(7)^4}$$

$\therefore$ $N = 7.91$ or 8 coils (iii)

Step IV *Total number of coils*

It is assumed that the spring has square and ground ends. The number of inactive coils is 2. Therefore,

$N_t = N + 2 = 8 + 2 = 10$ coils (iv)

Step V *Free length of spring*

The actual deflection of the spring is given by,

$$\delta = \frac{8PD^3N}{Gd^4} = \frac{8(1250)(42)^3(8)}{(81370)(7)^4} = 30.34 \text{ mm}$$

solid length of spring $= N_t\, d = 10(7) = 70$ mm

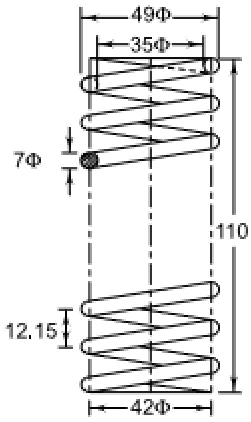
It is assumed that there will be a gap of 1 mm between consecutive coils when the spring is subjected to the maximum force. The total number of coils is 10.

The total axial gap between the coils will be $(10–1) \times 1 = 9$ mm.

$$\begin{aligned}\text{Free length} &= \text{Solid length} + \text{Total axial gap} + \delta \\ &= 70 + 9 + 30.34 \\ &= 109.34 \text{ or } 110 \text{ mm}\end{aligned}$$ (v)

Step VI *Pitch of the coil*

$$\text{Pitch of coil} = \frac{\text{Free length}}{(N_t - 1)} = \frac{109.34}{(10-1)} = 12.15 \text{ mm}$$ (vi)



Question two: A helical compression spring made of circular wire, is subjected to an axial force, which varies from 2.5 kN to 3.5 kN. Over this range of force, the deflection of the spring should be approximately 5 mm. The spring index can be taken as 5. The spring has square and ground ends. The spring is made of patented and cold-drawn steel wire with ultimate tensile strength of 1050 N/mm^2 and modulus of rigidity of 81370 N/mm^2. The permissible shear stress for the spring wire should be taken as 50% of the ultimate tensile strength.

Design the spring and calculate

(i) wire diameter;
(ii) mean coil diameter;
(iii) number of active coils;
(iv) total number of coils;
(v) solid length of the spring;
(vi) free length of the spring;
(vii) required spring rate; and
(viii) actual spring rate

Solution

Given $P = 2.5$ to 3.5 kN $\quad \delta = 5$ mm $\quad C = 5$
$S_{ut} = 1050$ N/mm^2 $\quad G = 81\,370$ N/mm^2 $\quad \tau = 0.5\, S_{ut}$

Step I *Wire diameter*

The permissible shear stress for the spring is given by,

$$\tau = 0.5\, S_{ut} = 0.5\,(1050) = 525 \text{ N/mm}^2$$

From Eq. (10.7),

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4(5)-1}{4(5)-4} + \frac{0.615}{5} = 1.3105$$

From Eq. (10.13),

$$\tau = K\left(\frac{8PC}{\pi d^2}\right) \quad \text{or} \quad 525 = (1.3105)\left\{\frac{8(3500)(5)}{\pi d^2}\right\}$$

$$\therefore \qquad d = 10.55 \quad \text{or} \quad 11 \text{ mm} \qquad \text{(i)}$$

Step II *Mean coil diameter*

$$D = C\,d = 5(11) = 55 \text{ mm} \qquad \text{(ii)}$$

Step III *Number of active coils*

From Eq. (10.8),

$$\delta = \frac{8PD^3N}{Gd^4} \quad \text{or} \quad 5 = \frac{8(3500-2500)(55)^3 N}{(81\,370)(11)^4}$$

$$\therefore \qquad N = 4.48 \quad \text{or} \quad 5 \text{ coils} \qquad \text{(iii)}$$

Step IV *Total number of coils*

For square and ground ends, the number of inactive coils is 2. Therefore,

$$N_t = N + 2 = 5 + 2 = 7 \text{ coils} \qquad \text{(iv)}$$

Step V *Solid length of spring*

solid length of spring $= N_t d = 7(11) = 77$ mm $\qquad$ (v)

Step VI *Free length of spring*

The actual deflection of the spring under the maximum force of 3.5 kN is given by,

$$\delta = \frac{8PD^3N}{Gd^4} = \frac{8(3500)(55)^3(5)}{(81370)(11)^4} = 19.55 \text{ mm}$$

It is assumed that there will be a gap of 0.5 mm between the consecutive coils when the spring is subjected to the maximum force of 3.5 kN. The total number of coils is 7. Therefore, total axial gap will be $(7-1) \times 0.5 = 3$ mm.

Free length = Solid length + Total axial gap + δ

$$= 77 + 3 + 19.55$$
$$= 99.55 \text{ or } 100 \text{ mm}$$

Step VII *Required spring rate*

$$k = \frac{P_1 - P_2}{\delta} = \frac{3500 - 2500}{5} = 200 \text{ N/mm}$$

Step VIII *Actual spring rate*

$$k = \frac{Gd^4}{8D^3N} = \frac{(81370)(11)^4}{8(55)^3(5)} = 179.01 \text{ N/mm}$$

8. Cam calculation

Draw the cam profile for following conditions: Follower type = Knife edged, in-line; lift = 50mm; base circle radius = 50mm; out stroke with SHM, for 600 cam rotation; dwell for 450 cam rotation; return stroke with SHM, for 900 cam rotation; dwell for the remaining period. Determine max. velocity and acceleration during out stroke and return stroke if the cam rotates at 1000 rpm in clockwise direction.

Displacement diagram:

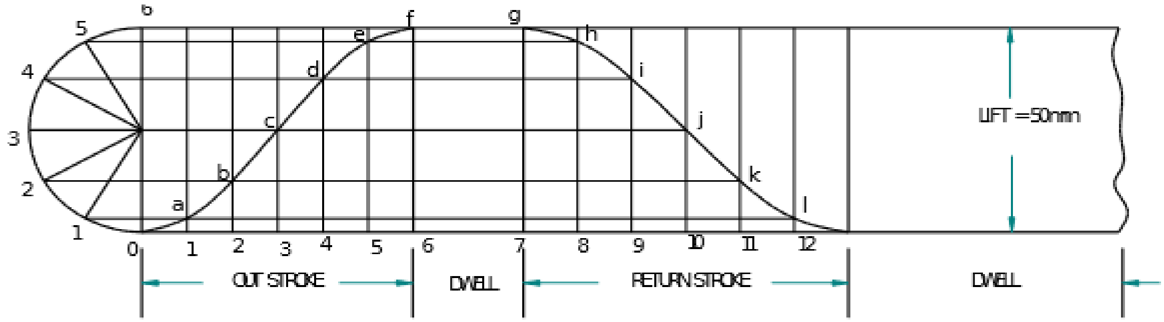


Figure 128.Displacement diagram:

Cam profile: Construct base circle. Mark points 1,2,3…..in direction opposite to the direction of cam rotation. Transfer points a,b,c…..l from displacement diagram to the cam profile and join them by a smooth free hand curve. This forms the required cam profile.

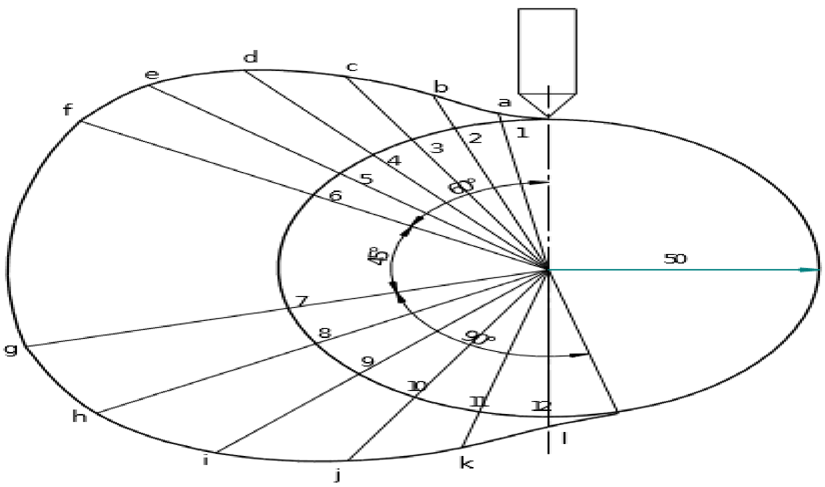


Figure 129.Cam profile

Angular velocity of cam = $\omega = \frac{2\pi N}{60} = \frac{2 \times \pi \times 1000}{60}$ =**104.76 rad/sec**

Max. velocity of follower during outstroke = $vo_{max} = \frac{\pi \omega s}{2\theta_o} =$

$= \frac{\pi \times 104.76 \times 50}{2 \times \pi/3}$ =7857mm/sec =**7.857m/sec**

Similarly Max. velocity of follower during return stroke = , $vr_{max} = \frac{\pi \omega s}{2\theta_r} =$

$= \frac{\pi \times 104.76 \times 50}{2 \times \pi/2}$ = 5238mm/sec = **5.238m/sec**

Max. acceleration during outstroke = $ao_{max} = r\omega^2_p$ (from d3) = $\frac{\pi^2 \omega^2 s}{2\theta_o^2} =$

$= \frac{\pi^2 \times (104.76)^2 \times 50}{2 \times \left(\pi/3\right)^2} =$ 2469297.96mm/sec^2 = **2469.3m/sec^2**

Similarly, Max. acceleration during return stroke = $ar_{max} = \frac{\pi^2 \omega^2 s}{2\theta^2_r} =$

$= \frac{\pi^2 \times (104.76)^2 \times 50}{2 \times \left(\pi/2\right)^2} =$ 1097465.76mm/sec^2 = **1097.5m/sec^2**

Draw the cam profile for the same operating conditions of problem (1), with the follower off set by 10 mm to the left of cam center. Displacement diagram: Same as previous case.

Cam profile: Construction is same as previous case, except that the lines drawn from 1,2,3…. are tangential to the offset circle of 10mm dia. as shown in the fig.

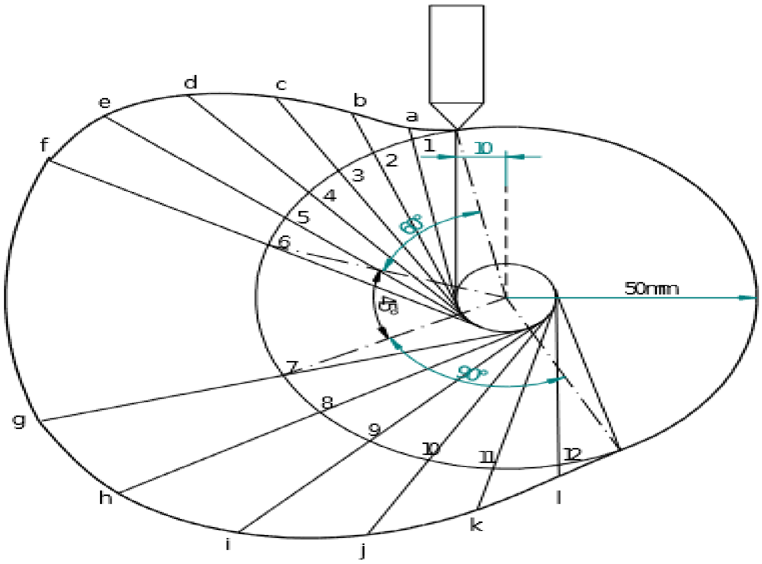


Figure 130

Draw the cam profile for following conditions: Follower type = roller follower, in-line; lift = 25mm; base circle radius = 20mm; roller radius = 5mm; out stroke with UARM, for 1200 cam rotation; dwell for 600 cam rotation; return stroke with UARM, for 900 cam rotation; dwell for the remaining period. Determine max velocity and acceleration during out stroke and return stroke if the cam rotates at 1200 rpm in clockwise direction.

Displacement diagram:

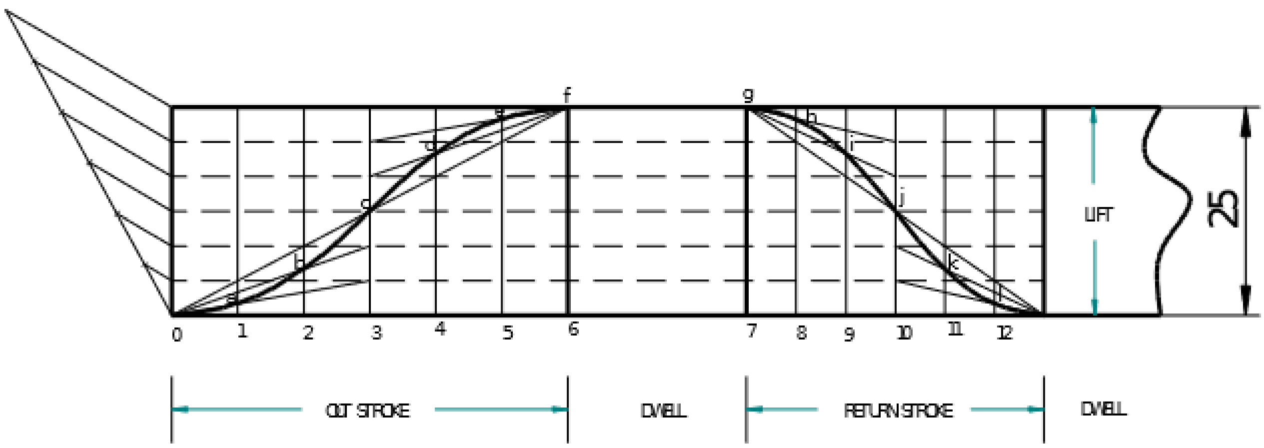


Figure 131.Displacement diagram

Cam profile: Construct base circle and prime circle (25mm radius). Mark points 1,2,3 in direction opposite to the direction of cam rotation, on prime circle. Transfer points a,b,c…..l from displacement diagram. At each of these points a,b,c… draw circles of 5mm radius, representing rollers. Starting from the first point of contact between roller and base circle, draw a smooth free hand curve, tangential to all successive roller positions. This forms the required cam profile.

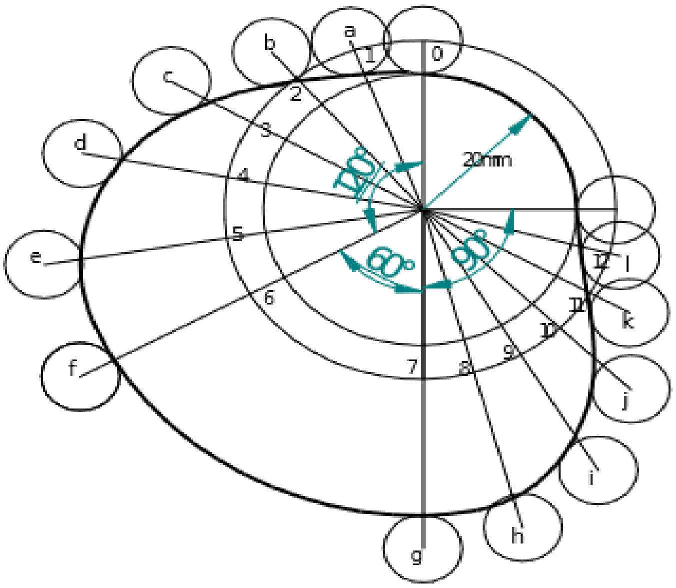


Figure 132.Cam profile

$$= \frac{4 \times (125.71)^2 \times 25}{\left(\pi/2\right)^2} = 639956\text{mm/sec}^2 = \mathbf{639.956 m/sec^2}$$

Angular velocity of the cam $= \omega = \frac{2\pi N}{60} = \frac{2 \times \pi \times 1200}{60} = \mathbf{125.71 rad/sec}$

Max. velocity during outstroke $= vo_{max} = \frac{2s}{t_o} = \frac{2\omega s}{\theta_o} =$

$$= \frac{2 \times 125.71 \times 25}{2 \times \pi/3} = 2999.9\text{mm/sec} = \mathbf{2.999 m/sec}$$

Max. velocity during return stroke $= vr_{max} = \frac{2s}{t_r} = \frac{2\omega s}{\theta_r} = \frac{2 \times 125.71 \times 25}{\pi/2} =$

$= 3999.86\text{mm/sec} = \mathbf{3.999 m/sec}$

Acceleration of the follower during outstroke $= a_o = \frac{vo_{max}}{t_o/2} = \frac{4\omega^2 s}{\theta_o^2} =$

$$= \frac{4 \times (125.71)^2 \times 25}{\left(2 \times \pi/3\right)^2} = 359975\text{mm/sec}^2 = \mathbf{359.975 m/sec^2}$$

Similarly acceleration of the follower during return stroke $= a_r = \frac{4\omega^2 s}{\theta_r^2} =$

Draw the cam profile for following conditions: Follower type = roller follower, off set to the right of cam axis by 18mm; lift = 35mm; base circle radius = 50mm; roller radius = 14mm; out stroke with SHM in 0.05sec; dwell for 0.0125sec; return stroke with UARM, during 0.125sec; dwell for the remaining period. During return stroke, acceleration is 3/5 times retardation. Determine max. velocity and acceleration during out stroke and return stroke if the cam rotates at 240 rpm.

Cam speed = 240rpm. Therefore, time for one rotation = $\dfrac{60}{240} = 0.25\,\text{sec}$

Angle of out stroke = $\theta_o = \dfrac{0.05}{0.25} \times 360 = 72^0$

Angle of first dwell = $\theta_{w1} = \dfrac{0.0125}{0.25} \times 360 = 18^0$

Angle of return stroke = $\theta_r = \dfrac{0.125}{0.25} \times 360 = 180^0$

Angle of second dwell = $\theta_{w2} = 90^0$

Since acceleration is 3/5 times retardation during return stroke,

$a = \dfrac{3}{5} r$ (from acceleration diagram) $\therefore \dfrac{a}{r} = \dfrac{3}{5}$

But $a = \dfrac{v_{max}}{t_a}; r = \dfrac{v_{max}}{t_r} \therefore \dfrac{a}{r} = \dfrac{t_r}{t_a} = \dfrac{3}{5}$

Displacement diagram is constructed by selecting t_a and t_r accordingly.

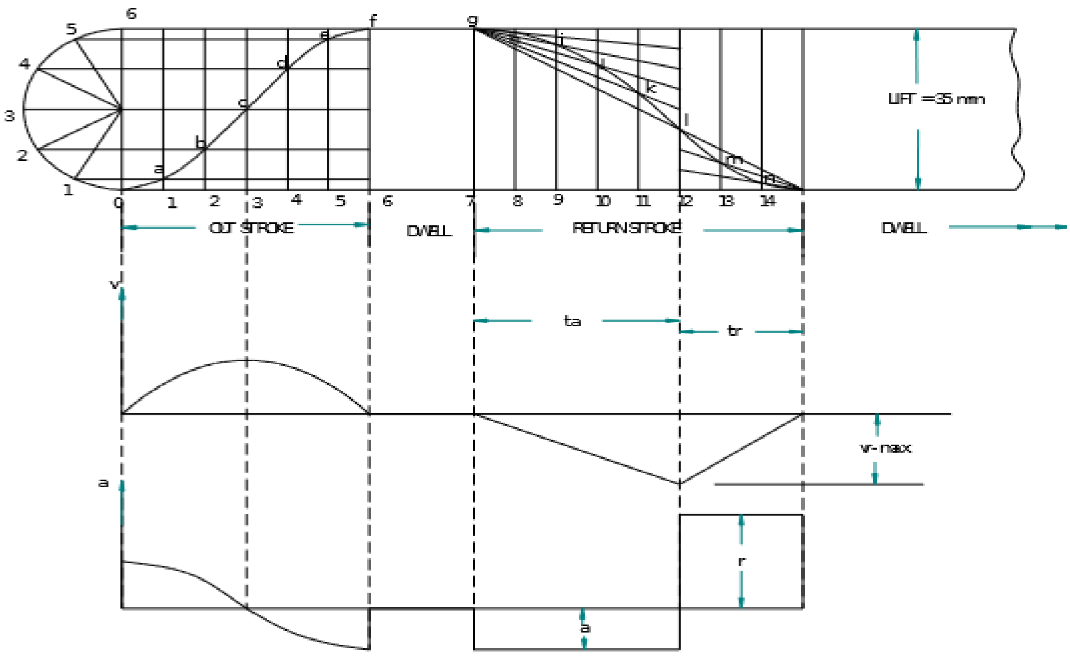


Figure 133

Angular velocity of cam = $\omega = \frac{2\pi N}{60} = \frac{2\times\pi\times 240}{60}$ =**25.14 rad/sec**

Max. velocity of follower during outstroke = $vo_{max} = \frac{\pi\omega s}{2\theta_o} =$

$= \frac{\pi\times 25.14\times 35}{2\times\left(2\times\pi/5\right)}$ = 1099.87mm/sec =**1.1m/sec**

Similarly Max. velocity during return stroke = $vr_{max} = \frac{2\omega s}{\theta_r} = \frac{2\times 25.14\times 35}{\pi} =$

= 559.9 mm/sec = **0.56m/sec**

Max. acceleration during outstroke = $ao_{max} = r\omega^2_p$ (from d3) = $\frac{\pi^2\omega^2 s}{2\theta_o^2}$ =

$= \frac{\pi^2\times(25.14)^2\times 35}{2\times\left(2\times\pi/5\right)^2}$ = 69127.14mm/sec^2 = **69.13m/sec^2**

acceleration of the follower during return stroke =

$a_r = \frac{vr_{max}}{t_a} = \frac{2\omega s/\theta_r}{5\times\pi/8\times\omega} = \frac{16\times\omega^2\times s}{5\times\pi\times\theta_r} = \frac{16\times(25.14)^2\times 35}{5\times\pi\times\pi}$ = 7166.37 mm/sec^2 = **7.17m/sec^2**

similarly retardation of the follower during return stroke =

$r_r = \frac{vr_{max}}{t_r} = \frac{2\omega s/\theta_r}{3\times\pi/8\times\omega} = \frac{16\times\omega^2\times s}{3\times\pi\times\theta_r} = \frac{16\times(25.14)^2\times 35}{3\times\pi\times\pi}$ = 11943.9 mm/sec^2 = **11.94m/sec^2**

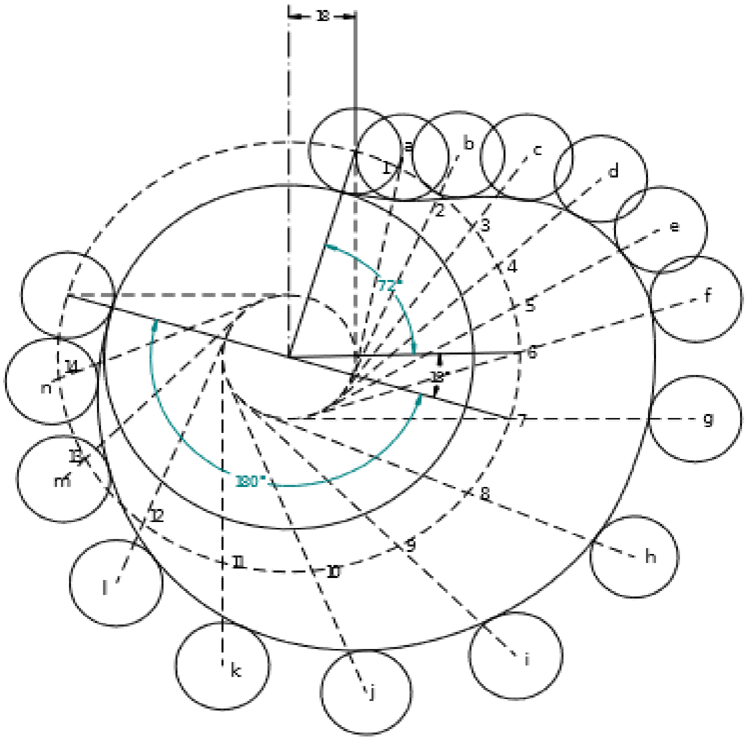
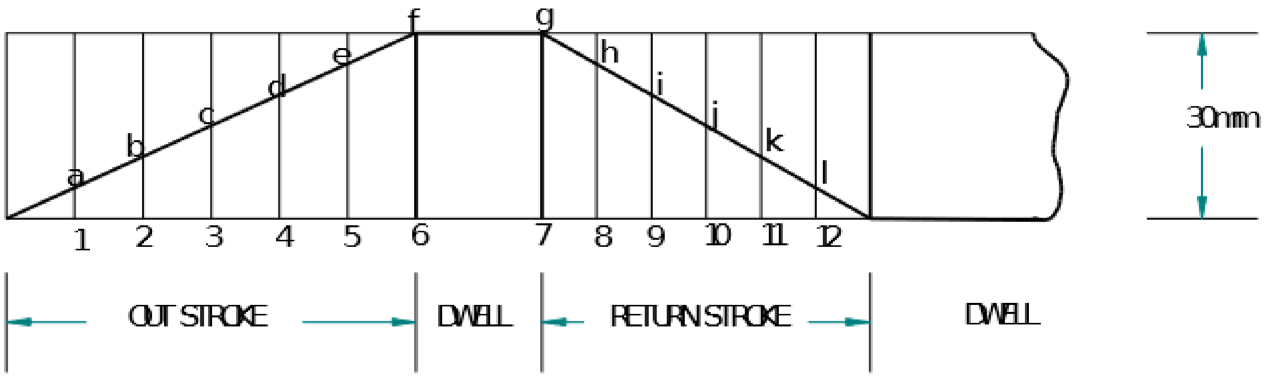


Figure 134

Draw the cam profile for following conditions: Follower type = knife edged follower, in line; lift = 30mm; base circle radius = 20mm; out stroke with uniform velocity in 1200 of cam rotation; dwell for 600; return stroke with uniform velocity, during 900 of cam rotation; dwell for the remaining period.

Displacement diagram:



Cam profile:

Figure 135

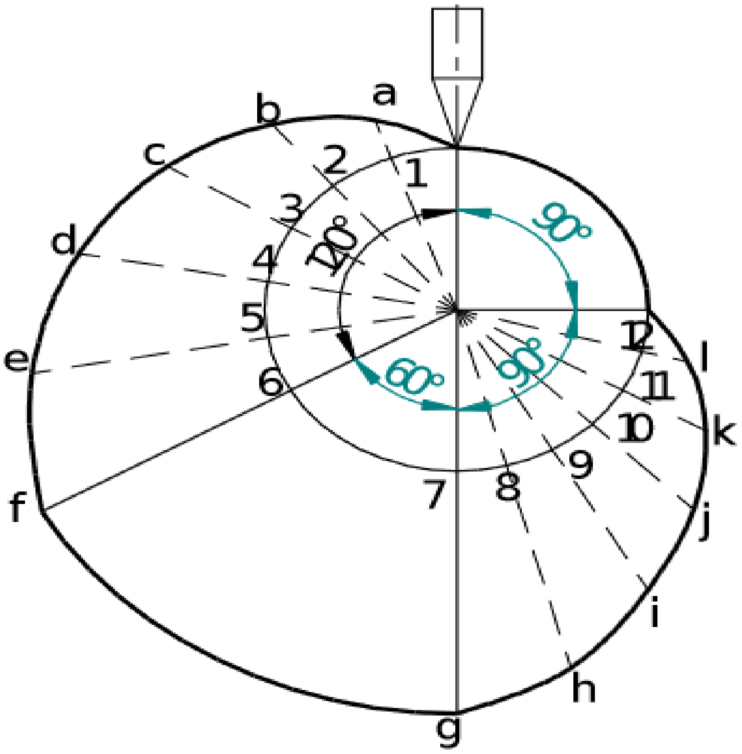


Figure 136

L O 2.3 -Connect working mechanisms

- Content/Topic1: Types of motion in mechanical systems

1.1. Rotary motion

Rotary motion is anything that moves in a circle. This type of motion was among the first discovered in ancient times. Think of a spinning wheel on which people spun wool. A car's engine works the same way. Like linear cylinders, rotary actuators are used across a wide range of industries and come in electric, pneumatic and hydraulic options.

1.2. Oscillating motion

Something that oscillates moves back and forth. Anything that repeats the motion cycle after a certain period is considered to be oscillating. This type of motion is found everywhere in our world: a sprinkler system, the pendulum of a clock or even sound waves.
You may be thinking that a rotary actuator functions as an oscillating device, and for that matter, so does a linear one when it repeats a continuous movement. When it comes to actuators, linear and rotary can be viewed as oscillating.

1.3. Linear motion

Simple enough, linear motion is anything that moves in a straight line, like our linear actuators. Time, as far as we know, moves in a linear fashion. Just like rotary devices, you can find linear cylinders in electric, pneumatic or hydraulic options. They have driven the field of automation, manufacturing, robotics, and others into a new age because, in the past, rotary motion was the only means to create motion.

1.4. Reciprocating motion

- Related to other forms of motion, in particular oscillating motion. In this form of motion, an object is translated, or moved linearly, in one direction and then back along the same path in the opposite direction until it returns to its starting point; the cycle is then repeated. One example is a power saw.
- Content/Topic2: Degree of freedom of movement calculation

Degree of freedom of movement Correct number (**DOF**) of a mechanical system is the number of independent parameters that define its configuration or state. It is important in the analysis of systems of bodies in mechanical engineering, structural engineering, aerospace engineering, robotics, and other fields.

The position of a single railcar (engine) moving along a track has one degree of freedom because the position of the car is defined by the distance along the track. A train of rigid cars connected by hinges to an engine still has only one degree of freedom because the positions of the cars behind the engine are constrained by the shape of the track.

consider a system of n rigid bodies moving in space has $6n$ degrees of freedom measured relative to a fixed frame. In order to count the degrees of freedom of this system, include the fixed body in the count of bodies, so that mobility is independent of the choice of the body that forms the fixed frame. Then the degree-of-freedom of the unconstrained system of $N = n + 1$ is

$$M = 6n = 6(N-1),$$

because the fixed body has zero degrees of freedom relative to itself. Joints that connect bodies in this system remove degrees of freedom and reduce mobility. Specifically, hinges and sliders each impose five constraints and therefore remove five degrees of freedom. It is convenient to define the number of constraints c that a joint imposes in terms of the joint's freedom f, where $c = 6 - f$. In the case of a hinge or slider, which are one degree of freedom joints, have $f = 1$ and therefore $c = 6 - 1 = 5$. The result is that the mobility of a system formed from n moving links and j joints each with

freedom f_i, $i = 1, ..., j$, is given by

$$M = 6n - \sum_{i=1}^{j} (6 - f_i) = 6(N - 1 - j) + \sum_{i=1}^{j} f_i$$

Recall that N includes the fixed link.

There are two important special cases: (i) a simple open chain, and (ii) a simple closed chain. A single open chain consists of n moving links connected end to end by n joints, with one end connected to a ground link. Thus, in this case $N = j + 1$ and the mobility of the chain is

$$M = \sum_{i=1}^{j} f_i$$

For a simple closed chain, n moving links are connected end-to-end by $n + 1$ joints such that the two ends are connected to the ground link forming a loop. In this case, we have $N = j$ and the mobility of the chain is

$$M = \sum_{i=1}^{j} f_i - 6$$

The number of degrees of freedom of a mechanism is also called the mobility, and it is given the symbol . When M .the configuration of a mechanism is completely defined by positioning one link,that system has one degree of freedom. Most commercially produced mechanisms have one degree of freedom. In constrast, robotic arms can have three, or more,degrees of freedom.

Degrees of freedom for planar linkages joined with common joints can be calculated through Gruebler's equation:

$$M = \text{degrees of freedom} = 3(n - 1) - 2j_p - j_h$$

Where:

N=total number of links in the mechanism

Jp=total number joint (pins or sliding joint)

Jn=total number of higher order joint (cam or join)

As mentioned,most linkages used in machines have one degree of freedom. A single degree-of-freedom linkage is shown

Linkages with zero, or negative, degrees of freedom are termed locked mechanisms. These mechanisms are unable to move and form a structure. A truss is a structure composed of simple links and connected with pin joints and zero degrees of freedom. A locked mechanism is shown in Figure

Linkages with multiple degrees of freedom need more than one driver to precisely operate them. Common multi-degree-of-freedom mechanisms are open-loop kinematic chains used for reaching and positioning, such as robotic arms and backhoes.Ingeneral,multi-degree-offreedom linkages offer greater ability to precisely position a link.A multi-degree-of-freedom mechanism is shown in Figure

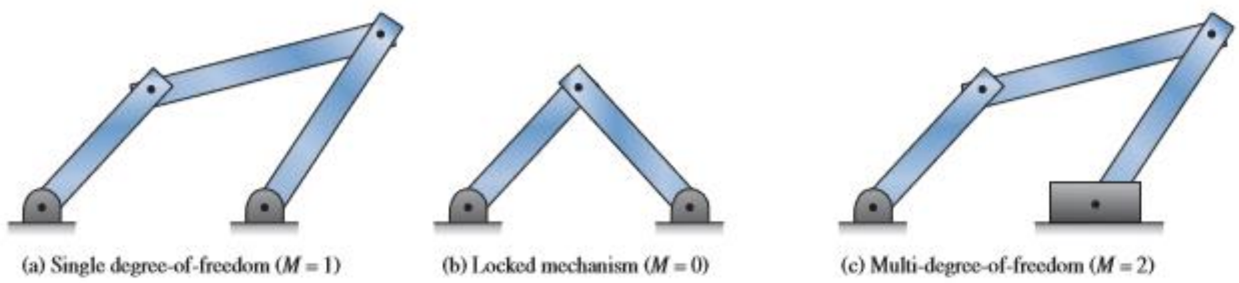


Figure 137.degree-of-freedom mechanism

EXAMPLE PROBLEM

shows a toggle clamp. Draw a kinematic diagram, using the clamping jaw and the handle as points of interest.Also compute the degrees of freedom for the clamp.

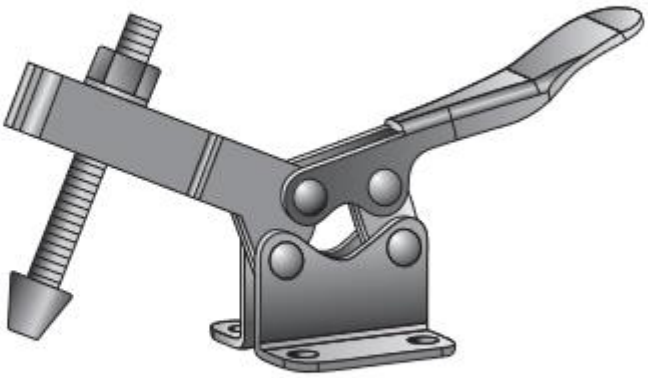


Figure 138.toggle clamp

SOLUTION: 1. Identify the Frame The component that is bolted to the table is designated as the frame. The motion of all other links is determined relative to this frame. The frame is numbered as link 1.

Identify All Other Links

Careful observation reveals three other moving parts:

Link 2: Handle

Link 3: Arm that serves as the clamping jaw

Link 4: Bar that connects the clamping arm and handle

Identify the Joints

Four pin joints are used to connect these different links (link 1 to 2, 2 to 3, 3 to 4, and 4 to 1). These joints are lettered A through D.

Identify Any Points of Interest

The motion of the clamping jaw is desired. This is designated as point of interest X. Finally, the motion of the end of the handle is also desired.This is designated as point of interest Y.

Draw the Kinematic Diagram

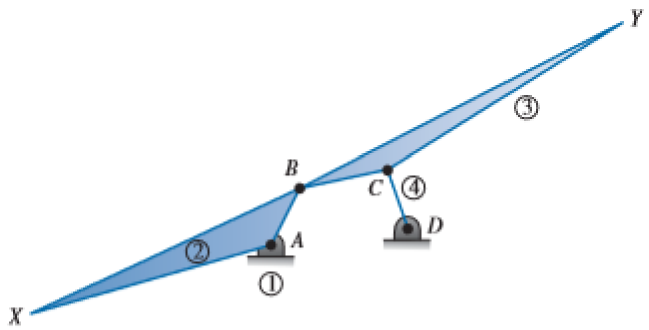


Figure 139.Kinematic diagram for Example Problem

Calculate Mobility Having four links and four pin joints,

$$n = 4, j_p = 4 \text{ pins}, j_h = 0$$

$$M = 3(n - 1) - 2j_p - j_h = 3(4 - 1) - 2(4) - 0 = 1$$

With one degree of freedom, the clamp mechanism is constrained. Moving only one link, the handle,precisely positions all other links in the clamp (R.S. KHURMI J.K. GUPTA, 2005)(bhandaari). (Henderson, 1996-2001)

Learning Unit 3 -analyses the stress-strain state of each element under loading

LO 3.1 - Identify forces acting on elements of machine

- Content/Topic1: **Types of forces**

1.1. Applied force

An applied force is a force which is applied to an object by a person or another object. If a person is pushing a desk across the room, then there is an applied force acting upon the object. The applied force is the force exerted on the desk by the person. The following four types of the load are important from the subject point of view:

- ✓ **Dead or steady load:** A load is said to be a dead or steady load, when it does not change in magnitude or direction.
- ✓ **Live or variable load :** A load is said to be a suddenly applied or shock load, when it is suddenly applied or removed.
- ✓ **Impact load**: A load is said to be an impact load, when it is applied with some initial velocity. Note: A machine part resists a dead load more easily than a live load and a live load more easily than a shock load.

1.2. Normal force

the normal force is the support force exerted upon an object which is in contact with another stable object. For example, if a book is resting upon a surface, then the surface is exerting an upward force upon the book in order to support the weight of the book. On occasions, a normal force is exerted horizontally between two objects which are in contact with each other.

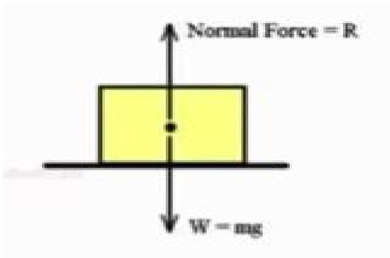


Figure 140.Normal force

1.3. Tension force

The tension is the force which is transmitted through a string, rope, or wire when it is pulled tight by forces acting from each end. The tensional force is directed along the wire and pulls equally on the objects on either end of the wire.

1.5. friction force

the friction force is the force exerted by a surface as an object moves across it or makes an effort to move across it. The friction force opposes the motion of the object. For example, if a book moves across the surface of a desk, then the desk exerts a friction force in the opposite direction of its motion. Friction results from the two surfaces being pressed together closely, causing intermolecular attractive forces between molecules of different surfaces. The friction force can be calculated using the equation:

$$F_{frict} = \mu \times F_{norm}$$

where μ = coefficient of friction

1.5. Compression force

Compression Force is the application of power, pressure, or exertion against an object that causes it to become squeezed, squashed, or compacted. Objects routinely subjected to compression forces include columns, gaskets, disc brakes, and the components of fuel cells

Compression force is usually captured in New tons (N); defined as a unit of force that give to a mass of one kilogram an acceleration of 1 meter per second squared (m/s^2, commonly represented as "a").

$$N = mx\ a$$

1.6. Torsion force

a torsion force is a load that is applied to a material through torque. The torque that is applied creates a shear stress. If a torsion force is large enough, it can cause a material to undergo a twisting motion during elastic and plastic deformation. Torsion force is commonly measured in Newton-meters.

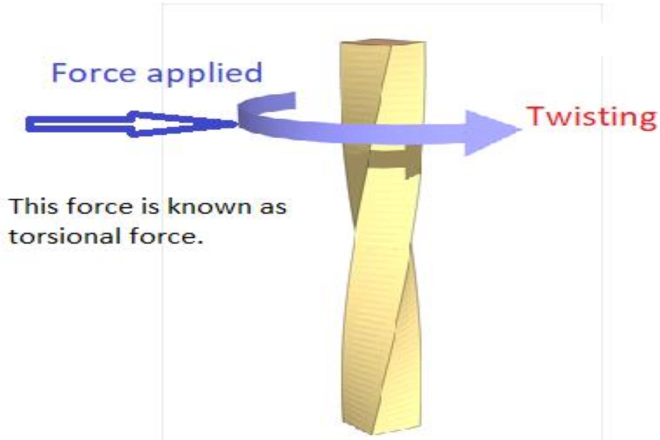


Figure 141.Torsion force

The torsion is the twisting of rod or beam as aresult of an applied torque.
The amount of tuisting (θ) depend on:

- applied torque T=FR
- length torque of rod or beam
- torsional constant J depend on shape

$$\boxed{\theta = \frac{\tau L}{GJ}}$$

- Modulus of rigidity (shear modulus)

$$G = \frac{F/A}{\Delta x/L}$$

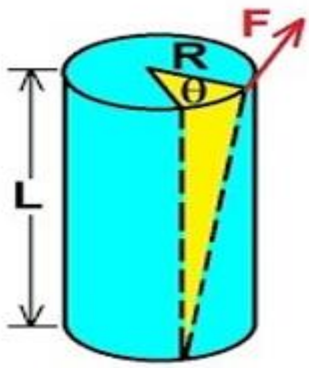
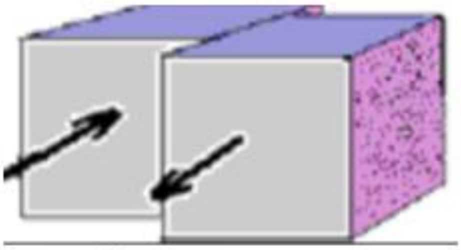
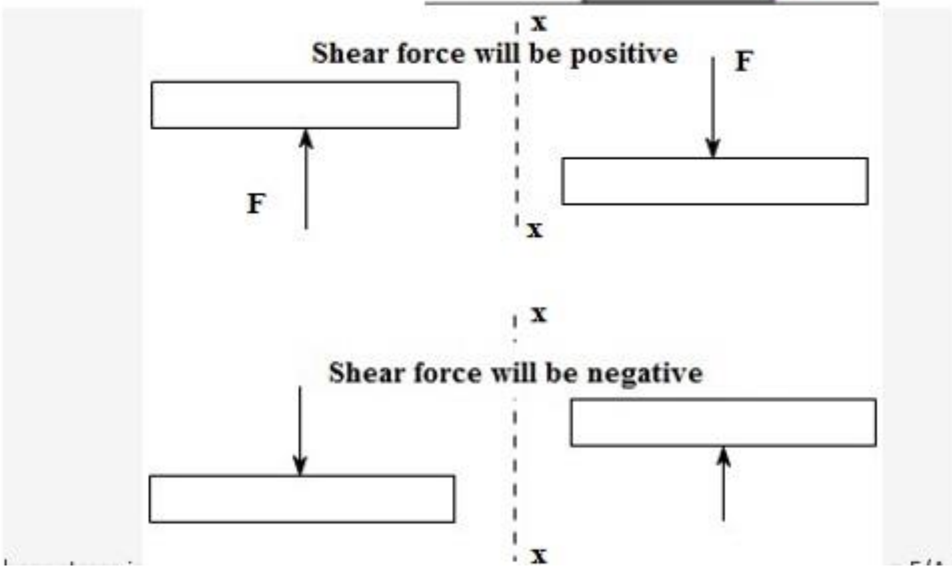


Figure 142.

1.8. Shear force:

Shearing forces are unaligned forces pushing one part of a body in one specific direction, and another part of the body in the opposite direction.





Shear stress is = F/A.

LO3.2 -Interpret stress-strain diagram

- **Content/Topic1: Stress-strain diagram**

it is necessary to know how the material will function in service. For this, certain characteristics or properties of the material should be known. The mechanical properties mostly used in mechanical engineering practice are commonly determined from a standard tensile test. This test consists of gradually loading a standard specimen of a material and noting the corresponding values of load and elongation until the specimen fractures. The stress is determined by dividing the load values by the original cross-sectional area of the specimen.

The elongation is measured by determining the amounts that two reference points on the specimen are moved apart by the action of the machine.

The original distance between the two reference points is known as gauge length.

The strain is determined by dividing the elongation values by the gauge length.

The values of the stress and corresponding strain are used to draw the stress-strain diagram of the material tested.

A stress-strain diagram for a mild steel under tensile test is shown in Fig.

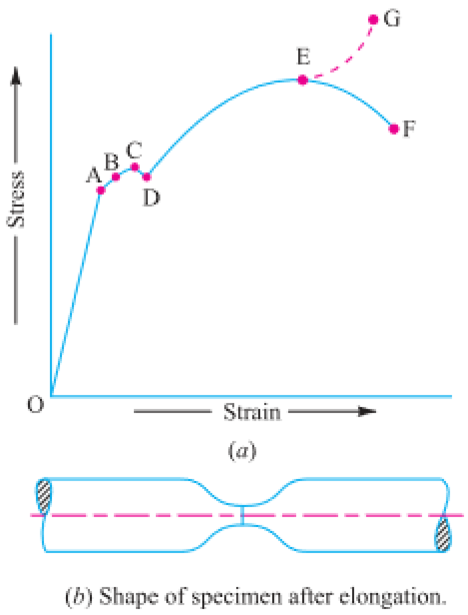


Figure 143.Stress-strain diagram for a mild steel

The various properties of the material are discussed below:

- **Proportional limit**

We see from the diagram that from point O to A is a straight line, which represents that the stress is proportional to strain. Beyond point A, the curve slightly deviates from the straight line. It is thus obvious, that Hooke's law holds good up to point A and it is known as proportional limit. It is defined as that stress at which the stress-strain curve begins to deviate from the straight line.

- **Elastic limit**

It may be noted that even if the load is increased beyond point A up to the point B, the material will regain its shape and size when the load is removed. This means that the material has elastic properties up to the point B. This point is known as elastic limit. It is defined as the stress developed in the material without any permanent set.

- **Yield point**

If the material is stressed beyond point B, the plastic stage will reach i.e. on the removal of the load, the material will not be able to recover its original size and shape. A little consideration will show that beyond point B, the strain increases at a faster rate with any increase in the stress until the point C is reached. At this point, the material yields before the load and there is an appreciable strain without any increase in stress. In case of mild steel, it will be seen that a small load drops to D, immediately after yielding commences. Hence there are two yield points C and D. The points C and D are called the upper and lower yield points respectively. The stress corresponding to yield point is known as yield point stress.

- **Ultimate stress**

At D, the specimen regains some strength and higher values of stresses are required for higher strains, than those between A and D. The stress (or load) goes on increasing till thepoint E is reached. The gradual increase in the strain (or length) of the specimen is followed with the uniform reduction of its cross-sectional area. The work done, during stretching the specimen, is transformed largely into heat and the specimen becomes hot. At E, the stress, which attains its maximum value is known as ultimate stress. It is defined as the largest stress obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece.

- **Breaking stress**

After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area of the specimen, A little consideration will show that the stress (or load) necessary to break away the specimen, is less than the maximum stress. The stress is, therefore, reduced until the specimen breaks away at point F. The stress corresponding to point F is known as breaking stress.

- **Percentage reduction in area**

It is the difference between the original cross-sectional area and cross-sectional area at the neck (i.e. where the fracture takes place).

This difference is expressed as percentage of the original cross-sectional area.

Let A = Original cross-sectional area,

and a = Cross-sectional area at the neck. Then reduction in area = A – a

and percentage reduction in area.

$$= \frac{A - a}{A} \times 100$$

- **Percentage elongation**

It is the percentage increase in the standard gauge length (i.e. original length) obtained by measuring the fractured specimen after bringing the broken parts together.

Let l = Gauge length or original length, and

L = Length of specimen after fracture or final length. ∴ Elongation = L – l

percentage elongation:

$$= \frac{L - l}{l} \times 100$$

Example calculation:

A mild steel rod of 12 mm diameter was tested for tensile strength with the gauge length of 60 mm. Following observations were recorded: Final length = 80 mm; Final diameter = 7 mm; Yield load = 3.4 kN and Ultimate load = 6.1 kN.Calculate : 1. yield stress, 2. ultimate tensile stress, 3. percentage reduction in area, and 4. percentage elongation.

Solution. Given : D = 12 mm ; l = 60 mm ; L = 80 mm ; d = 7 mm ; W_y = 3.4 kN = 3400 N; W_u = 6.1 kN = 6100 N

We know that original area of the rod,

$$A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(12)^2 = 113 \text{ mm}^2$$

and final area of the rod,

$$a = \frac{\pi}{4} \times d^2 = \frac{\pi}{4}(7)^2 = 38.5 \text{ mm}^2$$

1. Yield stress

We know that yield stress

$$= \frac{W_y}{A} = \frac{3400}{113} = 30.1 \text{ N/mm}^2 = 30.1 \text{ MPa} \quad \textbf{Ans.}$$

2. Ultimate tensile stress

We know the ultimate tensile stress

$$= \frac{W_u}{A} = \frac{6100}{113} = 54 \text{ N/mm}^2 = 54 \text{ MPa} \quad \textbf{Ans.}$$

3. Percentage reduction in area

We know that percentage reduction in area

$$= \frac{A - a}{A} = \frac{113 - 38.5}{113} = 0.66 \text{ or } 66\% \quad \textbf{Ans.}$$

4. Percentage elongation

We know that percentage elongation

$$= \frac{L - l}{L} = \frac{80 - 60}{80} = 0.25 \text{ or } 25\% \quad \textbf{Ans.}$$

LO3.3 - Calculate working stress according to factor of safety

- **Content/Topic1: Working stress calculation**

Working Stress When designing machine parts, it is desirable to keep the stress lower than the maximum or ultimate stress at which failure of the material takes place. This stress is known as the working stress or design stress. It is also known as safe or allowable stress.

Note: By failure it is not meant actual breaking of the material. Some machine parts are said to fail when they have plastic deformation set in them, and they no more perform their function satisfactory

- Working stress formula

- Content/Topic2:Factor of safety calculation

It is defined, in general, as the ratio of the maximum stress to the working stress. Mathematically,

$$\text{Factor of safety} = \frac{\text{Maximum stress}}{\text{Working or design stress}}$$

In case of ductile materials e.g. mild steel, where the yield point is clearly defined, the factor of safety is based upon the yield point stress. In such cases,

$$\text{Factor of safety} = \frac{\text{Yield point stress}}{\text{Working or design stress}}$$

In case of brittle materials e.g. cast iron, the yield point is not well defined as for ductile materials. Therefore, the factor of safety for brittle materials is based on ultimate stress.

$$\text{Factor of safety} = \frac{\text{Ultimate stress}}{\text{Working or design stress}}$$

This relation may also be used for ductile materials. Note: The above relations for factor of safety are for static loading.

- Content/Topic3: Factors for selection of factor of safety

The selection of a proper factor of safety to be used in designing any machine component depends upon a number of considerations, such as the material, mode of manufacture, type of stress, general service conditions and shape of the parts.

Selecting a proper factor of safety, a design engineer should consider the following points:

1.The reliability of the properties of the material and change of these properties during service
2. The reliability of test results and accuracy of application of these results to actual machine parts:
3. The reliability of applied load
4. The certainty as to exact mode of failure
5. The extent of simplifying assumptions
6. The extent of localized stresses
7. The extent of initial stresses set up during manufacture
8. The extent of loss of life if failure occurs
9. The extent of loss of property if failure occurs.

Each of the above factors must be carefully considered and evaluated. The high factor of safety results in unnecessary risk of failure. The values of factor of safety based on ultimate strength for different materials and type of load are given in the following table

Material	*Steady load*	*Live load*	*Shock load*
Cast iron	5 to 6	8 to 12	16 to 20
Wrought iron	4	7	10 to 15
Steel	4	8	12 to 16
Soft materials and alloys	6	9	15
Leather	9	12	15
Timber	7	10 to 15	20

Table: Values of factor of safety

Learning Unit4 - Assess machine failure elements

L O 4.1-Inspect machine elements

- Content/Topic1:Machine elements inspection sequences

3.1. Inspection of spares, storage and standby Equipment

Keeping spares and standby machinery in a prime, healthy state can be a daunting task, to say the least. Often machines and critical spare parts must be stored for years in a "ready for operation" state.

Several articles have been published in *Machinery Lubrication* magazine on maintaining idle and standby equipment. This is serious business to protect your investment and more importantly to sustain a state of readiness. Without a doubt, the central enemy for this type of equipment is water that condenses, settles, puddles and corrodes. Therefore, from the standpoint of inspection, looking for water entry points and the presence of invaded water is high on the list.

Many types of machines are internally flooded with oil during storage to minimize air movement between the headspace and the atmosphere. This also keeps internal surfaces oil wet, which would otherwise be exposed to condensation and other atmospheric contaminants.

In gearboxes, these surfaces would include all bearings, internal shafts and gears. Because of hydrostatic forces, these flooded machines are prone to leakage over time, often through gaskets and lip seals. All evidence of leakage, from dampness to oil puddles, should be noted and corrected.

Also, check to ensure:

- ✓ All shafts and couplings have protective coatings still in place.
- ✓ Lube lines and components are tightly sealed (caps, plugs, etc.), and hatches and covers are battened down.
- ✓ Reservoirs and sumps are clean and free of water and sludge.
- ✓ Shafts are being rotated frequently.
- ✓ Dirt and other debris have not accumulated on exterior surfaces.
- ✓ Parts and small assemblies are sealed (e.g., plastic sheets/bags) and oriented correctly (i.e., vertical versus horizontal), including hydraulic cylinders, bearings, gearboxes, pumps, etc.
- ✓ Storage areas do not expose spare parts, assemblies and stored machines to vibration.

4. Alignment of shafts and couplings

The process of shaft alignment is relatively straightforward nowadays, particularly with the arrival of easy-to-use and highly accurate laser alignment tools. But technology can't replacecommon sense and there is one crucial factor that should always be included in the shaft alignment process and that is *thorough coupling inspection*.

Shaft couplings are simple and relatively inexpensive devices which provide an extremely efficient method of transmitting power/torque from the driving machine to the driven machine, and if they are properly sized, assembled, aligned and lubricated, they should perform well over a long period of time. However, when there are problems, the fall-out can extend beyond the coupling itself to the bearings, seals, gears and other components and lead to costly stoppages and productivity losses.

5. Tightness and seal of lube lines and components (caps, plugs, etc.)

The seal oil system prevents gas from leaking out of the compressor case. Figure below is a schematic of a typical seal oil system working in combination with a buffer gas system. These systems require auxiliary equipment such as pumps, regulators, filters, and so on for proper operation. The oil that flows across the outboard ring mixes with the bearing lube oil and returns to a lube oil tank. On the inboard side of the rings, the seal oil mixes with the buffer gas. Since the pressure at each end of the compressor case is at or near suction pressure and maintaining the buffer gas at a pressure above the compressor suction pressure and seal oil pressure above buffer gas pressure, leakage of gas from the case and leakage of seal oil into the case are prevented.

6. **Cleanness and free of water and sludge of reservoirs and sumps**

A proven method of removing water contamination from lubricating fluids is using silica gel technology. Silica gel extracts water vapor from the air as it is drawn through a silica gel bed, as well as covering a wide range of temperatures. Most units using silica gel technology to remove moisture provide a fail-proof method of determining when replacement is necessary.

Obtaining a representative oil sample from a closed loop lubrication system is essential to any good oil analysis program. As oil analysis programs become more standardized for routine monitoring, especially within equipment warranty support programs, equipment makers provide more sampling points on new equipment. These access sampling points make it easier for operators to quickly and easily take representative oil samples. The particles in an oil sample tend to settle out. If they are large and dense, they settle out rapidly. The particles of wear, corrosion, oil degradation, and contamination provide valuable diagnostic information about the condition of the oil as well as the condition of the wearing surfaces of the machine. Since particles exist as a separate phase in the oil, they are not evenly distributed in the system. The very smallest particles tend to remain suspended and pass through all but the finest filters so they are generally distributed evenly throughout the oil piping system. Consequently, in order to capture a representative sample, the sampling location must be carefully considered

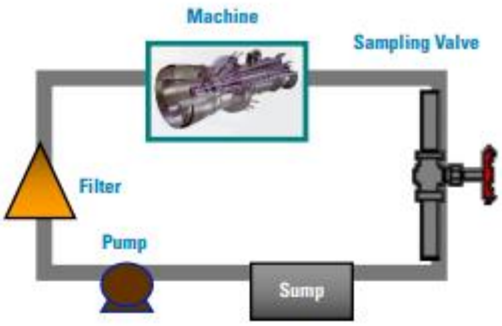


Figure 144

Free Water and Total Water Measurement

Water calibration using the Fluid Scan analyzer provides accurate determination of total water contamination in an oil sample of 1000 and above for all turbine oils and from 1 ppm and above for a growing number of turbine oils. Turbine oils typically are formulated to have high thermal stability, oxidation resistance, and excellent water separation. Lubricants available specifically for gas turbines or steam turbines are designed with specific additive formulations, but there are also many oils that can work with all different types of turbines. Gas turbines have the tendency to build up sludge and varnish whereas steam turbines may experience oxidation, foaming, and sludge. However, a concern of all turbine systems is water contamination. Water contamination – Oil and free water don't mix, not chemically anyway. But under certain circumstances, they will combine to form an emulsion which looks like coffee with cream, and this will actually increase the kinematic viscosity of oil.

7. **Frequent rotation of shafts**

The importance of the driven shafts (tooling spindles) is often neglected during the normal maintenance routine of a roll forming mill. One or more bent or excessively worn shafts can create numerous production and quality problems that are difficult to trace and solve. To follow are recommended guidelines to use for inspection, and rework justification for your driven shafts. By instituting this inspection process into your mill maintenance program, you can determine the value to your organization of reworking the driven shafts vs. replacing them. For clarity, the following examples show the spindles removed from the mill housings/towers. Use the same inspection process as described while the shafts are installed on the machine.

Check the outside diameter (O.D.) of the shaft.

Step 1: Begin by measuring the O.D. of the shaft closest to the tooling shoulder to establish a baseline diameter. Generally, the spindles are not worn in this area. (Caution: Sometimes driven shafts are not made to the nominal diameter when originally manufactured and if too far off, (undersize) can cut into your undersize tolerance right from the beginning.)

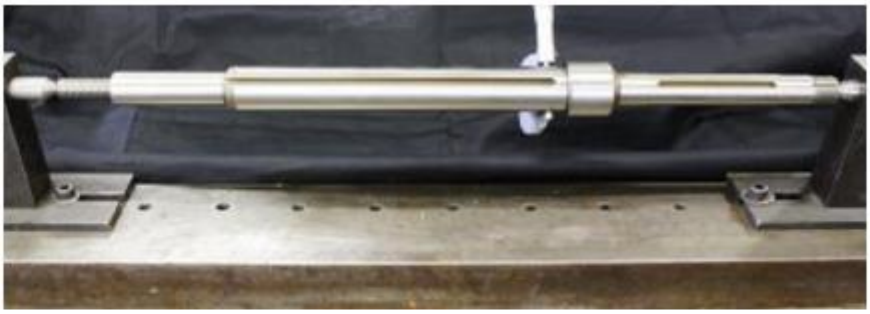


Figure 145.step measure the shaft closet to the tooling shoulder

Step 2: Next, measure the OD of the shaft in the area where the tooling is positioned (normally in the center area of the useable roll space), for this will be the area that most of the wear will occur.

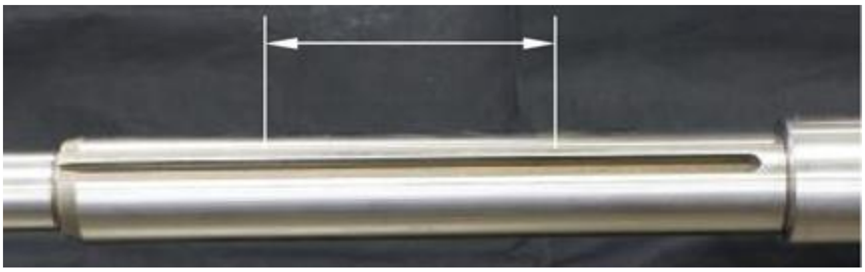


Figure 146

O.D. of Journal End

Check the O.D. of the journal end of the shaft. (If so equipped)

Step 3: Measure the O.D. of the journal end of the shaft for wear, but also look for variations from one end to the other such as taper.

The industry standard for maximum allowable undersize tolerance for the O.D. of a driven shaft is .001" per 1.000" shaft O.D. (Example, a 2.000" O.D. shaft, maximum undersize tolerance, .002", or 1.998" O.D.)

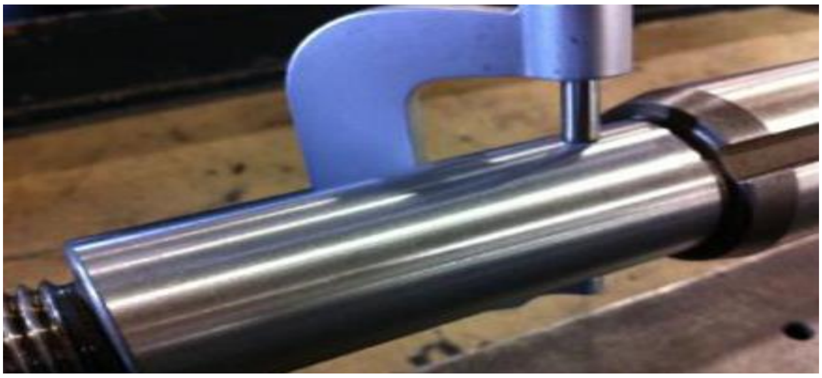


Figure 147.

Is the shaft bent?

Note: Before conducting this step (when the shaft is mounted on the mill)

make sure that both the inboard and outboard stands have been checked for loose bearings or bearing blocks to avoid the possibility of misidentifying a bent spindle.

Step 4: Measure for concentricity (run out) by placing the spindle between two centers (if not mounted on the mill) and using a dial indicator at the center area of the roll space, rotate the shaft around from one side of the keyway to the other and record the deviation.

The industry standard for the maximum allowable run out of a bent shaft is .001" per 1.000" shaft O.D. (Example, a 2.000" O.D. shaft, maximum run out, .002")

Step 4: Use dial indicator at the center portion of the shaft to check for runout

Figure 148

Rework Justification:

Driven shafts which have been earmarked for replacement from the previous guidelines can be reworked by grinding down the O.D. of the shafts, applying hard surface chrome, and grinding them back to their original O.D. (Keep in mind that the bearing journal surface damaged by a seized bearing can also be a candidate for hard surface chrome rework.)

Keyways

Chipped keyways are nearly impossible to repair. Many methods of welding and re-metalizing have been attempted, but subsequently failed, due to the fact that the process affects the hardness and integrity of the shaft.

Visually inspect keyways for chips

Figure 149

Shafts with chipped keyways that exceed the following guidelines should not be sent in for rework:

- Keyways chipped more than 25% of the total keyway depth in the shaft
  Example: 0.500" deep keyway in shaft
  25% of 0.500" = 0.125" maximum chip depth allowed
- Keyways chipped the length of the shaft that exceed twice the width of the keyway width
  Example: 0.750" wide keyway
  0.750" x 2 = 1.500" maximum chip length allowed

8. **Accumulation of dirt and other debris on exterior surfaces**

Accumulation of dirt and other debris on exterior surfaces with contain 4 Inspections follow:

- **Start-up Inspection**

  It is frequently said that the time a machine is most prone to failure is just after commissioning, major repairs or teardown. These episodes are critical states of change, and change presents risk from the standpoint of machine reliability.
- **Run Inspection**

  I’ve covered this subject in the last several *Machinery Lubrication* articles. Yes, there is a difference between Inspection 2.0 and conventional practices. An enhanced state of reliability demands an enhanced state of operator involvement. It is not just about quickly looking at a machine, but rather it’s about examining the machine frequently and intensely with a skilled, probing eye.
- **Stop Inspection**

  Stop inspections allow you to access those hard-to-reach machine conditions and frictional surfaces. Of course, as previously mentioned, you should avoid all unnecessary invasions that can introduce a root cause for failure.

  That said, you often can safely gain access to gear teeth, sump walls, couplings, shaft seals, bottom sediment and water (BS&W) bowls, magnetic plugs, bearing clearances, etc
- **Repair Inspection**

Repair and rebuild inspections present a valuable opportunity that too often goes untapped. What a wonderful learning opportunity failure can be. This not only involves what failed and why it failed, but also what else you can observe while you are performing the autopsy. Consider the following:
What are rebuild shops teaching you about the causes of failed electric motors, cylinders, gearboxes, pumps, etc

9. **Sealing of parts and small assemblies 'area and correct orientation (i.e., vertical versus horizontal), including**

Seal inspection is a particularly difficult and demanding application. With the advent of spiral wound, double-jacketed and kamm profile gaskets, gasket inspection has become a lot more difficult. The latest generation of gaskets are no longer manufactured from a simple 2D sheet, they are now a complex 2.5D or even 3D assembly made from a variety of flexible and solid materials

10. **Hydraulic cylinders, bearings, gearboxes, pumps, etc.**

- **Bearing inspection**

Bearing temperature generally rises with start-up and stabilizes at a temperature slightly lower than at start-up (normally 10 to 40 °Chigher than room temperature) in a certain time. The time before stabilizing depends on the size, type, speed, and lubrication system of the bearing and the heat dissipation condition around the bearing. It ranges from about 20 minutes to as long as several hours. If bearing temperature does not stabilize but continues to rise, High bearing temperature is not desirable in view of maintaining an adequate service life and preventing lubricant deterioration. A desirable bearing temperature is generally below 100 °C

- **Gearbox inspection**

Gearbox inspection is desirable in many situations, there may be constraints that limit the extent of the inspection such as cost, time, accessibility and qualified personnel. Cost and shutdown time might be perceived as prohibitive by management, but catching a problem in its earliest stages can save time and money in the long run. While it may seem too difficult to do a comprehensive inspection, a simple visual inspection of gear contact patterns through an inspection port can prevent future catastrophic failures. If in-house inspection expertise is not available, an expert can be hired to perform the inspection and train personnel.
Overcoming constraints in order to allow an inspection can help to extend gearbox life and avoid catastrophic failure. This might save time, money, injury to personnel and damage to adjacent equipment.
This article describes the equipment and techniques necessary to perform an on-site gearbox inspection.

Methods for Inspecting a Gearbox

- ✓ Visual walkaround
- ✓ Visual inspection through inspection ports
- ✓ Borescope inspection
- ✓ Measure temperature
    - Thermometers
    - Resistance temperature detector (RTD) probes
- ✓ Inspect magnetic debris collectors
- ✓ On-site analysis of lubricant
- ✓ Laboratory analysis of lubricant
- ✓ Magnetic particle inspection of gears
- ✓ Dye penetrant inspection of gears
- ✓ Documenting gear condition
    - Written

- Thermography
- Sketches
- Photography
- Contact patterns

✓ Measure oil pressure
✓ Measure sound and vibration
✓ Inspect filter elements

➢ **Pump inspection**

Pump inspection, we provide a full start-up report that compares the data sheet specs to your actual operational performance. You can then monitor any performance changes going forward. Regular preventive maintenance inspections address small issues before they become major problems. Regular maintenance will also help you troubleshoot the causes of issues and help to identify corrective actions to resolve and prevent these events from occurring in the future.

Content/Topic2: Start-up Inspection

It is frequently said that the time a machine is most prone to failure is just after commissioning, major repairs or teardown. These episodes are critical states of change, and change presents risk from the standpoint of reliability. When operators, mechanics and maintenance workers alter a machine, it is often difficult to precisely return it to the previous operating state. A great countermeasure to avoid start-up risk is thorough and continuous inspection along with condition monitoring. Respect all potential areas of danger. Inspect as many of these hazards as possible until operational stability is restored. These include:

2.1. Temperature (all critical zones, components and surfaces)

The temperature stability is essential to the success of mechanical systems. All hydraulic and lubricating fluids have practical limits on the acceptable operating temperature range both high and low level.

The machine loses stability and experiences conditional failure whenever the system's fluid temperature violates these limits. If left unabated, the conditional failure ultimately results in both material and performance degradation of machine components.

Fluid temperature instability is the result of various machine operating factors such as component integrity (design, selection, manufacture, application and maintenance), duty cycle severity (load application, magnitude and duration), environmental hostility and heat absorption/desorption.

✓ **Low-Temperature Effects**

Low temperature can damage the temperature stability of a hydraulic fluid or lubricant just as much as high temperature. Very low fluid temperatures usually result from exposure of some system part to the external environment, particularly when operation takes place in arctic or high-altitude conditions.

Such low temperatures can cause petroleum-based fluids to increase in viscosity and eventually reach the critical point where the fluid actually congeals and will no longer pour or flow.

✓ **High-Temperature Effects**

As industry continues to design systems of higher power density, fluid temperatures well above the current norms will become increasingly common. Such high-temperature conditions can disrupt the stability of conventional working fluids, compromise system performance and significantly reduce the life of operating components.

Fluid exposed to high temperature can experience permanent deterioration. For example, a substantial reduction in fluid viscosity normally accompanies asperity contacts (mechanical rubbing) and an increase in temperature.

2.2. Vibration

Vibration testing is an essential part of maintaining mission-critical machinery that powers your business. The foundation of all vibration testing is the data provided by the frequency of a motor or fan operating at its nominal RPM. An analyst working with a motor that turns at 900 RPM is looking for how the machine vibrates

the analyst must be skilled enough to detect minor differences in frequency in order to diagnose some of these common issues:

- ✓ Voltage imbalance
- ✓ Bad variable-frequency drive controller
- ✓ Eccentric stator
- ✓ Broken or bent rotor bars or rotor imbalance or rubs
- ✓ Coupling misalignment
- ✓ Worn bearings
- ✓ Soft foot
- ✓ Resonance
- ✓ Mechanical looseness

2.3. balance and alignment

Stricter rotor balancing standards are one of the biggest issues that we have been addressing for clients for the last couple of decades has been improperly balanced rotors. We have armed them with the proper balancing standards that they demand every one of their rotating assemblies (fans, pumps, motors, compressors, couplings, sheaves …) are balanced to with certifications from the manufacturers and contractors to prove this final balance condition of each rotor.

The reason we have provide this guidance was too many clients were having perpetual machinery failures which we found tied to poor initial rotor balance conditions. In an effort to cut costs, many balancing standards that were used successfully for nearly a century have all been loosened resulting in unbalanced rotors, premature machinery wear and frequent repairs.

Alignments Methods

The use of dial indicators in shaft alignment represents substantial step forward accurate alignment methods. One the earliest methods that uses dial indicators is the face and rim method in which two dial indicators are used to measure the total misalignment.

Advantages:

- Used when only one shaft can be rotated.
- Given the correct precautions, precision alignment is attainable with this method.

Disadvantages:

- End float affects face reading.
- Indicator bracket (bar) sag affects readings.
- Eccentric, skewed couplings or damaged surfaces will cause errors.
- Fixture looseness causes errors.

- Indicator stems not perpendicular to shaft causes errors.

2.4. Gauge readings temperature, pressure, vacuum, flow, speed, proximity, etc.)

More Applications for Differential Pressure Gauges

Like filters, other elements in a process system may degrade over time and may need to be monitored. Heat exchangers, pumps, valves, condensers and evaporators all create pressure drop when they become worn or clogged. Like filters, the problems they cause can be severe.

Differential Pressure in Flow

Flow is another common application for differential pressure gauges. There are countless types of flow meters in use today. The number one method for measuring flow is differential pressure flow monitoring. DP flow meters are especially attractive options for larger pipes. The cost of many flow meters goes up exponentially as the pipe size grows, because the scale of the flow element must increase to handle such large flows

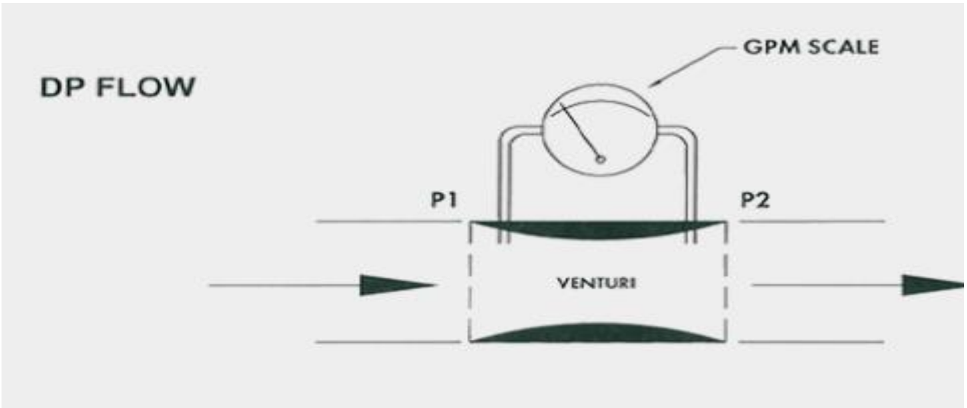


Figure 150.Differential Pressure in Flow

The popular solution is to cut the pipe, install an orifice plate and measure the pressure drop across the orifice plate. A DP flow meter is designed to handle the square root relationship between the flow and the differential pressure created, using Bernoulli's Law. This represents a high accuracy solution at a lost cost.

Like orifice plates, venturis and nozzles also create slight pressure drops that use differential pressure and square root flow to measure flow rates.

2.5. differential filter pressure

How Differential Pressure Gauges Work

Unlike straight pressure gauges, which measure pressure at a single point in a system, DP gauges measure the pressure at two points and display the pressure difference between those two points on a single dial. The pressure measured could be 115psi before a filter and 100psi after the filter, allowing the DP gauge to read the difference, which is 15psid – referred to as 15psi differential. That represents a 15psi pressure drop caused by the filter. The most popular application for differential pressure gauges is filtration. This application serves as an easy to understand explanation for the need for a differential pressure gauge. A filter removes unwanted particles or contaminants from a gas or liquid system. When a filter becomes clogged efficiency and pressure drops

2.6. Magnetic plug collections

Techenomics have been preforming magnetic plug inspections since 2002, covering equipment fleets such as haul trucks, rear dumps, dozers and loaders, the collected data has provided maintenance planners with the vital information directed at identifying iron wear generation, this service provides planners in sight information to the component life of wheel hubs, final drives and other planetary gearboxes. Preventing unnecessary failures is the key to cutting costs.

Preventative Maintenance is all about insight knowledge of the equipments compartments, Magnetic Plug Inspections provide this sight into the future by capturing wear particles on the end of a Magnetic Plug, these plugs sit inside the compartment, each time the oil passes the magnet, the magnet filters the oil by holding the wear particles, this stops wear generation from increasing dramatically caused by normal particle rubbing on internal components.

Magnetic wear is graded in levels of wear, level A being in good, clean condition, Level B being moderate amounts of debris and Level A being large amounts of debris being held on the plug, recommendations are then provided in the Online reporting via BLUE OCEANS.

2.7. Oil level, color and clarity at all sight glasses

The standard installed sight glasses on most pumps are often stained and discoloured over time, which makes inspecting the correct oil levels difficult.

A much more reliable option for ensuring proper oil levels is to use a 3D sight glass, permitting to view the oil level from any angle.

Figure 151.Oil level, color and clarity at all sight glasses

Investigations turn out that 60-70% of all lubrication-related problems are caused by contamination. With oil sight glasses the presence of contaminants, especially water, can easily be determined. Oil sight glasses allow users to isolate drain water and other contaminants from their oil. Water and sediment naturally sink to the bottom of the oil reservoir. Without any method to remove this, water will go unnoticed, and continue damaging your equipment and degrading the quality of your oil. Made from long-lasting, high strength acrylic, the oil side glass is available in a variety of sizes and styles. Their durable construction allows them to be installed in a range of environments, temperature and conditions. They are perfect for pumps, gearboxes, bearing houses and other lubrication critical pieces of equipment. Oil Sight Glasses will give operators the ability to catch small changes to the quality of the oil. Identifying small problems and taking corrective actions will prevent potential equipment failures.

When properly installed at the oil reservoir drain port, the Oil Sight Glass & Level Monitor allows the user to view the oil colour, clarity, and accumulation of water or other contaminates.

2.8. Leak zones

The technique of leakage monitoring requires the installation of flowmeters at strategic points throughout the distribution system, each meter recording the flows into a discrete area with a defined and permanent boundary. Such an area is called a District Meter Area (DMA).

The design of a leakage monitoring system has two aims:

- To divide the distribution network into a number of zones or DMAs, each with a defined and permanent boundary, so that night flows into each district can be regularly monitored, enabling the presence of unreported bursts and leakage to be identified and located.
- To manage the pressure in each district or group of districts so that the network is operated at the optimum level of pressure.

There are a number of techniques to detect where leakage is taking place in the network, including:

- ✓ sub-division of DMAs into smaller areas by temporarily closing valves or by installing meters;
- ✓ variations of the traditional step-test;
- ✓ the use of leak localizers;
- ✓ sounding surveys.

Table: Equipment for leak detection and location

equipment	Comments / Application	Limitations
'Basic' Listening Stick	Rudimentary sounding of SVs, FHs, MSTs etc.	Some smaller leak sounds may go undetected (good ear required by inspector).
'Electronic' Listening Stick	General sounding of SVs, FHs. MSTs etc. Better than 'Basic' Stick due to sound amplification. Is sometimes used to confirm 'best leak sound' position after correlation.	Few limitations, generally useful part of the inspectors 'tool kit'. Better than 'Basic' Stick, not as good as ground microphone (see below).
Electronic ground microphone	More sensitive than the electronic stick, generally used to confirm 'best leak sound' after correlation, powerful enough to listen to leak sounds through 'made roadways'. Can be used for general sounding with a probe screwed into microphone.	More 'cumbersome' to use than listening stick. Some inspectors do not like to use microphones, they prefer the electronic stick.
Electronic ground microphone with sound frequency filters	As sensitive as the ground microphone with the added advantage of the inspector being able to adjust filters and remove some unwanted sounds. Generally used to confirm 'best leak sound' after correlation. Powerful enough to listen to leak sounds through 'made roadways'. Can be used for	More 'cumbersome' to use than listening stick. Some inspectors do not like to use microphones. they prefer the electronic stick

	general sounding with a probe screwed into microphone.	
Acoustic Detection Loggers	'Stores' sounds within the distribution system usually between 02:00 and 04:00. Loggers are set up and downloaded using a PC. Leak sounds are identified by the 'range' of sounds recorded by the logger. Useful for areas where normal leak location activities cannot be used.	Does not locate actual leak position, can give identification that leak is taking place.
Step Test Unit	Mobile Advanced Step Tester (MAST) system. Used for remote monitoring of flows whilst carrying out step tests within distribution networks. Allows almost instant results of valve closure leading to minimum disruption to customers. Leak location activity can be carried out quickly rather than waiting for 'office based' analysis of step tests using dataloggers. Can also be used for remote monitoring of pressure during valve closure (critical node monitoring whilst setting up PMAs or DMAs).	Valve closure required. may cause discoloration / water quality problems. Difficult to use during day as some disruption to supplies will take place (unless areas are 'back fed' when valve closure takes place). Step tests need to be planned to gain best results.

- **Content/Topic3: Log book inspection checklist**

3.1 . Equipment or element identification

Inspections

Crane inspections are required at regular intervals. Inspection requirements are derived and defined in detail in Reclamation Safety and Health Standards, OSHA 1910, ANSI B30, State regulations, manufacturer's recommendations, and the rigging standards. Specific types of inspections are required on all cranes and hoists at prescribed intervals. The inspection criteria and interval differ between authorities and the duty cycle of the crane or hoist. For a more complete description of inspection techniques, requirements, and frequency, refer to the pertinent documents stated above for the type of equipment at your site. Each facility should develop an inspection program for each individual crane, hoist, fixture, and rigging that is based on the manufacturer's recommendations and all applicable standards. The general classifications of inspections are designated as "initial," "startup and daily," "frequent," and "periodic. Dated and signed inspection reports shall be kept on file and shall be readily available.

The inspection shall include the following functions:

(1) hoisting and lowering,

(2) trolley travel,

(3) bridge travel, and

(4) limit switches and locking and safety devices.

2. Start up and Daily Inspection On each shift, before operating the crane, the operator shall perform the following operations:

a. Test All Controls. Any controls that do not operate properly should be adjusted or repaired prior to the start of any operation.

4 b. Verify Operation of the Primary Upper-Limit Switch. The trip-setting of the primary upper limit switches shall be checked under no load conditions by inching the block into the limit (running at slow speed).

5 c. Visually Inspect Ropes and Load Chains. These visual observations should be concerned with discovering gross damage that may be a hazard.

6 d. Inspect hooks and latches for deformation, chemical damage, cracks, and wear.

7 e. Ensure inspections (wire rope, chains, and crane) are current via inspection sticker or other documentation.

Periodic Inspections A thorough inspection by a designated person requiring a record of the inspection as of apparent condition. a. Normal service – annually b. Heavy service – annually c. Severe service – quarterly.

Frequent Inspections A visual inspection by the user or other designated person with records not required to be maintained. a. Normal service – monthly Operating at less than 85 percent of rated load and not more than 10 lift cycles per hour except for isolated instances. b. Heavy service – weekly to monthly Operating at 85 to 100 percent of rated load or in excess of 10 lift cycles per hour as a regular specified procedure. c. Severe service – daily

Table 1.—Crane and hoist equipment inspection criteria

When to Inspect	Type of Inspection	Notes
Before initial use – new cranes	Initial inspection	Performed by manufacturer.
Before initial use – altered cranes	Initial inspection	Altered" is defined as any change to the original manufacturer's design configuration—that is, replacement of weight handling equipment, parts, or components with other parts or components. A qualified person must conduct this inspection.
Before initial use on a Reclamation project	Periodic inspection	"Initial use" refers to the first time Reclamation takes possession of and assembles a crane or whenever a non-Reclamation-owned crane is brought onto a jobsite and set up for use.
Before every operation (shift) Annually or as required by manufacturer (if more frequent)	Startup inspection Periodic inspection	If the hoisting equipment has not been in service, inspect prior to operation. However, do not use the equipment if you have not inspected it in more than 12 months.
Before using a crane which is not in use on a regular basis and which has been idle for more than 1 month but less than 6 months	Frequent inspection	Also inspect running ropes. Annual (periodic) inspection also applies.
Before using a crane that is not used on a regular basis and that has been idle for more than 6 months	Periodic and frequent inspection	Also inspect running ropes.
Standby cranes, at least semi-annually	Frequent inspection	Standby cranes are those not used regularly but are available, on a standby basis for emergencies (e.g., emergency operation and maintenance work); requirements for frequent inspections of standby cranes are in addition to the requirements for an annual (periodic) inspection.

a. Location of element or equipment

- ✓ Department
- ✓ Unit
- ✓ Section

LO 4.2 -Identify types of failure of elements

- Content/Topic1: Types of mechanical failure

1. Excessive deflection

The maximum load that may be applied to a member without causing it to cease to function properly may be limited by the permissible elastic strain or deflection of the member. Elastic deflection that may cause damage to a member can occur under these different conditions:

Elastic deflections that are the amplitudes of the vibration of a member sometimes associated with failure of the member resulting from objectionable noise, shaking forces, collision of moving parts with stationary parts, etc., which result from the vibrations.

2. Buckling

Buckling failure can be induced by external loading or by thermal conditions.

- Can involve elastic or plastic instabilities.
- Most dominant in columns and thin sheets subjected to compressive loads.

4. Ductile

When a ductile material has a gradually increasing tensile stress, it behaves elastically up to a limiting stress & then plastic deformation occurs. As stress is increased, the cross sectional area of the material is reduced & a necked region is produced. With a ductile material, there is a considerable amount of plastic deformation before failure occurs in the material, there is a considerable amount of plastic deformation before failure occurs in the necked region

5. Fracture

Fracture is the separation of a body into pieces subjected to stress. Fracture takes place whenever the applied loads (or stresses) are more than the resisting strength of the body. It starts with a crack that breaks without making fully apart. Fracture due to overstress is probably the most prevalent failure mechanism in mechanical/civil system and might be classified as ductile (shear) fracture, brittle

(cleavage) fracture, fatigue fracture, crazing, and de-adhesion

6. Impact

Can cause excess deformation or fracture. Impact or dynamic loading conditions that create high strain rates in metals tend to cause lower toughness and ductility.

7. Creep:

Creep can cause excess deformation or fracture. In metals it is most predominant at elevated temperatures.

Example: gas turbine engine blades due to centrifugal forces.

8. Relaxation

Relaxation is primarily responsible for loss of residual stress and loss of external load that can occur in bolted fasteners at elevated or ambient temperature.

9. Thermal shock

Thermal shock tends to promote cracking and/or brittle fracture.

Example: quenching operation during heat treatment of metals.

10. Wear

Wear Can occur at any temperature and include many possible failure mechanisms. It Dominant in roller or taper bearings and in gear teeth surfaces**.**

11. Stress corrosion

materials can be degraded by their environment by corrosion processes, such as rusting in the case of iron and steel. Such processes can also be affected by load in the mechanisms of stress corrosion cracking and environmental stress cracking.

12. Cracking

Crack growth can occur due to interaction with applied and/or residual stresses and the corrosive environment.

This interaction is called stress corrosion cracking, SCC, or environmental assisted cracking, EAC

13. Fatigue

Fatigue failures occur in every field of engineering.

For example, they can involve:

- thermal/mechanical fatigue failure in electrical circuit boards involving electrical engineers
- bridges involving civil engineers,
- automobiles involving mechanical engineers,
- farm tractors involving agricultural engineers,
- Aircraft involving aeronautical engineers, heart valve implants involving biomedical engineers,
- Pressure vessels involving chemical engineers, nuclear piping involving nuclear engineers.

L O 4.3 - Identify failure factors of elements

- Content/Topic1:Failure causes

1.1. Fracture:

Fracture is describing in various ways depending on the behavior of materials under stress upon the mechanism of fracture or even it appearance. The fracture can be classified either as ductile or brittle or brittle depend on whether or not plastic deformation of the materials before any catastrophe failure. A brief description of both types of fracture is given below**.**

✓ **Stress concentration**

Fracture: separation of a body into pieces due to stress, at temperatures below the melting point.

Steps in fracture:

crack formation

crack propagation

Depending on the ability of material to undergo plastic deformation before the fracture two fracture modes can be defined - ductile or brittle

Ductile fracture - most metals (not too cold):

Extensive plastic deformation ahead of crack

Crack is "stable": resists further extension unless applied stress is increased

Brittle fracture - ceramics, ice, cold metals:

Relatively little plastic deformation

Crack is "unstable": propagates rapidly without increase in applied stress

Stress Concentration

In order to break a small piece of material, one way is to make a small notch in the surface of the material and then apply a force. The presence of a notch, or any sudden change in section of a piece of material, can vary significantly change the stress at which fracture occurs. The notch or sudden change in section produces what **are called stress concentrations.** They disturb the normal stress distribution and produce local co-generations of stress. The amount by which the stress is raised depends on the depth of the notch, or change in section, and the radius of the tip of the notch. The greater the depth of the notch the greater the amount by which the stress is increased. The smaller the radius of the tip of the notch the greater the amount by which the stress is increased. This increase in stress is termed the **stress concentration factor**.

A crack in a brittle material will have quite a pointed tip and hence a small radius. Such a crack thus produces a large increase in stress at its tip. One way of arresting the progress of such a crack is to drill a hole at the end of the crack to increase its radius and so reduce the stress concentration. A crack in a ductile material is less likely to lead to failure than in a brittle material because a high stress concentration at the end of a notch leads to plastic flow and so an increiLse in the radius of the tip of the notch. The result is then a decrease in the stress concentration.

Fracture strength of a brittle solid is related to the cohesive forces between atoms. One can estimate that the theoretical cohesive strength of a brittle material should be ~ E/10. But experimental fracture strength is normally E/100 - E/10,000.

This much lower fracture strength is explained by the effect of **stress concentration** at microscopic flaws. The applied stress is amplified at the tips of micro-cracks, voids, notches, surface scratches, corners, etc. that are called **stress raisers**. The magnitude of this amplification depends on micro-crack orientations, geometry and dimensions.

Stress Concentration

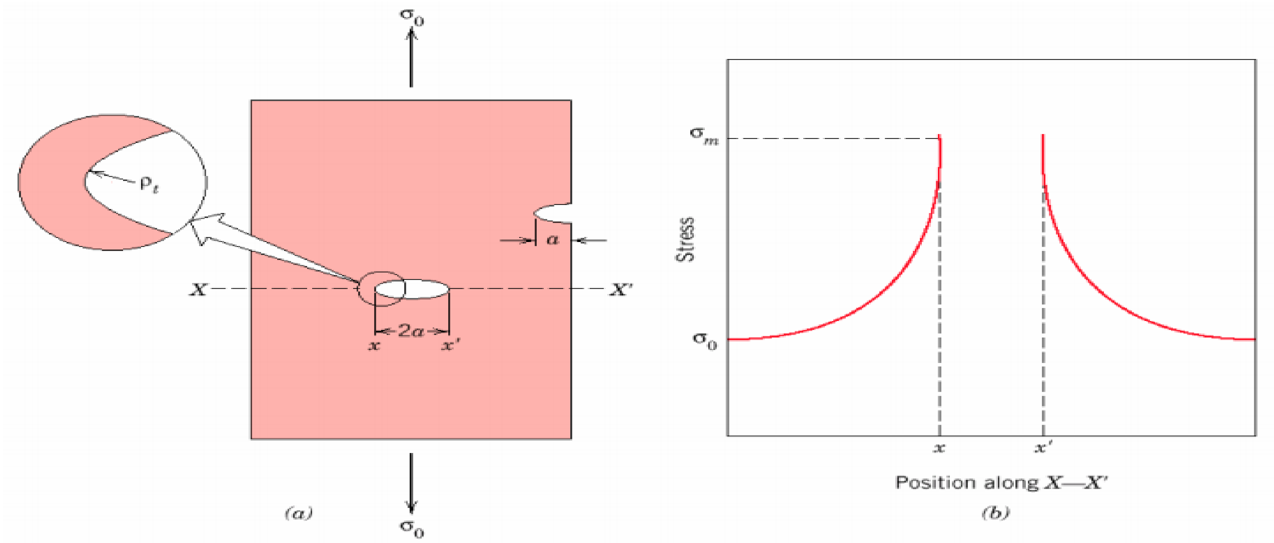


Figure 152

For a long crack oriented perpendicular to the applied stress the maximum stress near the crack tip is:

$$\sigma_m \approx 2\sigma_0\left(\frac{a}{\rho_t}\right)^{1/2}$$

where **σ0** is the applied external stress, a is the half-length of the crack, and **ρt** the radius of curvature of the crack tip. (note that a is half-length of the internal flaw, but the full length for a surface flaw). The stress concentration factor is:

$$K_t = \frac{\sigma_m}{\sigma_0} \approx 2\left(\frac{a}{\rho_t}\right)^{1/2}$$

- ✓ **Speed of loading**

Another factor which can affect the fracture of a material is the speed of loading. A sudden blow to the material may lead to fracture where the same stress applied more slowly would not. With a very high rate of application of stress there may be insufficient time for plastic deformation of a material to occur under normal conditions, a ductile material will behave in a brittle manner.

Fracture strength, also known as **breaking strength**, is the stress at which a specimen fails via fracture.[2] This is usually determined for a given specimen by a tensile test, which charts the stress–strain curve . The final recorded point is the fracture strength.

Ductile materials have a fracture strength lower than the ultimate tensile strength (UTS), whereas in brittle materials the fracture strength is equivalent to the UTS.If a ductile material reaches its ultimate tensile strength in a load-controlled situation, it will continue to deform, with no additional load application, until it ruptures. However, if the loading is displacement-controlled the deformation of the material may relieve the load, preventing rupture.

In brittle crystalline materials, fracture can occur by *cleavage* as the result of tensile stress acting normal to crystallographic planes with low bonding (cleavage planes). In amorphous solids, by contrast, the lack of a crystalline structure results in a conchoidal fracture, with cracks proceeding normal to the applied tension.
The theoretical strength of a crystalline material is (roughly)

$$\sigma_{\text{theoretical}} = \sqrt{\frac{E\gamma}{r_o}}$$

where: –

E is the Young's modulus of the material,
γ is the surface energy, and
r_o is the equilibrium distance between atomic centers.

$$\sigma_{\text{elliptical crack}} = \sigma_{\text{applied}} \left(1 + 2\sqrt{\frac{a}{\rho}}\right) = 2\sigma_{\text{applied}} \sqrt{\frac{a}{\rho}} \text{ (For sharp cracks)}$$

σ_{applied} is the loading stress,
a is half the length of the crack, and
ρ is the radius of curvature at the crack tip.
Putting these two equations together, we get

$$\sigma_{\text{fracture}} = \sqrt{\frac{E\gamma\rho}{4ar_o}}.$$

The basic steps in ductile fracture are void formation, void coalescence (also known as crack formation), crack propagation, and failure, often resulting in a cup-and-cone shaped failure surface. Voids typically coalesce around precipitates, secondary phases, inclusions, and at grain boundaries in the material. Ductile fracture is typically transgranular and deformation due to dislocation slip can cause the shear lip characteristic of cup and cone fracture

- ✓ **Temperature**

The temperature of a material can affect its behavior when subject to stress. Many metals which are ductile at high temperatures are brittle at low temperatures. For example, steel may behave as a ductile material above, say, O oc but below that temperature it becomes brittle. The ductile—brittle transition temperature is thus of importance in determining how a material will behave in service. The transition temperature with steel is affected by the alloying elements in the steel. Manganese and nickel reduce the transition temperature. Thus for low-temperature work, a steel with these alloying elements is to be preferred. Carbon, nitrogen and phosphorus increase the transition temperature.

But belong the tools ,it’s concern follow :when the amount ot heat generated at the cutting edge of the tool became excessive ,the heat causes the tool to soften and failure occurs. Excessively high cutting speed and heavy cuts will causes some types of tools to soften, with resulting wearing away and rupturing of the cutting edge. This types of failure occurs quite rapidly rapidly after reaching a certain point.

- ✓ **Thermal shock**

When hot water is poured into a cold glass it causes the glass to crack which is known as thermal shock. The layer of glass in contact with the hot water tends to expand but is restrained by the colder outer layers of the glass, these layers not heating up quickly because of the poor thermal conductivity of glass. The result is the setting up of stresses which can be sufficiently high to cause failure of the brittle glass.

1.1. Creep failure causes

High temperature above 40% of melting temperature of material

Creep occurs under certain load at elevated temperature normally above 40 % of melting temperature of the material. Boilers, gas turbine engines, and ovens are some of the examples whereby the components experiences creep phenomenon. An understanding of high temperature materials behavior over a period of time is beneficial in evaluating failures of component due to creep. Failures involving creep are usually easy to identify due to the deformation that occurs. A typical creep rupture envelops. Failures may appear ductile or brittle manner due to creep. Cracking may be either trans granular or inter-granular, if creep testing is done at a constant temperature and load, actual components may experience damage or failure at various temperatures and loading conditions.

In a creep test, a constant load is applied to a tensile specimen maintained at a constant temperature. Strain is then measured over a period of time. The slope of the curve, is the strain rate of the test during stage 11 or the creep rate of the material. Primary creep (known as stage I) is a period of decreasing creep rate. Primary creep is a period Of primarily transient creep. During this period deformation takes place and the resistance to creep increases until stage 11_ Secondary creep (or stage Il) is a period of approximate constant creep rate- Stage LI is referred to as steady state creep. Tertiary creep (stage Ill) occurs when there is a reduction in cross sectional area due to necking or effective reduction in area due to internal void formation. Subsequently, increase in creep rate leading to the creep fracture or stress rupture.

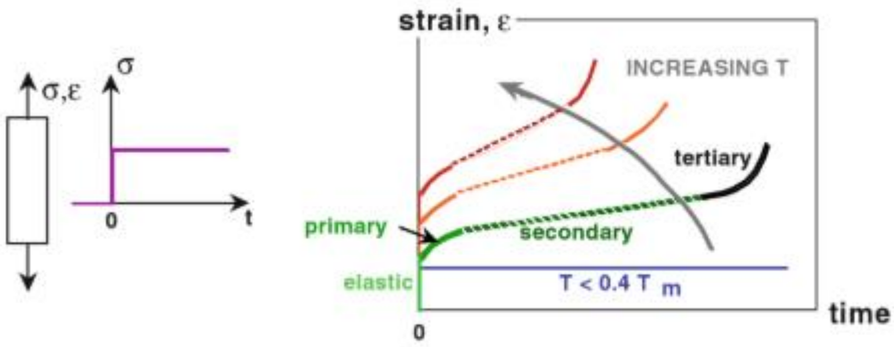


Figure 153.creep

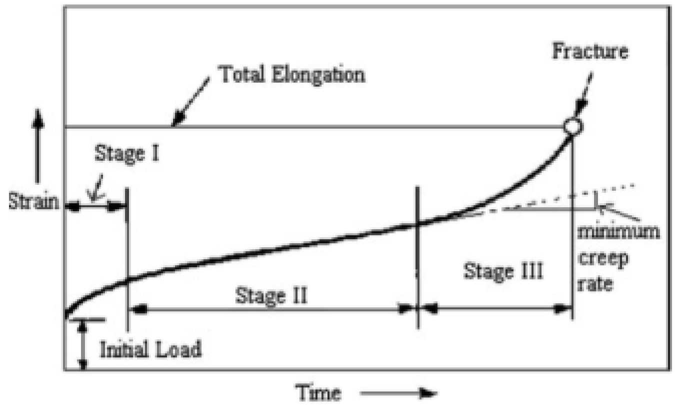


Figure 154strain rate

1. **Fatigue failure cause**

Metal fatigue is caused by repeated cycling of the load. It is a progressive localized damage due to fluctuating stresses and strains on the material. Metal fatigue cracks initiate and propagate in regions where the strain is most severe. Figure 2.5 shows typical S—N curve for the fatigue strength of a metal.

The process of fatigue consists of three stages:

- Initial crack formation
- Progressive crack growth across the part
- Final but sudden fracture of the remaining cross section

Prevention Fatigue failure cause

The most effective method of improving fatigue performance is improvements in design. The following design guideline is effective in controlling or preventing fatigue failure:Fig. Schematic of S-N Curve showing increase in fatigue life with decreasin

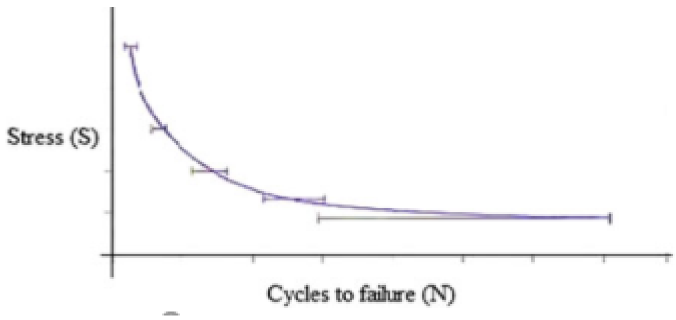


Figure 155

- Eliminate or reduce stress raisers by streamlining the part or component.
- Avoid sharp surface tears resulting from punching, stamping, shearing, or other processes.
- Prevent the development of surface discontinuities during processing.
- Reduce or eliminate tensile residual stresses caused by manufacturing.
- Improve the details of fabrication and fastening procedures.

✓ **Repeated cycling of the load and Fluctuating stress strains on the material**

Cyclic loading is the application of repeated or fluctuating stresses, strains, or stress intensities to locations on structural components. The degradation that may occur at the location is referred to as fatigue degradation
Some machine elements are subjected to static loads and for such elements static failure theories are used to predict failure (yielding or fracture). However, most machine elements are subjected to varving or

fluctuating stresses (due to the movement) such as shafts, gears, bearings, cams & followers,etc.

Fluctuating stresses (repeated over long period of time) will cause a part to fail .(fracture) at a stress level much smaller than the ultimate strength (or even the yielc strength in some casses).

Unlike static loading where failure usualy can be detected before it happens (due to the large deflections associated with plastic deformation), fatigue failures are usualy sudden and therefore dangerous.

Fatigue failure is somehow similar to brittle fracture where the fracture surfaces are prependicular to the load axis.

According to Linear-Elastic Fracture Mechanics (L EFM), fatigue failure develops in three stages:

Staqe: development of one or more micro cracks (the size of two to five grains) due to the cyclic local plastic deformation.

Staqe2: the cracks progress from micro cracks to larger cracks (macro cracks) and keep growing making a smooth plateau-like fracture surfaces with beach marks.

Staqe3: occurs during the final stress cvcle where the remaining material cannot support the load, thus resulting in a sudden fracture (can be brittle or ductile fracture).

• Fatigue cracks can also initiate at surfaces having rough surface finish or due to the presence of tensile residual stresses. Thus all parts subjected to fatigue loading are heat treated and polished in order to increase the fatigue life.

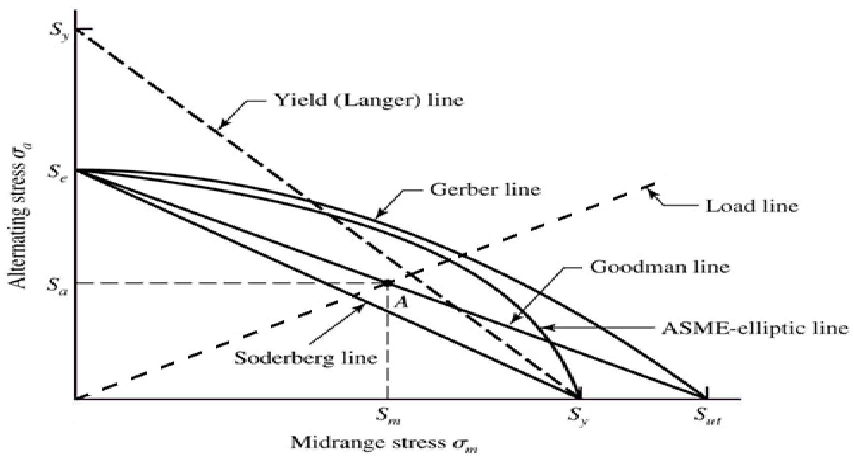
Fatigue failure is due to crack formation and propagation. Fatigue cracks usually initiate at locations with high stresses such as discontinuities (hole. notch, scratch, sharp corner, crack, inclusions. etc.

Fatigue Failure Criteria for Fluctuating Stress

When a machine element is subjected to completely reversed stress (zero mean, am = O) the endurance limit is obtained from the rotating-beam test (after applying the necessary modifying factors).

However, when the mean (or midrange) is non-zero the situation is different and a fatigue failure criteria is needed.

• If we plot the alternating stress component (aa) vs. the mean stress component (am), this will help in distinguishing the different fluctuating stress scenarios.



- When $\sigma_m = 0$ & $\sigma_a \neq 0$, this will be a <u>completely reversed fluctuating stress</u>.
- When $\sigma_a = 0$ & $\sigma_m \neq 0$, this will be a <u>static stress</u>.
- Any <u>combination</u> of σ_m & σ_a will fall between the two extremes (*completely reversed & static*).

<u>Different theories</u> are proposed to predict failure in such cases:

<u>Yield (Langer) line</u>:It connects Sy on the aaaxis with Sy on axis. But it is not realistic because Sy is usually larger than Se.

<u>Soderberg line</u>:The most conservative, it connects Se on axis with Sy on axis.

- $$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = \frac{1}{n}$$ Where (n) is the design factor

ASME-elliptic line: Same as Soderberg but it us line.

Same as Soderberg but it uses an ellipse instead of the straight

It fits $$\left(\frac{n\sigma_a}{S_e}\right)^2 + \left(\frac{n\sigma_m}{S_y}\right)^2 = 1$$ experimental data better (see fig 6-25)

- Goodrnan line: It considers failure due to static loading to be at Sut rather than Sy, thus it connects Se on axis with Sut on axis using a straight line.

- $$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n}$$

Gerber line:Same as Goodman but it uses a parabola instead of the straight line.

$$\frac{n\sigma_a}{S_e} + \left(\frac{n\sigma_m}{S_{ut}}\right)^2 = 1$$

- The factor of safety is found as: $n_f = \frac{1}{2}\left(\frac{S_{ut}}{\sigma_m}\right)^2 \frac{\sigma_a}{S_e}\left[-1 + \sqrt{1 + \left(\frac{2\sigma_m S_e}{S_{ut}\sigma_a}\right)^2}\right]$

It should be noted that Se is the modified endurance limit.

The fatigue stress concentration factor (Kr) should be multiplied with both aa & $\sigma_m$for conservative results.

The load line represents any combination of aa and ,the intersection of the load line with any of the failure lines gives the limiting values Sa and Sm according to the line it intercepts.

2. Corrosion causes

Corrosion of metallic materials occurs in a number of forms which differ in appearance. Failure due to corrosion is a major safety and economic concern. Several types or corrosion are encountered in metallic materials, among those: general corrosion, galvanic corrosion, crevice corrosion, pitting, intergranular, stress corrosion etc. This can be controlled using galvanic protection, corrosion inhibitors, materials selection, protective coating and observing some design rules.

Corrosion is chemically induced damage to a material that results in deterioration of the material and its properties. This may result in failure of the component. Several factors should be considered during a failure analysis to determine the effect Of corrosion in a failure. Examples are listed below:

- type of corrosion
- Corrosion rate
- The extent of the corrosion
- Interaction between corrosion and other failure mechanisms.

❖ **improper lubrication of material**

Bearings fail due to material fatigue with the bearing having reached the end of its service life. Industry sources vary with respect to the reasons for premature bearing failure however all are in agreement that incorrect lubrication is the main cause. Conservative industry estimates of the reasons for bearing failures are:

- Improper lubrication / lubrication failure: — 50%
- Improper mounting: —30%
- Other causes: 20% (such as harsh operating conditions, faulty storage and handling)
- Natural fatigue limit reached: <1070

Proper lubrication prevents metal-to-metal contact between the rolling elements. raceways and retainers: it protects the bearing against corrosion and wear, helps dissipate heat, seals out solid and liquid contamination, and reduces bearing noise. A bearing will not reach its maximum service life without proper lubrication.

There are many causes for lubricant or lubrication failure, including:

- Insufficient lubricant quantity
- Deterioration due to prolonged service without replenishment
- Excessive operating temperatures
- Contamination with foreign matter
- Water contamination from humidity, condensation or other liquid contamination •Incorrect grease for a particular application
- Over- lubricating — which causes rapid temperature increases, churning, and may cause oxidation of the oil and breakdown of the grease

❖ **poor coatings**

Coatings are a complex combination of raw materials that must be mixed, applied to a prepared substrate, and dried and cured correctly to perform to their maximum capability. They must be able to be applied in diverse environmental conditions and then be expected to protect the substrate from the damaging effects of rain, sunlight, wind, heat, cold, humidity, and oxygen in various combinations and cycles and still retain their integrity and often their aesthetic qualities.

Coating Failures and Defects

9. Abrasion

The mechanical action of rubbing, scraping, scratching, gouging, or erosion.

Probable Causes: Removal of a portion of the surface of the coating or, in severe cases, removal to expose the substrate by contact with another object, such as the use of metal chains for lifting, cargo, fenders, or the grounding of a ship.

Prevention: Use of abrasion-resistant coatings formulated with particular regard to resins and extender pigments. With severe cases of abrasion, the effects will be reduced or limited only by an abrasion resistant coating.

10. Adhesion Failure

Probable Causes: Surface contamination or condensation, incompatibility between coating systems, or exceeding the overcoating time.

Prevention: Ensure that the surface is clean, dry, and free from any contamination and that the surface has been suitably prepared. Use the correct coating specification and follow the advised overcoating times.

11. Alligatoring (Crocodiling).

Probable Causes: Internal stresses in the coating where the surface shrinks faster than the body of the paint film. Excessive film thickness and limited paint flexibility. Application of a hard topcoat over a more flexible softer undercoat. Application of topcoat before the undercoat has dried.

Prevention: Use correct coating specification and compatible materials. Avoid excessive film thickness. Avoid application at high ambient temperatures.

12. Bleeding.

Probable Causes: Bleed through is generally a full or partial redissolving of the previous coat or an ingredient of a previous coat and can occur when strong solvents are used in the topcoats.

Prevention: Use correct coating specification and materials. Use compatible materials. Use appropriate sealer coat if possible.**Etc..**

- **<u>Improper designs</u>**

Reasons Structures Fail

Defective construction that causes failure may be due to numerous reasons that may not be easy to predict before or during the construction. The major causes of structural failure are defective designs that have not determined the actual loading conditions on the structural elements. Inferior construction materials may also be the cause since the loads are calculated for materials of specific characteristics.

Design Deliberations

Construction imperfection in design and manufacturing can be extremely expensive to settle. Architectural design and construction defects cause a structure to be improper for its proposed intent. Correct structural design is significant for all buildings, but exceptionally essential for tall buildings. Even a slight probability of failure is not acceptable since the results can be disastrous for human life and property. Therefore, civil engineers are required to be exceptionally careful and methodical in ensuring an appropriate building design that can sustain the applied loads.

Foundation Failure

Many building foundations are not properly designed and constructed for the existing site soil conditions. Since suitable land is often not available, buildings are constructed on soil that has inadequate bearing capacity to support the weight of the structure. Furthermore, the near surface soils may consist of expansive clays that shrink or enlarge as the moisture content is changed. Movement of foundation may occur if the clay moistening and drying is not uniform. Vegetation, inadequate drainage, plumbing leakage, and evaporation, may cause soil variation.

Design Deliberations

Construction imperfection in design and manufacturing can be extremely expensive to settle. Architectural design and construction defects cause a structure to be improper for its proposed intent. Correct structural design is significant for all buildings, but exceptionally essential for tall buildings. Even a slight probability of failure is not acceptable since the results can be disastrous for human life and property. Therefore, civil engineers are required to be exceptionally careful and methodical in ensuring an appropriate building design that can sustain the applied loads. All failure modes need to be examined by using modern software on the subject.

- ❖ **<u>Changing the environment</u>**

Most standard commercial property insurance policies contain the following basic exclusions:

- ✓ Explosion of steam boilers, steam engines, steam turbines, or vessels under steam pressure;
- ✓ Artificially generated electric currents; - arcing, or short circuiting – of motors, generators, circuit breakers, electrical distribution boards, cables, and transformers;
- ✓ Mechanical breakdown, and
- ✓ Centrifugal force

Any loss (such physical damage to the equipment, business interruption, spoilage, etc.) resulting from these causes of failure might not be covered by a property insurance policy. As a result, most industrial, utility, commercial, institutional, processing and light manufacturing risks carry the so called Equipment Breakdown Insurance. Many insureds carry insurance for boilers and air conditioning, while ignoring to also insure other equipment risks such as transformers, generators, pumps, compressors, and so on.

Figure 156

Example of arcing damage to equipment not covered by standard property damage policies

Typical commercial property insurance will provide property damage coverage by including the peril of accidental breakdown that is:

i. Sudden;
ii. Accidental;
iii. Manifests itself in physical damage to the equipment and necessitates repair or replacement of the equipment or part thereof

Figure 157

important restrictions to the above definition of the covered peril are: Water damage caused by worn water supply pipes

- ✓ Wear and tear;
- ✓ Cracking of certain parts of gas turbines;
- ✓ Leakage at valves, seals or fittings;
- ✓ Corrosion of the equipment components;
- ✓ Depletion, deterioration or erosion of the equipment;
- ✓ Failure of a safety devise, such pressure or vacuum relief valve;
- ✓ Breakdown of certain electronic components;
- ✓ Combustion explosions;
- ✓ Faulty or improper material, workmanship or design;
- ✓ Pollution or contamination;
- ✓ Gradual deterioration, latent defect or inherent vice

From the above list of exclusions it can be seen clearly that an Equipment Breakdown Insurance is an essential insurance for all properties that use equipment for heating, cooling, process, etc. Equipment Breakdown Insurance is a form of property damage insurance and its purpose is to insure against the financial losses, such as property damage, business interruption, extra expense and spoilage (consequential damage) losses that result from defined accidents to specified kinds of mechanical, electrical and pressure equipment.

The cause of damage investigation

During an investigation of a loss caused by equipment or machinery failure, the most important questions that always must be answered by an investigating expert are:

- ✓ What is the cause or causes of the failure (loss) and how it happened;
- ✓ If there are multiple causes of failure, is there a direct (or proximate) cause, and what is the sequence of the failure events?
- ✓ Are there any contributing factors that led to the failure (loss)?

Equipment failure claims can be very complicated because of the magnitude of the loss, the age of equipment, the lack of maintenance or repair records, any prior damages or electrical or mechanical failures, any product recalls or defects and so on. These types of losses also present significant subrogation potential, i.e., trying to recover the loss from a responsible third party or parties.

Failed boiler tube *Pitting corrosion in a boiler tube*

Figure 158

13. Wear causes

- <u>**Removal or displacement of material by mechanical action of a contacting solid, liquid, or gas**</u>

Material removal in most cases takes place gradually due to a repetitive motion, but wear is a complicated process and influenced by several parameters. Among these are the following:

- ✓ contact geometry
- ✓ length of exposure
- ✓ environmental conditions
- ✓ material composition and hardness
- ✓ interacting material surfaces
- ✓ normal force

Types of Wear

It is the type of relative motion that is used to define the generated wear. The complexity of wear means that a number of modes can be recognized.

- ✓ **Abrasive Wear** - Wear because of hard protuberances or hard particles forced against and moving along a solid surface is known as abrasive wear. These hard particles may be common abrasives such as silicon carbide and aluminum oxide, or naturally occurring contaminates such as dust particles and sand [crystalline silica (quartz)].
- ✓ **Adhesive wear** – Adhesive wear is caused by localized bonding between contacting solid surfaces resulting in material transfer between the two surfaces or loss from either surface. Adhesive wear is not as common as abrasive wear and takes place when materials slide against each other without any lubrication.
- ✓ **Catastrophic wear** – Accelerating or rapidly occurring surface damage, deterioration or change of shape caused by wear to the degree that the component's service life is considerably shortened or function destroyed.
- ✓ **Corrosive wear** – In this kind of wear, electrochemical or chemical reactions with the environment are significant.

- ✓ **Crocking** – Color transfer from a colored fabric surface to an adjacent area of the same fabric or to another surface by rubbing action.
- ✓ **Cutting wear** – In solid impingement erosion, cutting wear is the erosive wear related to the dissipation of kinetic energy of impact arising from the tangential component of the velocity of the impacting particles.
- ✓ **Deformation wear** – In solid impingement erosion, deformation wear is the erosive wear of a material related to the dissipation of kinetic energy of impact arising from the normal component of the velocity of the impacting particles. It is hence the sole component of wear for particles impacting at a 90° angle of attack.
- ✓ **Erosion** – This is the damage caused by particulate in liquids or gases striking a surface.
- ✓ **Rolling wear** – This includes wear because of the relative motion between two non-conforming solid bodies whose surface velocities in the nominal contact location are identical in magnitude, direction and sense.
- ✓ **Rolling abrasion** – This is an abrasion form that takes place when debris or abrasive particles or debris are allowed to "roll" between the surface and a contacting substance.

➢ **<u>Contamination of Lubricant by solid particles</u>**

presence solid contaminants in lubricating oils responsible for the failure of machine parts. Contamination of lubricating oils causes wear of piston ring, which generates more contamination. This proceeds via internal wear generating fresh wear debris leading to the opening of the dynamic sealing surfaces. Bore polishing is another undesired impact of small abrasive particles the lubricating oils, while larger particles can cause scratches in the bore. The total friction in an engine immediately after a cold-start is four to five times higher than at fully warmed-up conditions. The source of the contamination particles may be soot from combustion, silica dust and similar minerals, and wear particles consisting of ferrous, lead, chromium, copper, aluminum, nickel alloys and tin The diesel soot interacts with lubricating oils and ultimately leads to wear of engine parts.

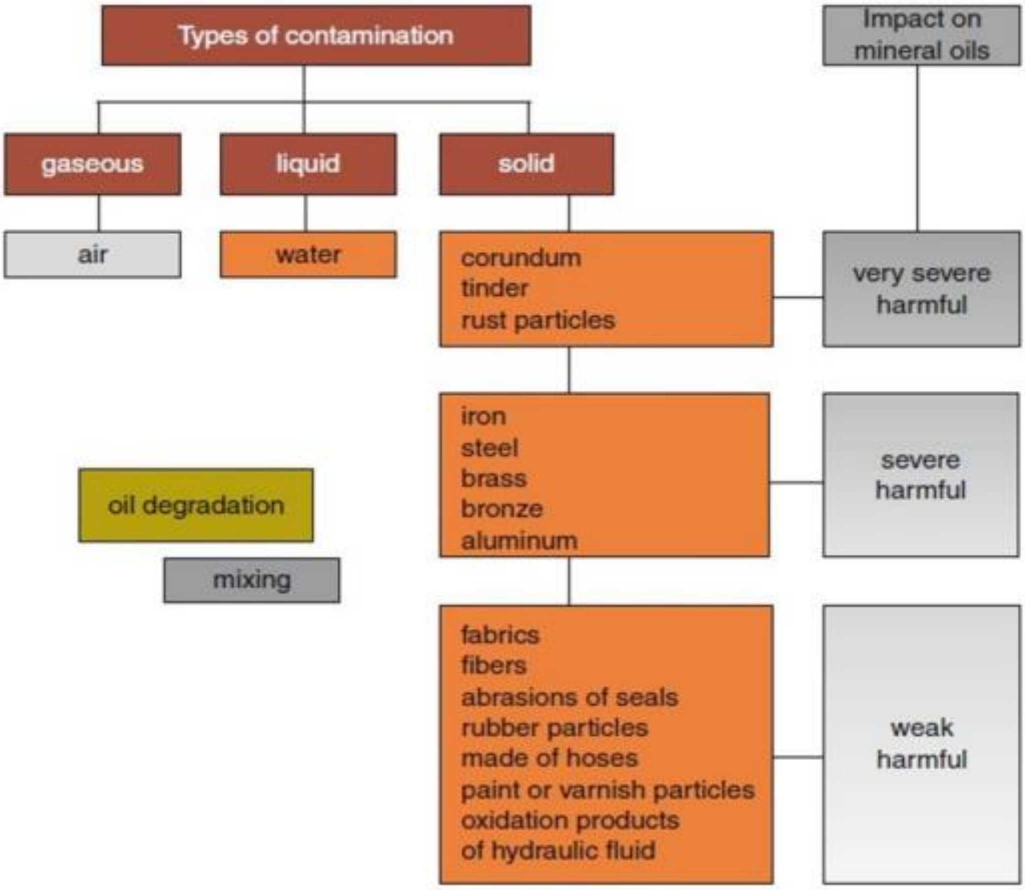


Figure 159Contamination types in lubricating oils and Harmful effects

The harmful effect of solid particles in the lubricating oils is particularly obvious with softer parts like piston skirts, bearings, and cam-follower contacts. The major categories of solid contaminant particles in crankcase oils are carbon, or combustion particles and metallic or wear particles.

The carbon contaminants particle can be determined by means of Fourier transform infra-red spectroscopy (FTIR), a method that is suitable for the determination of the presence of different organic compounds including reaction products. It is necessary for all lubricated systems that the particle size should remain well below the oil film thickness between surfaces in any lubricated mechanism, and this very well applies to engines. Oil filters in experimental applications can have nominal retention rates in the order of 15 to 20 µm. Lubricant oil in the region of the piston rings is far more contaminated than the oil in the engine sump. Exhaust gas recirculation without soot filters cause an increase in the level of carbon particles in the lubricating oil.

L O 4.4 -Select potential failure solutions

- Content/Topic1:Risk management failure of equipment

3. Regular inspection

Risk management is at the heart of business success. It basically boils down to "protect your backside," and companies that don't do that, don't last long.

Consider these business risks:

- ✓ Equipment that must be prematurely replaced due to excessive wear and tear.
- ✓ Productivity losses and operational delays due to equipment failures.
- ✓ Potentially enormous costs (including liability exposure) of on-the-job accidents due to defective equipment.

These are four business processes companies can implement to minimize the risk of equipment failure.

*1. **Regular inspections**.*

Whether it's a chainsaw or a multimillion-dollar crane, it's common sense that thoroughly inspecting equipment before each use is the best way to ensure that it won't fail during operation.

With an easy-to-use, equipment-specific inspection checklist system such as The Checker, you can ensure pre-use inspections are done properly and documented.

*3. **Preventative maintenance.***

Companies that sufficiently appreciate the negative financial impact of equipment failure don't wait until equipment fails to give it attention. They have a formalized maintenance process to stay on top of the routine care the equipment requires and replace parts according to schedule.

They also train personnel on how to use the equipment in as non-destructive a way as possible which technically may not be "maintenance" but surely helps reduce wear on equipment. (Because it's human nature to care more about things you're responsible for, the process of having personnel conduct regular pre-use inspections usually leads to gentler treatment of equipment.

Learning Unit 5 -Report the machine elements maintained

L O 5.1 -Record data of elements conditions

- **Content/Topic1: Data records for parts maintained**

1. **Task**

1.1. Oiling

Lubricating oils are commercially available under different trade names. Indian Oil Corporation[2] manufactures a wide range of lubricating oils and greases. These commercial lubricants contain a base oil and a group of additives, which are suitable for a given application. The properties of lubricating oils used for automotive applications are given in Table 16.6. There are five grades **from SAE 10 to SAE 50.** These base oils are blended with viscosity index improver, detergent additives and oxidation inhibitors. These oils are used for engine lubrication of petrol and diesel vehicles. They are also used for generators and pumping sets operating on diesel engines. There are two different classes of crankcase oil—Servo Engine Oil and Servo Super.

The second is superior and costly. It is used for heavy duty internal combustion engines.

Properties of lubricating oils for automotive crankcase applications

Properties	*Servo Engine Oil/Servo Supper*				
	10	*20*	*30*	*40*	*50*
1. SAE Grade	10 W	20	30	40	50
2. Kinematic viscosity (cSt at 100°C)	5 (min)	6–8	10–12	13–15	18–20
3. Viscosity index (min)	100	95	95	90	90
4. Flash point (°C) (min)	190	200	220	225	230
5. Pour point (°C) (max)	–27	–21	–6	–6	–6

Commercial lubricating oils for gears consist of SAE 80, SAE 90 and SAE 140 as base oils and a mixture of extreme-pressure additives, oxidation inhibitors and oiliness additives. The properties of these oils are given.

The following are the advantages of these oils:

(i) They have excellent chemical stability even at high temperatures.

(ii) They can withstand extremely high local pressures in high-torque and low-speed conditions.

(iii) They protect gear assemblies against rust and corrosion.

They are used for the lubrication of all types of gears.

Properties of gear oils

Properties	*Servo Gear HP / Gear Super*		
	80	*90*	*140*
1. SAE Grade	80	90	140
2. Kinematic viscosity (cSt at 100°C)	9–11	16.5–18.5	31–33
3. Viscosity index (min.)	85	85	80
4. Flash point (°C) (min.)	165	180	190
5. Pour point (°C) (max.)	–27	–9	0

1.1. Greasing

Grease is a semisolid substance, composed of mineral oil and soap. Sometimes additives are added to this mixture to achieve specific properties, such as chemical stability or oilness. The soap is present in the form of fibres, which form a matrix for the oil by the swelling mechanism.

the grease in the clearance space is quite rigid and immobile but when the journal starts rotating, the viscosity of the grease approaches to that of the base oil in the grease. Grease is normally recommended for inaccessible parts, where leakage of oil is objectionable. It is also used in applications where clearance is large due to rough machining.

Greases are classified on the basis of the soap employed. *Lime-base grease* consists of calcium soap in mineral oils of grades **SAE 10 to SAE 40.** It is insoluble in water. It is buttery and offers resistance to flow. However, it has a tendency to channel and separate out from the machine component by centrifugal action. It is, therefore, not suitable for ball bearings. It is used for chassis parts including suspension, and for steering systems of vehicles. It is also used for open or semi-enclosed gears and chains. *Soda-base grease* is produced from sodium soap. It has more resistance to decomposition at high temperatures and pressures. Soda-base grease is water-soluble and possesses a sponge-like structure. It does not have a tendency to channel. It is mainly used for the lubrication of automotive wheel bearings. Lithium soap grease has excellent resistance to oxidation, and is used for water pumps, wheel bearings and chassis fittings.

1.2. Alignment

shaft alignment is the process whereby two or more_machine (typically a motor and pump) positioned such a' the point of power transfer from one shaft other. the axes of rotation of both should be collinear when the machine is running under normal conditions.

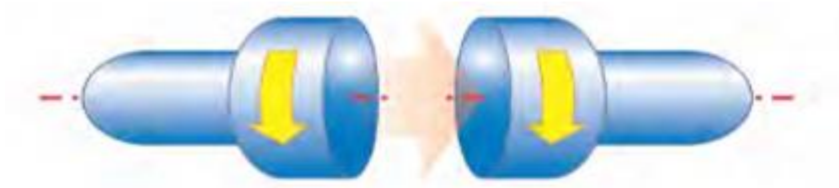


Figure 160.shaft alignment

Don’t confuse shaft alignment with coupling alignment. the coupling surface should not be usedto define alignment conditions since they don’t represent rotation axis of shaft.

The alignment condition can change when the machine is running. This can be for a number reasons including thermal growth. piping strain. machine torque. foundation movement and bearing play. Since shaft alignment is usually measurement with machines cold. The alignment condition as measurement is not necessarily the zero alignment condition of the machines.

Alignment condition should be measured while turning the shafts in the normal direction of rotation. Most pumps. fans motors etc. have arrows on the end easing showing direction of rotation.

Machinery catenary

The amount or shaft deflection a machine depends upon several factors such as the stiffness of the shaft, amount of weight between over-hanging supports. the bearing design and the distance between the supports.

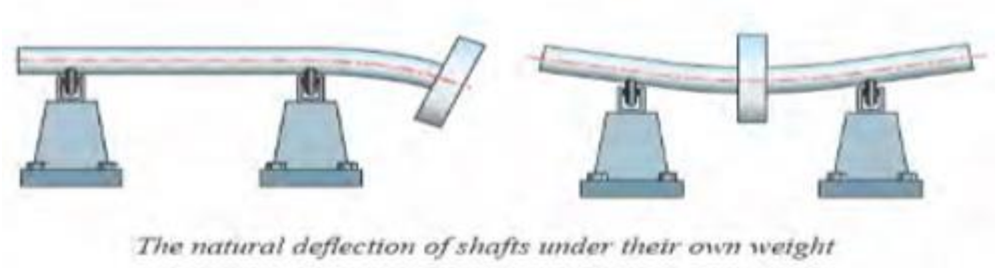


Figure 161.Machinery catenary

Alignment parameter

Since shaft needed to be measured and subsequent corrected, a method of quantifying and alignment condition is necessary

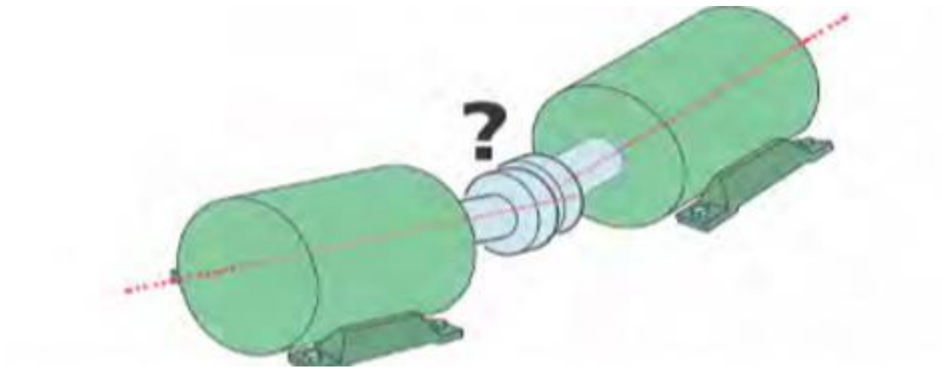


Figure 162.Alignment parameter

Traditionally alignment has been described in terms of dial indicator readings at the coupling face or position values at the machine feet, the measured values from both of these methods are dependent upon the dimensions of the machines. Since there are many different methods for mounting dial indicators (reverse indicator. rim and face. double rim for example) the comparison of measurements and application of tolerances can be problematic. Additionally, the fact that rim indicator readings show twice the true offset and sign reversals must be observed depending on whether the indicator measures an internal or external. left or right coupling face or rim.

A more modern and easily understandable approach is to describe machine alignment condition in terms of angularity and offset in the horizontal (plan view) and vertical (side view), Using this method four values can then be used to express alignment condition as shown ill the following diagram.

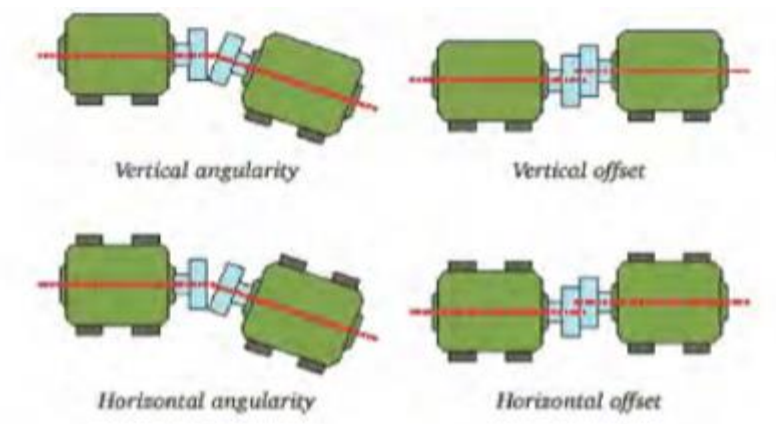


Figure 163

Angularity,gap and offset angularity describe the angle betueen two rotating axes

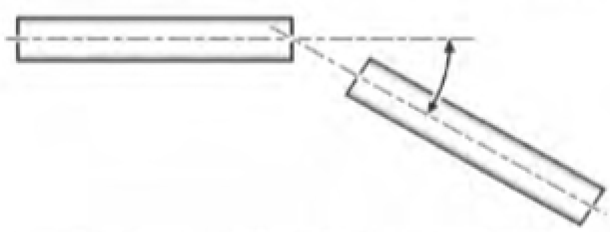


Figure 164

Angularity can be expressed directly as an angle in degrees or in terms of slope in mils/inch. This latter method is since the angularity multiplied by the coupling diameter gives an equivalent gap difference at the coupling rim.
Thus the angle is more popularly expressed in terms of “Gap per diameter. The gap itself is not meaningful, it must be divided by the diameter to have meaning. The diameter is correctly referred to as a "working diameter". but is often called a coupling diameter. The working diameter can be any convenient value. It is the relationship between gap and diameter that is Important.
Thus the angle is mote popularly expressed in terms of GAP per diameter. The gap itself is 1101 meaningful. it must be divided by the diameter to have meaning, The diameter is correctly referred to as n "working diameter", but is often called coupling diameter, The working dimeter can be any convenient value, It is the relationship between gap and diameter that is Important.

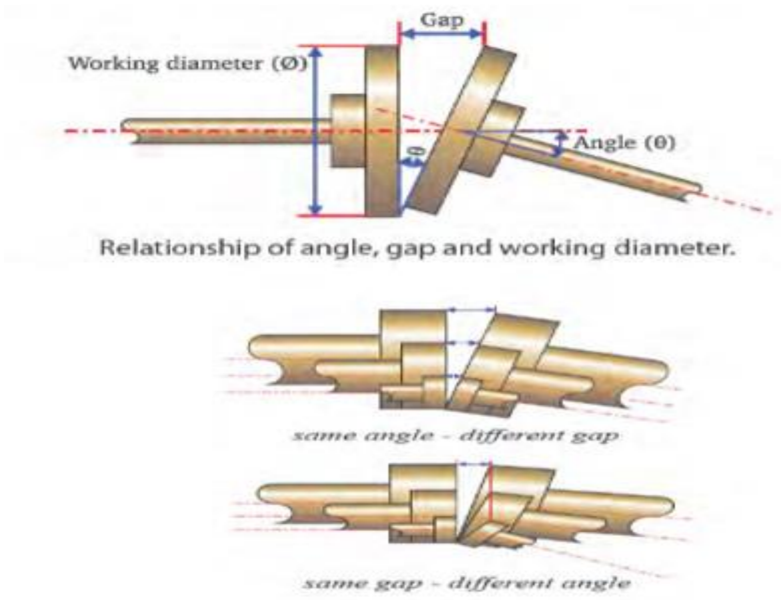


Figure 165

Offset describes the distance between rotation axes at a given point. Offset is sometimes incorrectly referred or as parallel offset or rim misalignment, the shaft rotation axes ate however rarely parallel and the coupling or Shan rim has an unknown relationship to the shaft rotation axes

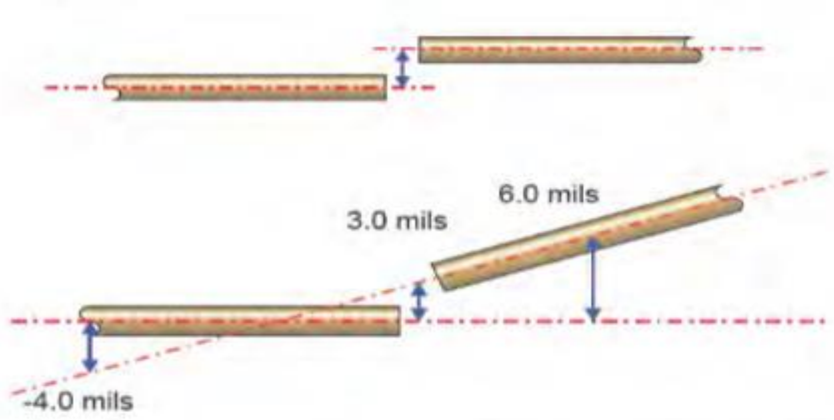


Figure 166

As shown above. for the same alignment condition. the offset value varies depending upon the location where the distance between two shaft rotation axes is measured. In the absence of any other instruction. offset is measured in mm or thousandths of an inch at the coupling center. (This definition refers to short flexible couplings, for spacer coupling offset should be measured oat the power transmission planes of the coupling).

1.3. Temperature analysis:

as expected, spectrum analysis alone did not give any discriminative feature for diagnosis. However, combining temperature with vibration data may produce results that are useful for diagnosis. Therefore, two combined vibration and temperature analysis approaches have been attempted in this section.

Computation of steady state temperatures

Assuming the majority of heat loss from the bearing was via conduction to the steel pedestals, which could have acted as a large heat sink, Figure represents a heat transfer model of the experimental rig bearings. This model became the basis of deriving the simple equation of the thermodynamics of the bearing which was used to approximate steady state temperatures attained The derivation is given hereafter

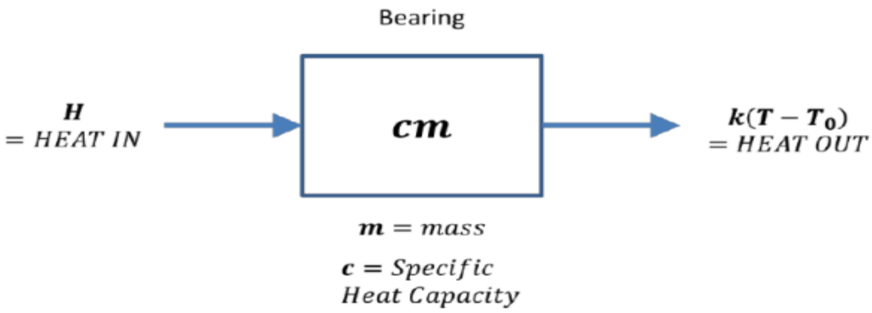


Figure 11 Bearing heat transfer model

the fundamental equation of heat transfer, where ambient temperature = To, Bearing mass= m, Specific Heat Capacity= c, heat input = H and bearing heat loss coefficient = k

$$cm\frac{dT}{dt} = H - k(T - T_0)$$

At steady state conditions, t -& bearing temperatures tend to maximum value =T_{max} , therefore;

$$cm\frac{dT}{dt} = k(T_{max} - T)$$

$$cm\int_{T_0}^{T}\frac{dT}{T_{max} - T} = k\int_{0}^{t} dt$$

$$[-cm\ln[T_{max} - T]]_{T_o}^{T} = [kt]_0^t$$

$$\ln[T_{max} - T] - \ln[T_{max} - T_0] = \frac{-k}{cm}t$$

$$\frac{T_{max} - T}{T_{max} - T_0} = e^{\frac{-k}{cm}t}$$

$$T = T_{max} - (T_{max} - T_0)e^{\frac{-k}{cm}t}$$

$$H = k(T_{max} - T_0)$$

If $\frac{k}{cm} = A$;

$$\Delta T = (T_{max} - T_o)e^{-At}$$

where ΔT is temperature increase, T_{max} is steady state temperature, T_O is ambient temperature, A is an arbitrary variable and t is time.

The Temperature analysis focus on experimental set up and data collection by follow that figure below

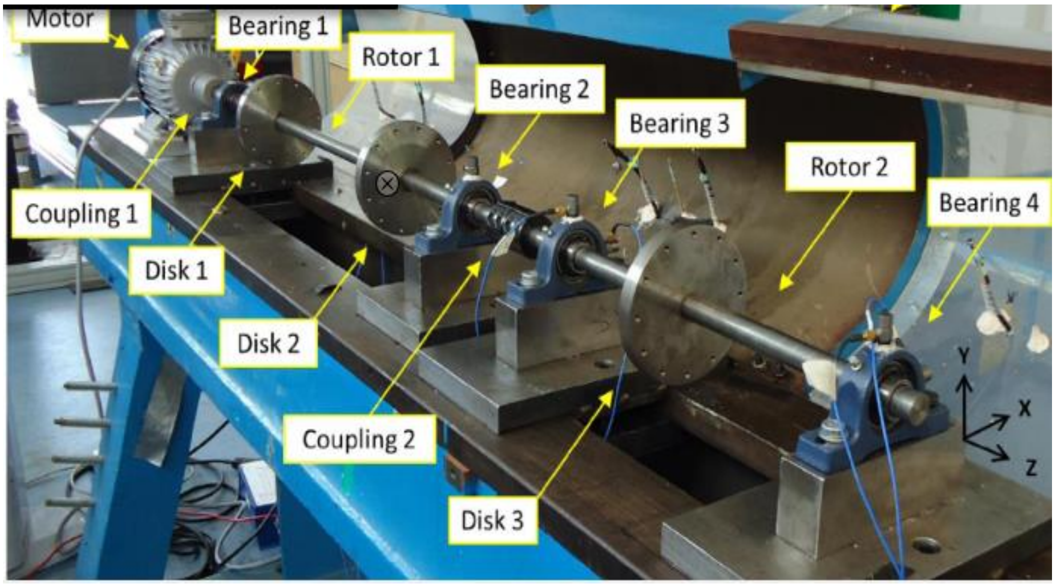


Figure 167.Experimental rig mechanical layout.

1.4. Vibration analysis

Vibration analysis is defined as a process for measuring the vibration levels and frequencies of machinery and then using that information to analyze how healthy the machines and their components are. While the inner-workings and formulas used to calculate various forms of vibration can get complicated, it all starts with using an accelerometer to measure vibration. Anytime a piece of machinery is running, it is making vibrations. An accelerometer attached to the machine generates a voltage signal that corresponds to the amount of vibration and the frequency of vibration the machine is producing, usually how many times per second or minute the vibration occurs.

Faults identified through vibration analysis techniques

Vibration analysis technique is capable of identifying almost all the faults that a machine can have. As a result, occasionally analysis need complementary methods to confirm a diagnosis.

The following are the most common faults that vibration analysis identifies:

1. Imbalance
2. Bearing failures
3. Mechanical looseness
4. Misalignment
5. Resonance and natural frequencies
6. Electrical faults in motors
7. Bent shaft
8. Gearbox failures
9. Cavitation in pumps

10. Critical speeds

Vibration Measurement parameters

Vibration Analysis techniques identify 3 major measurement parameters. Each one of these parameters gives particular importance to certain ranges of frequencies. **Acceleration** gives higher importance to high frequencies. It is useful to see bearing condition. **Velocity** gives equal importance to high and low frequency. It is related to the destructive force of the vibration and therefore the most important unit available. **Displacement** prefers low frequencies. It is useful for during dynamic balancing, orbits and ODS (Operating Deflection Shapes).

2. Description of work

2.1. Oil changing

oil changes will be necessary only a few times during the life of the engine. Until then, timely oil changes, maintaining the engine and changing the air filter will help prolong engine life.

Changing the oil and filter sounds easy. However, several tricks can simplify the task. Performing the job in a set sequence with the proper tools will not only make it easier, it will also reduce the likelihood of making a mess, or worse, a costly mistake

Here are two methods of performing an oil change. The first is to normalize the engine operating temperature. That requires the engine to idle for a time or a few trips around the neighborhood. Now that the engine is warm, drain the sump and change the filter. The thought behind this is contaminates are suspended in the oil and when the sump is drained, the harmful contaminates drain with the oil.

The second method is to drain the oil when the engine is cool, when a good percentage of the oil has settled into the sump. For a couple of years, I practiced the first method. Both have merit; however, the second method works best for me.

With either method, the most important step is to develop a routine that works. The location where the procedure is performed should be reasonably level as most sump bottoms are formed to tilt toward the drain plug.

Draining the oil when the engine is cool (method number two) offers the following benefits: It is not necessary to circle the block a few times or have the engine at high idle for 20 minutes trying to warm the oil; when the engine has been at rest for several hours and the drain plug is removed, the oil is cool and drain flow is more controllable; pressure between the filter and the filter housing has bled off, making filter removal a cleaner process. A cleaner process is generally a quicker process.

Oil Change Procedure

1. Gather the necessary supplies. Wearing disposable gloves, use a paper towel to clean the area around the oil fill cap.
2. Open the cap to break any minor vacuum created when the drain plug is removed. Place the drain pan slightly off-center of the drain plug.
3. Using a suitable wrench (preferably six- point), unscrew the drain plug to the end of the threads. Slowly tip the plug upward and away to get a feel of where to place the drain pan to catch the oil. Be sure to place the pan so when draining slows, the pan is catching the oil.

4. While the sump is draining, open a container of new oil and pour oil into the center of the new filter. Pressure from the pump pushes the oil through the media (from outside to inside), out the center to the oil galleries.
5. Slowly roll the filter around; this allows the filter media to absorb the oil and minimize oil starvation at the bearings upon initial start-up.
6. While the filter media is wicking up the oil and the sump is draining, remove the old filter and turn it over to drain. In many cases, the friction provided by the disposable gloves will allow a sufficient grip to unscrew the old filter without using a filter wrench. A filter wrench will be required if the filter is inaccessible by hand, if the filter was over-tightened or the engine was overheated

2.2. Grease changing

Greases : stand out in particular for their excellent adhesion to the surfaces to be lubricated; also, they are insoluble in water, they resist to shearing and last longer. Generally speaking, grease cannot be heated above 300°C (temperature at which the base oil separates from the thickener). Beyond this temperature, copper or aluminum-based thermal pastes or coatings are more suitable.In addition to its lubricating role (reduction of mechanical fatigue and energy losses due to friction), grease creates a waterproof barrier against external elements (dust, water, solvents, heat, etc.)

Greases consist of:

- ✓ 0 to 95 % base oil (mineral, synthetic or vegetable)
- ✓ 0 to 10 % additive as previously mentioned
- ✓ 3 to 20 % thickening or gelling agent which increases the viscosity of the lubricant (semi-fluid, fluid, soft or hard) and to trap base oil and additives and to avoid leaking.

Grease Change-Over Procedure

(i) Verify that the bearing arrangement allows excess lubricant to be bled from the system. Bearing damage may result in sealed-for-life systems or systems with oil tight sealing arrangements.

(ii) Verify that the new lubricant and the previous lubricant are fully compatible. Mixing two incompatible products may result in chemical or physical changes which will lead to improper lubrication. Contact the lubricant manufacturers to verify compatibility.

(iii) Verify that the subject bearing is operating properly prior to switching products. Improper fits, clearances, bearing configurations or existing bearing damage cannot be corrected by changes in lubrication.

(iv) Verify that the bearing operating condition can accept a 100% fill condition. This procedure should not be applied to bearings which are designed to operate with limited grease quantities. Excessive bearing operating temperature may result in these cases

2.3. Bearing changing

the bearings are used in many a type of vehicles or stationery moving heavy equipment and machinery etc. like diesel engines, earth movers, material handling equipment, compressors and much more. These are used to hold the tires of these machines and are therefore, pretty important irrespective of their size

It is though not very easy to replace these bearings but it is also not that difficult a task that it cannot be done without expert help. It can very well be managed on a person's own as we know that flat tires can be changed without mechanics as well.

Step by step process of changing the bearings from old to new is listed below.

Step 1: The vehicle or machine or equipment should be standing stable on the ground. If it is a car, it should not be in movable condition. Any movement during the change of bearing could lead to some damage the car. The car, therefore should be parked and in neutral gear with the power brakes on.

Step 2: We need to elevate the tire whose bearings need to be replaced. So, the other tires should be blocked using chocks first. Now, in order to elevate the tire, we need to put the jack and pull the tire up as we do to replace a flat tire. With the jack, in place and the chocks preventing the other tires from any kind of slippage, we are now ready to replace the bearings. Do not forget to loosen the lug nuts before you lift the tires.

Step 3: Now unscrew the lug nuts completely and remove the wheel. After the wheel is removed, slowly remove the brake caliper, the dust flap or cover if any, the cotter pin and the castle nut. **Ensure to keep these parts safe with you as you would need to replace them after you change the bearings.**

Step 4: It is the turn to remove the rotor now. It should be done carefully as it can be damaged. Use your palm to get it uplifted from its place and then remove it cautiously.

Step 5: Unscrew the hub bolts and the old hub: We need to remove the old hub which holds the bearings inside it. Since the bearings are clutched quite inside these hubs, you may use a wrench to clear all holds and then finally take off the bearings.

Step 6: Now is the time to replace the old bearings with the new set. Always ensure to use a lot of grease on the bearings. You can do that either using your hand or through a bearing packer for applying grease which comes as a ready tool.

Step 7: Now is the time to go reverse the order and replace all parts again.

2.4. Gauging measurements

There are many instruments, tools and gauges which are used mechanical.

They are usually of two types- inside and outside calliper. They are used to measure internal and external size (e.g. diameter) of an object. It requires an external scale to compare the measured value. This tool is used on those surface where a straight ruler scale cannot be used. After measuring the body/ part, the opening of the calliper mouth is kept against the ruler to measure the length or diameter.

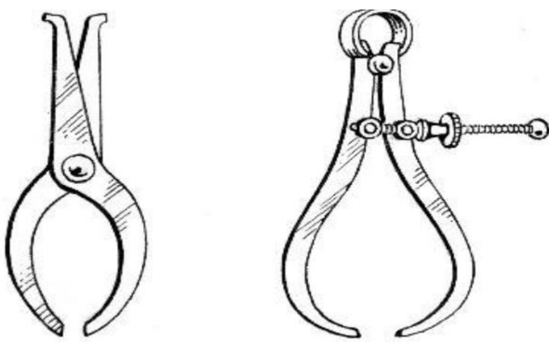


Figure 168.Calipers

Vernier Caliper

It is counted in the list of quality measuring instruments, which are used to measure small parameters with high accuracy. It has got two different jaws to measure outside and inside dimensions of an object. It can be a scale, dial or digital type Vernier calliper. Vernier calliper is one of the most used mechanical measuring tools

Micrometer

It is an excellent precision tool which is used to measure small parameters and is much more accurate than the verniercalliper. The micrometre size can vary from small to large. The large micrometrecalliper is used to measure large outside diameter or distance. E.g. Large micrometre is used as a special mechanical measuring tool for main engine to record the outer diameter of the piston rod.

They are available in two types- Inside micrometre

(to measure inside diameter) and Outside micrometre (for measuring outside diameter).

The Least count of the micrometre is 0.01 mm or 0.001cm.

Figure 169,micrometre

Feeler gauge

Feelers gauges are a bunch of fine thickened steel strips of different thickness bundled together. The thickness of each strip is marked on the surface of the strip. The feeler gauge is used to measure the clearance or gap width between surface and bearings.

E.g. The feeler gauge is widely used to measure piston ring clearance, engine bearing cleaner, tappet clearance etc.

Figure 170.Feeler gauge

Telescopic Feeler Gauge

Similar to the functionality of feeler gauge, this type of gauge is also known as tongue gauge, and it consists of long feeler gauge inside a cover with tongue or curved edge.
The long feeler strips protrude out of the cover like a telescope so that it can be inserted into remote places where feeler gauge access is not possible. E.g. It is used to measure the bearing clearance of the top shell.It is essential that after the use of the telescope gauge, the strip should be cleaned and retracted back to its housing, else it may damage the feeler strip.

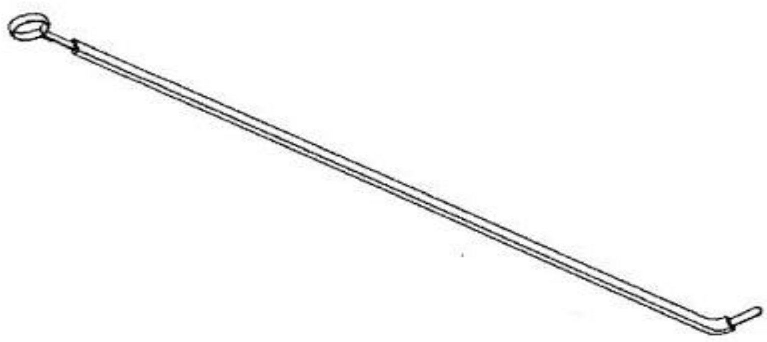


Figure 171.Telescopic Feeler Gauge

Bridge Gauge
As the name suggests, Bridge gauge looks like bridge carrying the measuring instrument at the centre of the bridge. They are used to measure the amount of wear of Main engine bearing. Typically the upper bearing keep is removed, and clearance is measured for the journal. A feeler gauge or depth gauge can be used to complete the process.

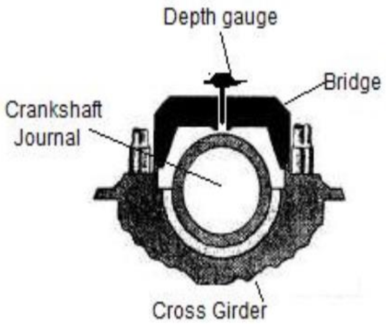


Figure 172.Bridge Gauge

Wire Gauge
American wire gauge or AWG is a standard tool which is circular and has various slots of different diameter in its circumference. It is used to measure the cross section of an electric cable or wire. This tool is usually kept in the electrical workshop of the ship, and electrical officer uses it for measuring wire thickness.

Figure 173.Wire Gauge

Dial Gauge

The dial gauge is utilised in different tools as stated above and can be separately used to measure the trueness of the circular object, jumping off an object, etc. It consists of an indicator with the dial, which is connected to the plunger carrying the contact point. Once the contact point is kept in touch with an object (to be measured), any unevenness or jumping will cause the plunger to move.

Figure 174.Dial Gauge

3. **Frequency of work:**
    - ✓ Every day
    - ✓ 2xevery week
    - ✓ Once every week
    - ✓ Once every three month

L O 5.2 -Prepare report of corrective maintenance

- **Content/Topic1: Corrective maintenance report format**
    - ✓ **Machine part name**

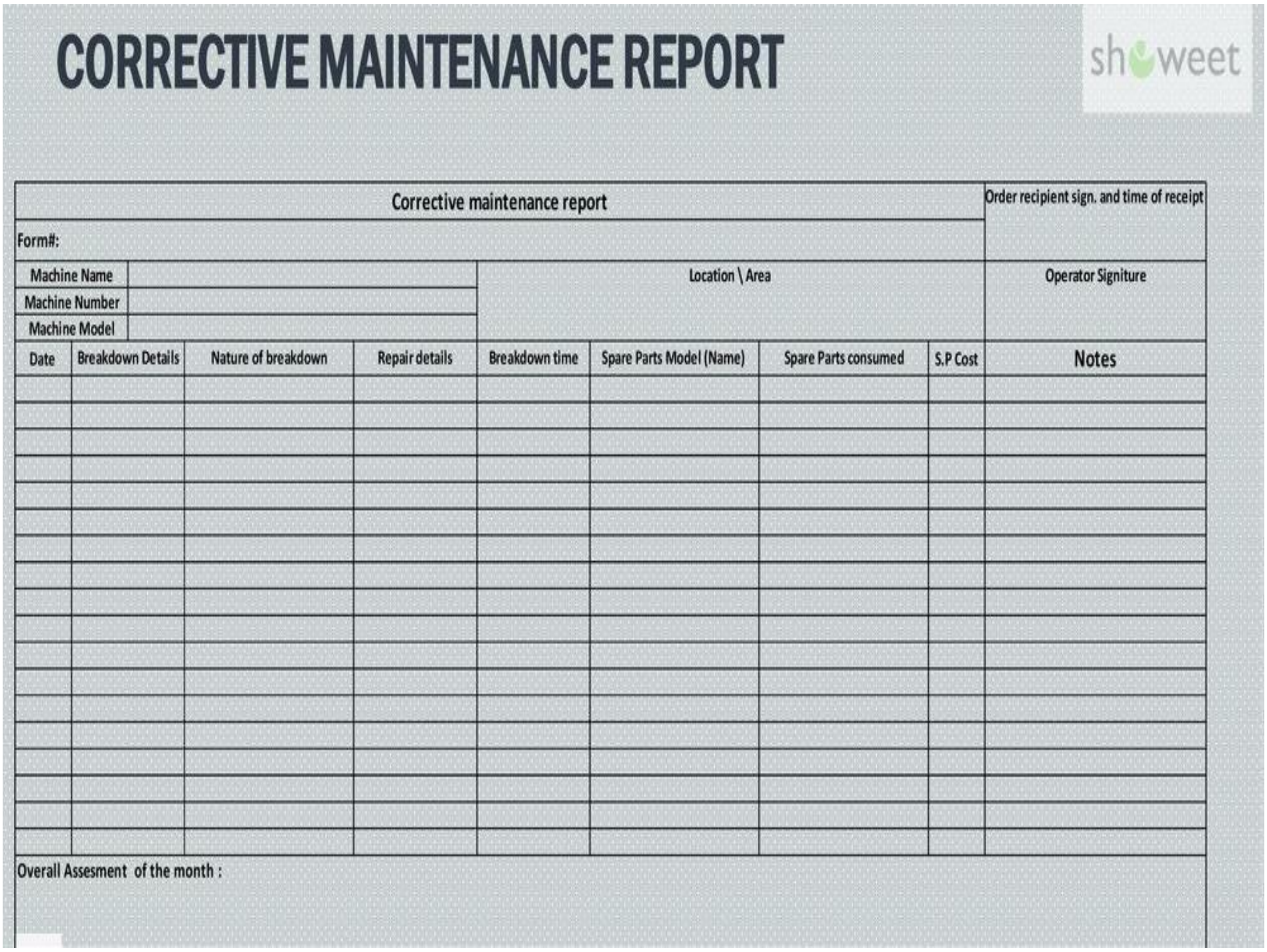
Corrective (Reactive) Maintenance

Corrective maintenance (CM) involves the replacement or repair of equipment after it fails. In response to equipment failure, CM tasks identify the failure (it may be an equipment component or equipment item) and rectify the failure so that the equipment can be reinstated and the facility production restored. CM tasks are prioritized so that the high-priority tasks that may be safety related or affecting production are addressed first.

Corrective Tasks

Corrective maintenance consists of the action(s) taken to restore a failed component to operational status. Corrective maintenance is performed at unpredictable intervals because a component's failure time is not known a priori.

#	Task Description	Condition	Type	Interval	Crew	Duration
1	Repairing the leaking crack		Corrective		Fatigue or corrosion of the body shell or piping_Crew	1.000000 day



CORRECTIVE MAINTENANCE REPORT

Corrective maintenance report								Order recipient sign. and time of receipt
Form#:								
Machine Name				Location \ Area				Operator Signiture
Machine Number								
Machine Model								
Date	Breakdown Details	Nature of breakdown	Repair details	Breakdown time	Spare Parts Model (Name)	Spare Parts consumed	S.P Cost	Notes
Overall Assesment of the month :								

Corrective maintenance report

Instrument	Raman lidar
Location	Central facility
Date/time (gmt)	12/12 2020
Technician	martin
Main types	Unschudule cm
Problem cause	HW failure
Problem fixed	no
component	laser
Problem description	The raman lidar laser energy hand fallen to around 200 mj The lamp were due to be replace but the energy seemed to drop off rather quickl the past couple or day
Action performed	Inspection of the optics during lamp replacement found that the pockells cekk was burned ,the pockells cell had just been installed by continuum on 10/9 and should be under warranty. Scheduled a continuum service call for 12 /18.system off line until then.

- ✓ **Part number**

ANOMALIAS MUY CRITICAS	N.Total Peticiones	N.Petic. Iniciadas	N.Petic. Finalizadas	T.Total Intervenc.	T.Medio Respuesta	Resp. > R2	N.Interv. 10.000 I.c.
Semana 9 (02-03-1998 - 08-03-1998)							
Semana 10 (09-03-1998 - 15-03-1998)							
Semana 11 (16-03-1998 - 22-03-1998)							
Semana 12 (23-03-1998 - 29-03-1998)							
Total del Mes							

ANOMALIAS CRITICAS	N.Total Peticiones	N.Petic. Iniciadas	N.Petic. Finalizadas	T.Total Intervenc.	T.Medio Respuesta	Resp. > R2	N.Interv. 10.000 I.c.
Semana 9 (02-03-1998 - 08-03-1998)	1	1	1	2:00	2:00		0,00708
Semana 10 (09-03-1998 - 15-03-1998)							
Semana 11 (16-03-1998 - 22-03-1998)	1	1	1	2:00	2:00		0,00708
Semana 12 (23-03-1998 - 29-03-1998)	1	1	1	4:00	4:00		0,00708
Total del Mes	3	3	3	8:00	2:40		0,02124

ANOMALIAS NO CRITICAS	N.Total Peticiones	N.Petic. Iniciadas	N.Petic. Finalizadas	T.Total Intervenc.	T.Medio Respuesta	Resp. > R2	N.Interv. 10.000 I.c.
Semana 9 (02-03-1998 - 08-03-1998)	1	1	1	2:00	2:00		0,00708
Semana 10 (09-03-1998 - 15-03-1998)	1	1	1	3:00	3:00		0,00708
Semana 11 (16-03-1998 - 22-03-1998)							
Semana 12 (23-03-1998 - 29-03-1998)							
Total del Mes	2	2	2	5:00	2:30		0,01416

ANOMALIAS ON CALL	N.Total Peticiones	N.Petic. Iniciadas	N.Petic. Finalizadas	T.Total Intervenc.	T.Medio Respuesta	Resp. > R2	N.Interv. 10.000 I.c.
Semana 9 (02-03-1998 - 08-03-1998)	4	4	4	8:00	2:00		0,02832
Semana 10 (09-03-1998 - 15-03-1998)							
Semana 11 (16-03-1998 - 22-03-1998)	11	11	11	30:00	2:44		0,07788
Semana 12 (23-03-1998 - 29-03-1998)	3	3	3	2:30	0:50		0,02124
Total del Mes	18	18	18	40:30	2:15		0,12745

- ✓ Check for:
- ➢ Good condition
- ➢ Alignment
- ➢ Smooth transmission
- ➢ Cleanliness
- ➢ Flatness
- ✓ Maintain method
- ✓ Remark
- ✓ Maintenance level
- ✓ Cause of failure
- ✓ Task types:
    - ➢ Preparation
    - ➢ Fault isolation

- ➢ Disassembly
- ➢ Interchange
- ➢ Reassembly
- ➢ Alignment
- ➢ checkout
- ➢ Start up

✓ Task description

➢ **Remove pulley**

Disclosed is an improved puller especially adapted for removing pulleys from the shafts of engines such as lawn mower engines, small tractor engines, etc. The puller comprises generally a box-like member of heavy steel plates with one open wall. A jack screw is threadedly associated with one wall and a pulley, mounted on the shaft, is adapted to be received in position for operative thrust engagement with an opposite wall. Running in on the jack screw forces the hub or central portion of the pulley into operative thrust contact with the opposite wall, whereby the shaft is moved axially of the pulley, thus effectively removing the pulley without damage.

➢ **Replace bearing**

Ordering the correct replacement bearing is a critical task – but one that is not difficult if you take time to gather the right information.

Just follow these steps:

IDENTIFY – the type of bearing you need to replace.

- ✓ Ball Bearing –Single Row, Double Row, Angular Contact
- ✓ Roller Bearing – Cylindrical, Spherical, Tapered
- ✓ Thrust Bearing – Ball or Roller
- ✓ Split Pillow Block – Pillow Blocks ›
- ✓ Super Precision – Angular Contact Ball, Cylindrical Roller, Ball Screw Support

LOCATE – the identification number on the bearing. Bearing identification numbers are usually located on the inner ring face, outer ring face or bearing O.D. Mounted units are identified by a number tag fastened to the unit or by a housing number cast into the housing cap.

MEASURE – if you need to. › If a bearing identification number is not legible, you will need to determine the following:

1. Inner ring bore (inside diameter)
2. Outer ring outside diameter
3. Inner width and outer width (these may be different)

Shape of the bore and/or outside diameter of bearing – spherical, tapered or cylindrical

RECORD – additional relevant information. › The more information available, the easier it will be to identify the replacement bearing needed. Record:

1. Unique features such as lubrication holes, snap ring grooves, machined shoulders, etc.
2. Application/equipment data(Zane Winberg, 2006)
(INC, 2002)

➢ **Task code:**

- PRE

- FISO
- DIS
- INT
- REAS
- ALG
- CHK
- STRT

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