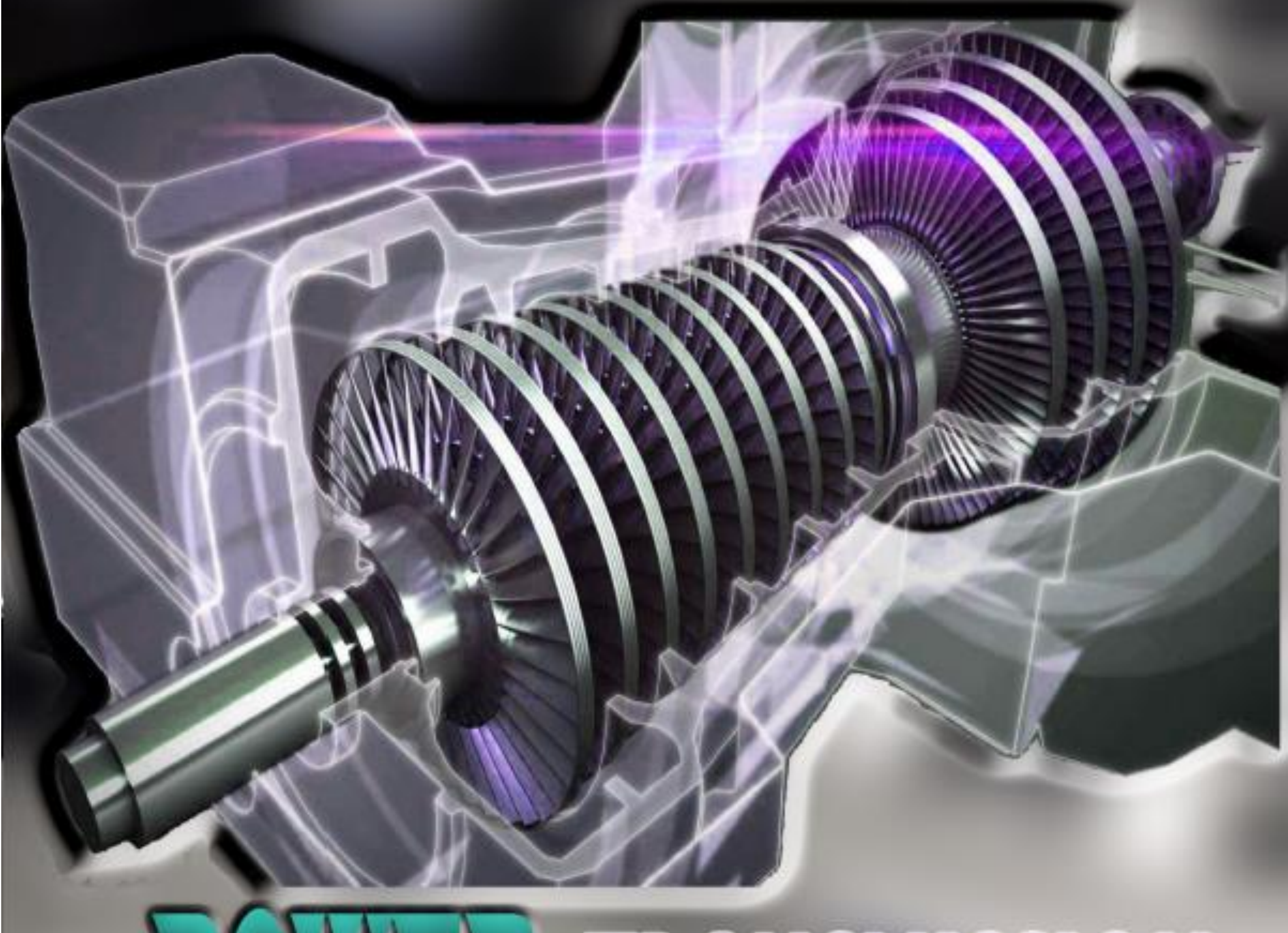
FIRST EDITION



POWER TRANSMISSION MECHANISM

MALAYSIAN POLYTECHNIC

POWER

TRANSMISSION MECHANISM

PREPARED BY

Rosliza Binti Hasan

Mohd Nazrulazlan bin Abd Rasid

Mohamad Tarmizi Bin Arbain

Penerbit

Politeknik Nilai

2024

AUTHORS

ROSLIZA BINTI HASAN

MOHD NAZRULAZLAN BIN ABD RASID

MOHAMAD TARMIZI BIN ARBAIN

EDITOR

ROSLIZA BINTI HASAN

First edition: 2024

Perpustakaan Negara Malaysia Cataloguing-in-Publication Data
Power Transmission Mechanism / Rosliza binti Hasan

e ISBN 978-967-2742-33-3

Editor:

Rosliza binti Hasan

Authors:

Rosliza bin Hasan

Mohd Nazrulazlan bin Abd Rasid

Mohamad Tarmizi bin Arbain

Cover Design and Interior Layout:
Rosliza binti Hasan

Published by:

POLITEKNIK NILAI

KEMENTERIAN PENDIDIKAN TINGGI MALAYSIA

Kompleks Pendidikan Bandar Enstek
71760 Bandar Enstek
Negeri Sembilan

PREFACE

The 1st edition of the Power Transmission Mechanism specifically based on Malaysian Polytechnic syllabus for DJM 6122 Power Transmission Mechanism. The scope is intended to give adequate coverage to mechanical engineering students of Malaysian Polytechnics. Every chapter is written by three authors among polytechnic lecturers. Power Transmission Mechanism is basically to store or generate power, but all are useless without a transmission to transmit the power from a source to the place where it is needed in a form in which it can be used. There are amazing arrays of transmission components, and engineers need to become familiar with what is available, and what niche each element occupies.

Accordingly, this chapter will focus on some of the elements of transmission systems, such as chain, bearing, gear, cam, followers, coupling, clutch and brake. This book gives knowledge on the working principle of elements power transmission mechanism and students should be able to choose and form power transmission mechanism.

In this textbook, we emphasize on the elements of theoretical, the techniques, an example and tutorial. All the theories are given in a simple explanation so that student should be able to get the concepts of the chapter easily. The techniques describe the method in a sequence of steps that student can follow systematically. The examples are presented in full details of explanation. In order to make this volume more useful for them, most of these examples are taken from the recent examination questions, to make the students familiar with the type of questions, usually, set in their examinations.

At the end of each chapter, a few exercises have been added for the students to solve them independently. However it is too much to hope that these are entirely free from errors. In short, it is earnestly hoped that the book will earn appreciation of all the teachers and students alike. Although every care has been taken to check mistakes and misprints, yet it is difficult to claim perfection. Any errors, omissions and suggestions for the improvement of this treatise, brought to our notice, will be thankfully acknowledged and incorporated in the next edition.

ACKNOWLEDGEMENT

In the Name of Allah, Most Gracious, Most Merciful, all praise and thanks are due to Allah, and peace and blessings be upon His Messenger Mohammad. The first edition of Mechanics of Machines textbook is finally released. We would like to express the most sincere appreciation to all parties who made these work possible, advisory members, friends, family, directly or indirectly in the completion of this book.

Primarily thanks addressed to thanks to 'Hamba Allah', who play a very important role in the growth and development from the beginning until end of publication. We want to express our sincere gratitude for all the great team members, giving full support, encouragement and a priceless cooperation, in preparing this book. Your help indeed opened our eyes to many opportunities we never thought of.

We also appreciate any input from our students in contributions toward improving and strengthening the content of this book. A huge appreciation and acknowledge are extended to our family for their support and true love. We are grateful for the countless sacrifices they made. The book would not be possible to publish without all of their support.

Finally, we would like to express our deep appreciation and indebtedness to everyone who are involved directly or indirectly for their endless support in the writing of this book.

Writers' Team

Chapter One and Three: Mohd Nazrulazlan Bin Abd Rasid, (PNS)
Chapter Two, Four, Five, Six and Eight: Rosliza Binti Hasan, (PNS)
Chapter Seven: Mohamad Tarmizi Bin Arbain, (PPD)
Lead Editor: Rosliza Binti Hasan, (PNS)

ABSTRACT

The 1st edition of the Power Transmission Mechanism specifically based on Malaysian Polytechnic syllabus for DJM6122 Power Transmission Mechanism. The scope is intended to give adequate coverage to mechanical engineering students of Malaysian Polytechnics. This book will focus on some of the elements of transmission system such as chain, bearing, gear, cam, followers, coupling, clutch and brake. This book gives knowledge on the working principle of elements power transmission mechanism and students should be able to choose and form power transmission mechanism. This book emphasizes on the elements of theoretical, the techniques, an example and tutorial. All the theories are given in a simple explanation so that student should be able to get the concepts of the chapter easily. The techniques describe the method in a sequence of steps that student can follow systematically.

TABLE OF CONTENTS

LIST OF FIGURES AND TABLES

CHAPTER 1: COUPLING

OBJECTIVES:

In the end of this chapter, student should know to:

- Explain the concept of coupling.
- Describe the measurement methods for coupling

1.1 INTRODUCTION

This topic introduces coupling and its application, and the measurements involved.

1.2 DEFINITION

- A coupling is a device used to connect two shafts together at their ends for the purpose of transmitting power.
- Couplings do not normally allow disconnection of shafts during operation, however there are torque limiting couplings which can slip or disconnect when some torque limit is exceeded.
- The primary purpose of couplings is to join two pieces of rotating equipment while permitting some degree of misalignment or end movement or both.

1.3 FUNCTION

- To provide for the connection of shafts of units that is manufactured separately such as a motor and generator and to provide for disconnection for repairs or alternations.
- 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 distribute power supply to the system without any reduction during the speed.

1.4 TYPES OF COUPLING

Coupling can be divided into TWO (2) main groups:

1.4.1 RIGID COUPLING

- A rigid coupling is a unit of hardware used to join two shafts within a motor or mechanical system
- It is used to connect two separate systems, such as a motor and a generator, or to repair a connection within a single system.
- It may also be added between shafts to reduce shock and wear at the point where the shafts meet.
- Rigid couplings are used when precise shaft alignment is required; shaft misalignment will affect the coupling's performance as well as its life.

1.4.2 FLEXIBLE COUPLING

- Flexible couplings are designed to transmit torque while permitting some radial and axial and angular misalignment.
- Flexible couplings can accommodate misalignment up to a few degrees in some parallel misalignment.
- Flexible couplings can also be used for vibration damping or noise reduction.

1.4.3 EXAMPLES OF COUPLING

Rigid Coupling	Flexible Coupling
1. Compression or Clamp 2. Solid/Sleeve Muff Coupling 3. Bellow Coupling 4. Flange Coupling	1. Universal Coupling 2. Disc Coupling 3. Spider/Jaw Coupling 4. Chain Coupling 5. Oldham Coupling

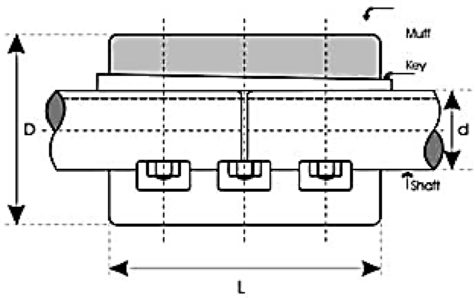
1. **Clamp or Compression Coupling**

It is also known as split muff coupling. The muff or sleeve is made into two halves and bolted together. The halves of the muff are made of cast iron. The shaft ends are made to a butt each other and a single key is fitted directly in the keyways of both the shafts. One-half of the muff is fixed from below and the other half is placed from above. Both the halves are held together by means of mild steel studs or bolts and nuts.

This type of coupling is important in piping system connection. It is used to connect two pipes or tubes. The connection process is simple and strong and it can support the 2 surface from leakage.

The advantage of this coupling is that the position of the shafts do not need to be changed for assembling or disassembling of the coupling. This coupling may be used for haevy duty and moderate speeds. There is difficulty in maintaining dynamic balancing of the coupling. Therefore, it is not possible to use the clamp for high speed application.

2. **Solid or Sleeve Muff Coupling**



This is the simplest type of rigid coupling. It is made from cast iron and very simple to design and manufacture. It consists of a hollow cylinder (muff) whose inner diameter is the same as diameter of the shaft (sleeve). It is fitted over the ends of the two shafts by means of a gib head key. The power is transmitted from one shaft to the other shaft by means of a key and a sleeve.

This type of coupling offers an easy and cost effective method for extending a shaft or adapting between two different shaft sizes. It has high torque capabilities to make it is suitable for higher RPM power transmission applications. No lubrication is required in this coupling.

3. **Bellow Coupling**

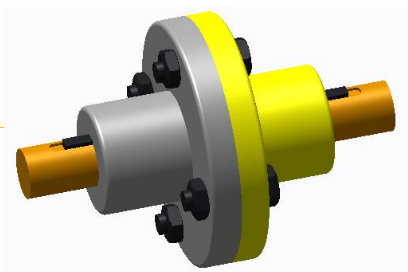
Bellows couplings are typically made from a stainless steel tube hydroformed to create deep corrugations that make them flexible across axial, angular, and parallel shaft misalignments. The bellows absorbs any slight misalignments between the mounting surfaces of two components. When coupling shafts, bellows couplings absorb slight misalignments from perpendicularity and concentricity tolerances between the mounting surfaces of the two connected components.

Bellows couplings tend to have the highest torsional stiffness of any servo motor coupling, and so are used in applications requiring high precision positioning. It is also light in weight and good value for money because of maintenance free.

4. **Flange Coupling**

Flange coupling is a type of Coupling used between rotating shafts that consists of flanges one of which is fixed at the end of each shaft, the two flanges being bolted together with the bolts and nuts fitted circumferentialy to complete the drive. Flange coupling can only be used when both shaft are perfectly aligned in static or dynamic condition. It is used when two shaft are to be coupled at no angular, as well as no change in centre axis line.

This type of coupling mostly preferred coupling in many applications. It is easy and simplest type of coupling and the cost is also low. Flange couplings are typically used in pressurized piping systems where two pipe or tubing ends have to come together. The connecting methods for



flange couplings are usually very strong because of either the pressure of the material or the sometimes hazardous nature of materials passed through many industrial piping systems. High thread count nut and bolt connections are used to secure the flange couplings in place. These nuts and bolts are usually made from tempered steel or alloys to provide enduring strength and the ability to be tightened to the utmost level to ensure the piping system does not leak at any flanged junction. Most flange couplings utilize four, six, or up to 12 bolt assemblies.

5. **Universal Coupling**

Universal coupling also called as universal joint shaft or U-joint is a joint that is used to join two rods which are inclined to each other at their axes. This coupling is mostly used in shafts that used to transmit rotary motion or power. It compensates angular misalignment between the shafts in any direction.

This type of coupling facilitates torque transmission between shafts which have angular misalignment. It is cheap and cost effective also simple to be assemble and dismantle. The torque transmission efficiency is high and permits angular displacements. It is commonly use with U-joint in the drive shaft on rear-wheel or four-wheel drive vehicle.

6. **Disc Coupling**

Disc coupling transmit torque from driving to a driven bolt tangentially on a common bolt circle. Torque is transmitted between the bolts through a series of thin, stainless steel discs assembled in a pack.

This type of coupling is a high performance motion control (servo) coupling designed to be the torque transmitting element (by connecting two shafts together) while accommodating for shaft misalignment. It is designed to be flexible, while remaining torsional strong under high torque loads.

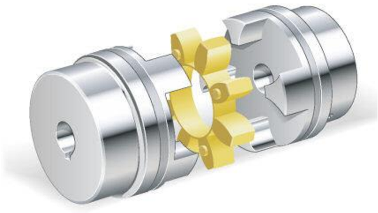
There are two different styles of disc coupling which is single disc and double disc. The difference between the two styles is that single disc coupling cannot accommodate parallel misalignment due to the complex bending that would be required of the lone disc. Double disc styles allow the two discs to bend in opposite directions to better manage parallel offset.

The disc coupling less need for lubrication and maintenance. This coupling can be inspected without disassembly. It was torsionally rigid without any backlash. It has no wearing parts because it was high resistance to harsh environmental condition. Infinite life used if properly aligned.

7. **Spider or Jaw Coupling**

A jaw coupling is a type of general purpose power transmission coupling that also can be used in motion control (servo) applications. It is designed to transmit torque (by connecting two shafts) while damping system vibrations and accommodating misalignment, which protects other components from damage. Jaw couplings are composed of three parts: two metallic hubs and an

elastomer insert called an element, but commonly referred to as a "spider". The elastomer of the spider can be made in different hardnesses, which allows the user to customize the coupling so that it absorbs more or less vibration.



Jaw couplings are well balanced and able to tolerate high RPM. With its damping capability and interchangeable spiders, jaw couplings make a great solution for shock absorption. The drawback of the jaw coupling is the lack of misalignment capability. Too much axial motion will cause the coupling to come apart, while too much angular or parallel misalignment will result in bearing loads that are higher than most other servo/motion control couplings. If the spider fails, the jaws of the two hubs will mate, much like teeth on two gears, and continue to transmit torque.

8. **Chain Coupling**

Chain Couplings transmit torque through two hubs with hardened sprocket teeth and a double width roller chain. It is often used to convey power to the wheels of a vehicle, particularly bicycles and motorcycles. It is also used in a wide variety of machines besides vehicles.

By using this type of coupling, the force can be transferred in long distance. The noise is low for toothed operation. It has high bending reliability for power transfer using sprocket. High heat resistance for chain and easy to maintain.

9. **Oldham Coupling**

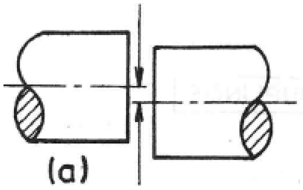
An Oldham coupling has three discs, one coupled to the input, one coupled to the output, and a middle disc that is joined to the first two by tongue and groove**.** The tongue and groove on one side is perpendicular to the tongue and groove on the other. The middle disc rotates around its center at the same speed as the input and output shafts. Its center traces a circular orbit, twice per rotation, around the midpoint between input and output shafts. Often springs are used to reduce backlash of the mechanism.

An advantage to this type of coupling, as compared to two universal joints, is its compact size. It also protects support bearings by exerting consistently low reactive forces, even under large misalignments. The disadvantage of this type of coupling is it accommodates a relatively small angular misalignment. This type of coupling typically used in printing and copying machines, robotics and servo application.

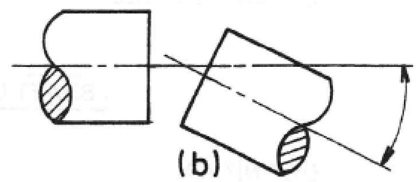
1.5 TYPE OF MISALIGNMENT

There are three basic types of misalignment:

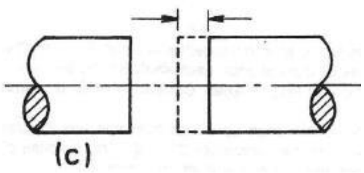
- Parallel



- Angular



- Axial



1.6 MAINTENANCE OF COUPLINGS

There are a few things to do a maintenance on couplings:

- Always do an inspection to make sure there is no loose in connection of couplings
- Always put a grease for longer used of coupling
- Always do maintenance according to the schedule given by the manufacturer
- Make sure the coupling is free from dirt. It need to install a cover for backup.

Even with proper maintenance, couplings can be fail. Underlying reasons for failure, other than maintenance, include:

- Improper installation of the coupling.
- Poor coupling selection used.
- Operation beyond design capabilities.

The only way to improve coupling life is to understand what caused the failure and to correct it prior to installing a new coupling. Some external signs that indicate potential coupling failure include:

- Abnormal noise, such as screeching, squealing or chattering.
- Excessive vibration or wobble.
- Failed seals indicated by lubricant leakage or contamination.

1.7 REQUIREMENT OF GOOD COUPLING SETUP

The requirements of good coupling setup are:

- Transmit the full power from one shaft to other without losses.
- Allow some misalignment between two adjacent (near) shaft rotation axes.
- It is the goal to minimize the remaining misalignment in running operation to maximize power transmission and machine run time.
- Should have no projecting part.

1. Describe the definition and functions of Coupling
2. Explain the types of coupling below:
   a) Disc type
   b) Muff type
3. State **FOUR (4)** advantages of compression coupling
4. Explain how to make a maintenance for coupling

Previous Years' Question Papers

DEC 16

a. Explain the function of coupling in power transmission mechanisms. 3M
b. List 2 advantages and disadvantages of split-muff couplings. 8M

JUNE 16

a. Define the following terms:
   i) Rigid Coupling 2M
   ii) Flexible Coupling 2M

b. Identify 3 disadvantages of coupling in engineering 6M

DEC 15

a. Define:
   i) Coupling 3M
   ii) Rigid Coupling 3M

b. State 3 function of flexible coupling 9M
c. Explain the terms below:
   i) Compression Coupling 5M
   ii) Bush Pin Type Flange Coupling 5M

JUNE 15

a. Define the meaning of coupling system. 2M
b. List TWO (2) group of coupling. 2M
c. From question 1 (b) List THREE (3) type of coupling for each group. 6M
d. Point of THREE (3) advantages and disadvantages split-muff coupling. 12M
e. State TEREE (3) types of misalignment. 3M

DEC 14

a. State the purpose of coupling in mechanical engineering. 5M
b. State 2 characteristics that differentiate between rigid coupling and flexible coupling. 4M
c. List 2 types of rigid coupling and 5 types of flexible coupling. 7M
d. From Q1C, choose any three types of coupling and describe them in detail.

CHAPTER 2: CLUTCHES AND BRAKES

OBJECTIVES:

In the end of this chapter, student should know to:
- Explain the concept of clutch.
- Explain the concept of brake.
- Explain the operation of hydraulic brake.

2.1 INTRODUCTION

This topic introduces clutch and brake and its role in power transfer. It introduces types of clutch and the calculation of power transferred by the clutch. It also covers understanding of brake and the operation of hydraulic brake types.

2.2 CLUTCH

In automobile, a gear box is required to change the speed and torque of the vehicle according to the driving requirement. If we change a gear, when the engine is engaged with gear box or when the gears are in running position then it can cause of wear and tear of gears. To overcome this problem a device is used between gear box and engine, known as clutch. Clutch is the first element of power train.

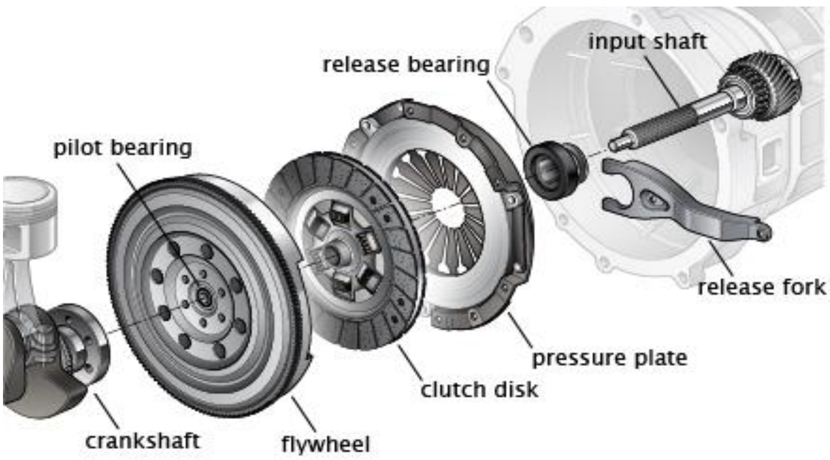


FIGURE 2.1: CLUTCH SYSTEM

2.2.1 DEFINITION

A clutch is a mechanical device which engages and disengages power transmission especially from driving shaft to driven shaft, at the will of the operator or during shifting of gear.

In the simplest application, clutches connect and disconnect two rotating shafts (drive shafts or line shafts). When the clutch is in engage position, the power flows from the engine to the wheel and when it is in disengage position, the power is not transmitted. In these devices, one shaft is typically attached to an engine or other power unit (the driving member) while the other shaft (the driven member) provides output power for work. While typically the motions involved are rotary, linear clutches are also possible.

In a torque-controlled **drill**, for instance, one shaft is driven by a motor and the other drives a drill chuck. The clutch connects the two shafts so they may be locked together and spin at the same speed (engaged), locked together but spinning at different speeds (slipping), or unlocked and spinning at different speeds (disengaged).

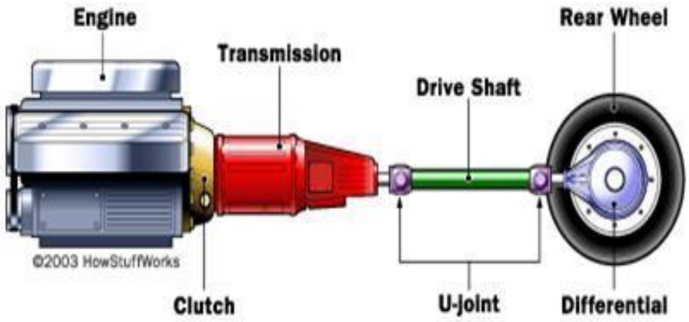


FIGURE 2.2: CLUTCH

2.2.2 PRINCIPLE OF CLUTCH

The very first principle on which a clutch works is **friction**. When two revolving friction surfaces are brought into contact and pressed, then they are united and start revolve at **same speed** due to friction force between them. This is the basic principle of clutch. The friction between these two surfaces depends on the area of surface, pressure applied upon them and the friction material between them. The **driving member** of a clutch is the **flywheel** mounted on the engine crankshaft and the **driven member** is pressure plate mounted on the transmission shaft. Some friction plates, sometimes known as clutch plates are kept between these two members. This whole assembly is known as the clutch.

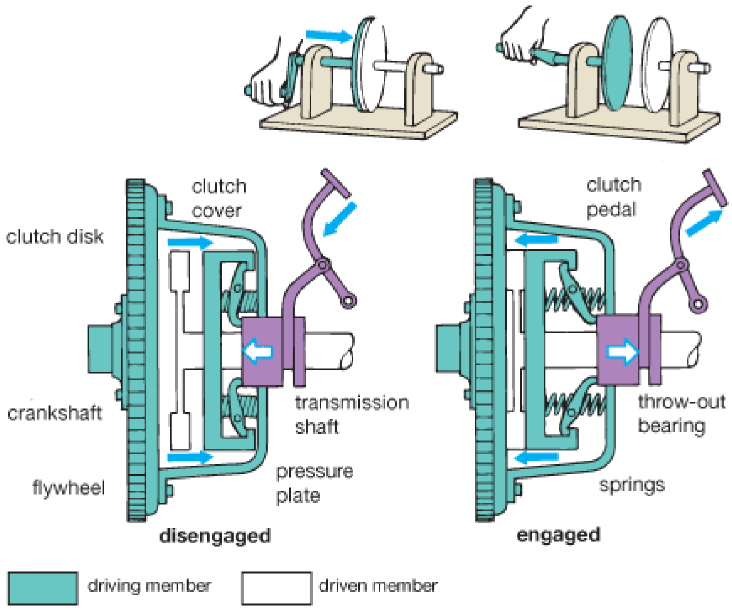


FIGURE 2.3: PRINCIPLE OF OPERATION

2.2.3 AN AUTOMOBILE CLUTCH FUNCTION

An automobile clutch has following function;

i. It can be disengaged. This allows engine cranking and permits the engine to run without delivering power to the transmission.
ii. While disengaging, it permits the driver to shift the transmission into various gear according to operating condition.
iii. While engaging, the clutch slips momentarily. this provides smooth engagement and lessens the shock on gears, shaft and other parts of automobile.
iv. While engaging, the clutch transmits the power to the wheel without slipping, in ideal condition.

2.2.4 Types Of Clutches

There are so many types of clutches. According to the method of transmitting torque, these may classified as follow:

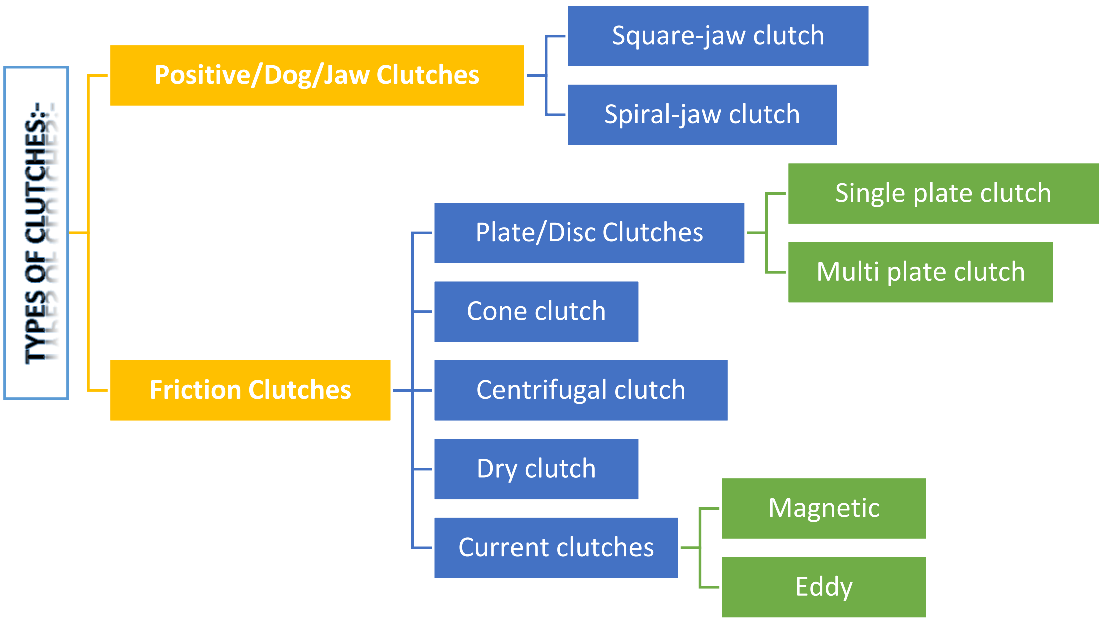


Figure 2.4 Types of Clutches

2.2.4.1 Positive/Dog/Jaw Clutch

In this type of clutch, the engaging clutch surfaces interlock to produce rigid joint they are suitable for situations requiring simple & rapid disconnection, although they must be connected while shaft are stationary & unloaded.

The jaw may be square jaw type or spiral jaw type. They are designed empirically by considering compressive strength of the material used.

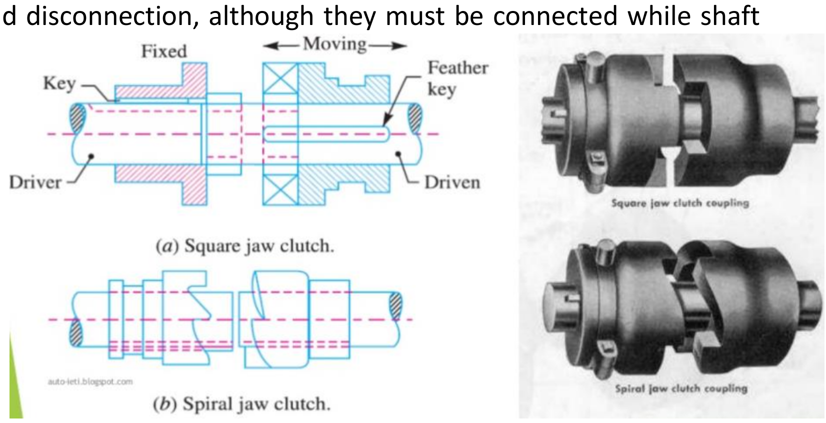


Figure 2.5 : Square and Spiral Jaw

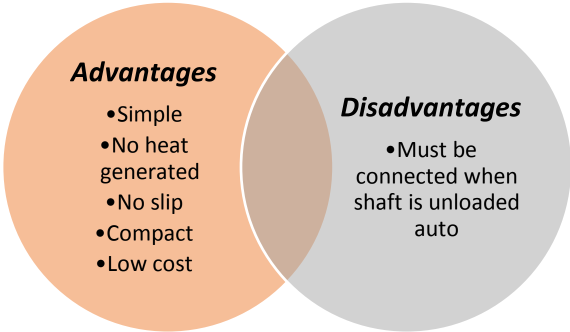


FIGURE 2.6: POSITIVE CLUTCH ADVANTAGES AND DISADVANTAGES

2.2.4.2 FRICTION CLUTCH

Friction Clutches work on the basis of the frictional forces developed between the two or more surfaces in contact. Friction clutches are usually over the jaw clutches due to their better performance.

2.2.4.2.1 PLATE OR DISC CLUTCH

In this category clutches are classified by the construction of clutches. In plate or disc clutch friction plate or pressure plate is used and for this, this types clutches are known as plate or disc clutch. Types of these clutches are as follows.

a. Single plate clutch

As per name single plate clutch consist a single friction plate or clutch plate. It consist different parts but principle of working is same as per discuss above that is by sliding clutch plate, engage and disengage of shaft is done. It is use there, where radial space is more like in trucks and buses.

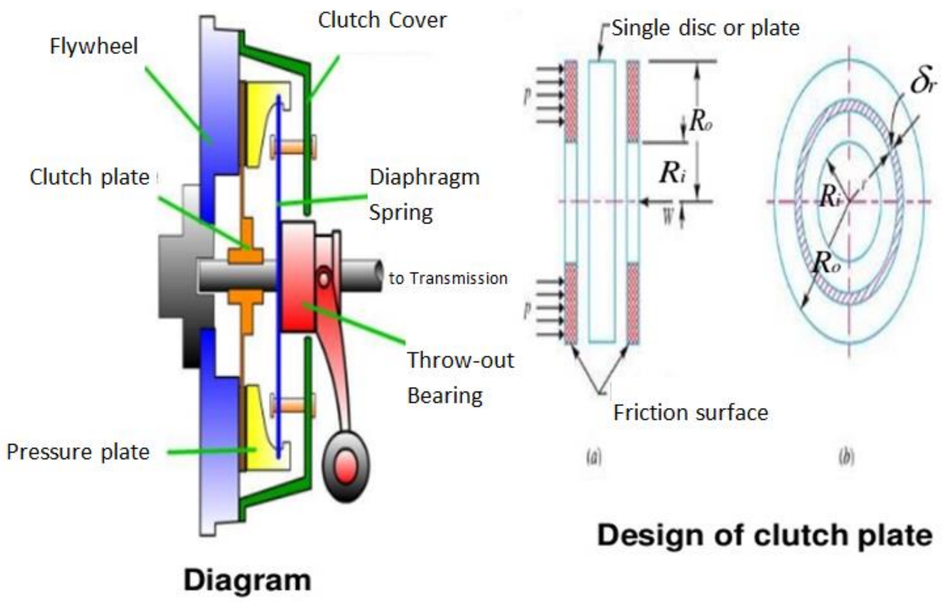


FIGURE 2.7: SINGLE PLATE CLUTCH

b. Multi plate clutch

Multi plate clutch consist multiple pressure plate and this pressure plate is use to develop friction force. And this friction force is used to transmit torque. The operating principle is same as single plate clutch, by sliding pressure plate engage and disengage of shafts is done. This type clutches use where space is limited like in motorcycle.

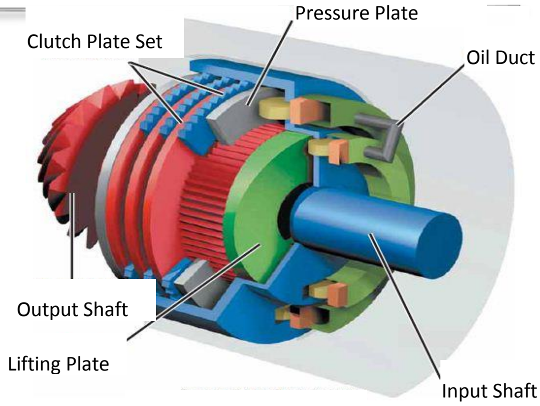


FIGURE 2.8: MULTI PLATE CLUTCH

2.2.4.2.2 *CONE CLUTCH*

Cone clutch consist cup and cone. Cup has inner conical cavity and cone has outer conical shape. Cone is inserted in cup and on outer surface of cone friction material or friction lining is used. When cone is inserted in cup friction force is develop and this friction force is used for transmitting torque driving shaft to driven shaft. Cup is fixed to driving shaft and cone is free to slide axially on splined driven shaft. By sliding cone engage and disengage is done. Cone clutch is not widely used because high axial thrust is required to engage or disengage driven shaft from driving shaft.

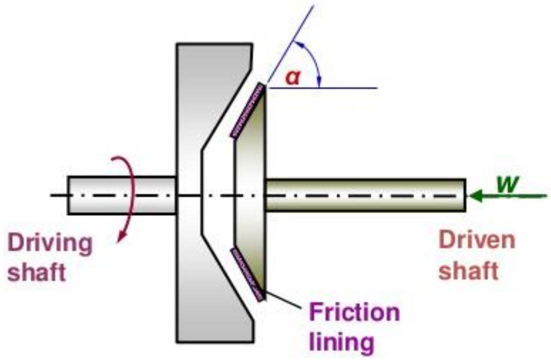


FIGURE 2.9: CONE CLUTCH

2.2.4.2.3 *CENTRIFUGAL CLUTCH*

The name centrifugal clutch is come from the centrifugal force is used in clutch. Principle of working of centrifugal clutch is that, it consist clutch drum of circular shape, spider, helical spring, shoes with friction lining at outer side. Clutch drum is fixed to driven shaft and spider and shoes is connected to driving shaft. Shoes with outer friction lining are connecting at centre with the help of helical spring and it free to move or slide in spider as centrifugal force is increase. Centrifugal force is increase with increase in speed. Then shoes move outside and come in contact with drum or engage with drum and due to friction lining friction force is develop and this is use to transmit torque. And when speed is reducing shoes come back and disengage with clutch drum. In this way cone clutch is work. Centrifugal clutches are use in scooter where automatic gear transmission is used.

FIGURE 2.10: CENTRIFUGAL CLUTCH

2.2.5 ADVANTAGES OF CLUTCHES IN ENGINEERING

Advantages of Clutches in Engineering

1. The engagement is smooth
2. 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 operations requires frequent starts and stops
3. In certains 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 transmissions chain

2.3 BRAKE

A brake is a mechanical device which produces frictional resistance against moving machine member, in order to **slow down** or **stop** the motion of machine.

In the process of performing this function, the brake absorbs kinetic energy of moving member and the brake absorbs potential energy of lowering member. The energy absorbed by brakes is released to surrounding in form of heat.

2.3.1 TYPES OF BRAKES

Brakes may be broadly described as using friction, pumping, or electromagnetics. One brake may use several principles: for example, a pump may pass fluid through an orifice to create friction:

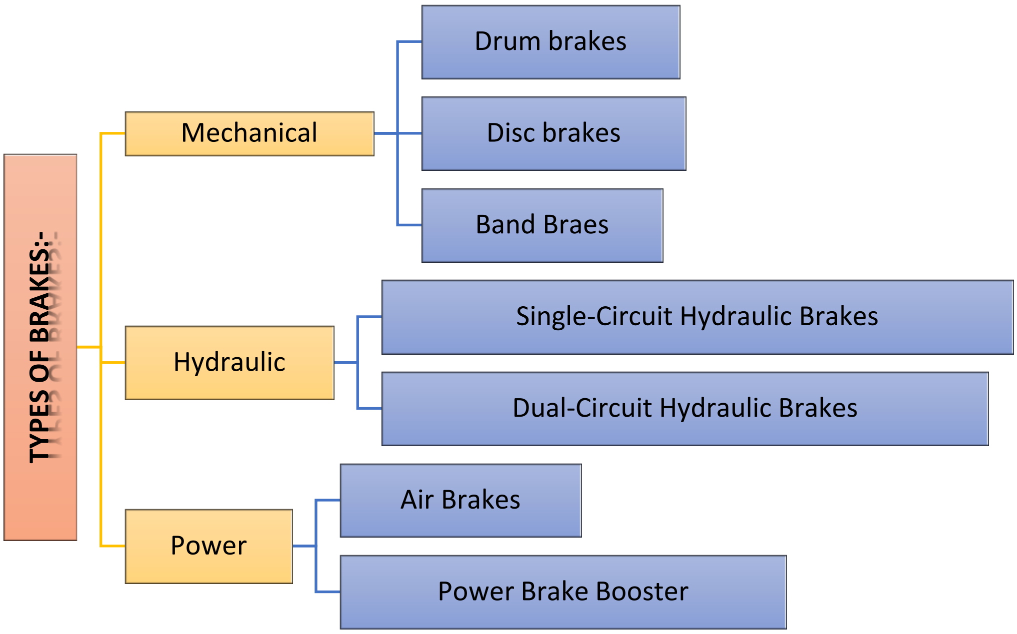


FIGURE 2.11: TYPES OF BRAKES

2.3.2 Drum Brakes

Drum brakes work on the same principle as disc brakes. Shoes press against a spinning surface. In this system, that surface is called a drum.

Many cars have drum brakes on the rear wheels and disc brakes on the front. Drum brakes have more parts than disc brakes and are harder to service, but they are less expensive to manufacture, and they easily incorporate an emergency brake mechanism.

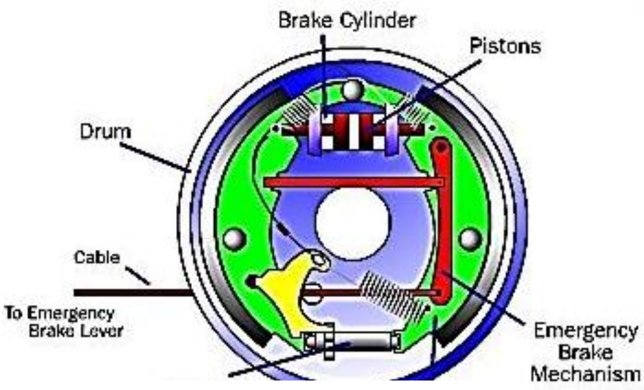


Figure 2.12: Drum Brakes

2.3.3 Disc Brakes

A disc brake consists of a cast iron disc bolted to wheel hub and stationary housing called caliper. Caliper is connected to some stationary part of vehicle like axle. When brakes are applied, piston move friction pads into contact with disc, applying equal and opposite force on disc. On releasing brakes, the rubber sealing rings act as return springs and retract piston and friction pads away from disc.

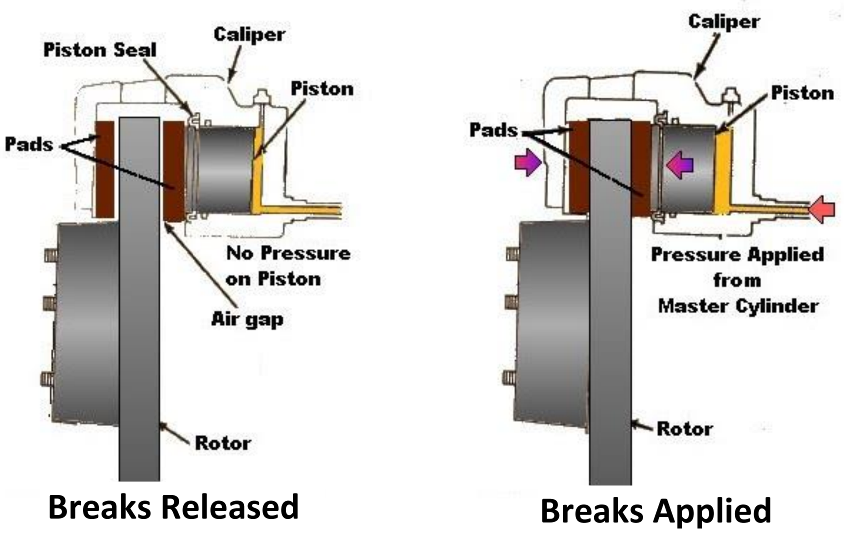


Figure 2.13: Disc Brakes

2.3.4 Band Brakes

Band brakes tighten a ribbon of high-friction material around a pulley attached to the rotating axle. They are often employed on bicycles. If the pull on the band is in the direction of axle rotation the brake is self-energizing. Differential band brakes attach both ends of the brake ribbon to the lever to

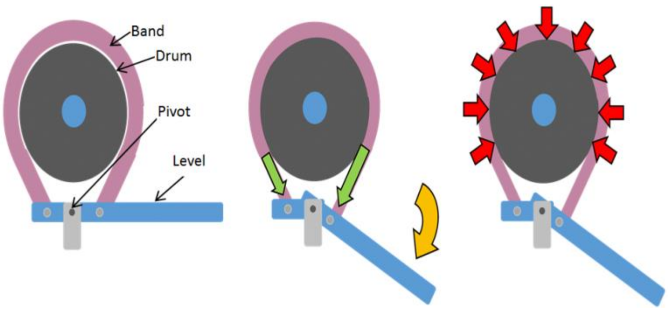


Figure 2.14: Band Brakes

supply braking power for bi-directional shafts.

2.3.5 HYDRAULIC BRAKES OPERATION

A hydraulic brake is an arrangement of braking mechanism which uses brake fluid, typically containing glycol ethers or diethylene glycol, to transfer pressure from the controlling mechanism to the braking mechanism.

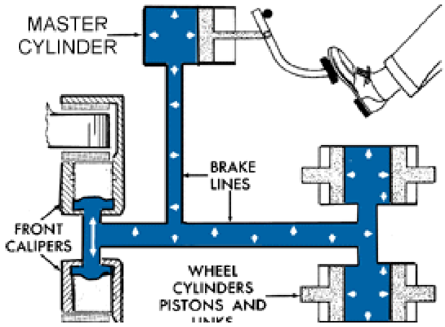


FIGURE 2.15: HYDRAULIC BRAKES

As the brake pedal is pressed, a pushrod exerts force on the piston(s) in the master cylinder. This forces fluid through the hydraulic lines toward calipers. The brake caliper piston(s) then apply force to the brake pads. This causes them to be pushed against the spinning rotor, and the friction between the pads and the rotor causes a braking torque to be generated, slowing the vehicle.

2.3.6 ADVANTAGES AND DISADVANTAGES OF BRAKING SYSTEM

	Advantages	Disadvantages
Drum Brake	• High brake factor (low actuation effort) • Easy to integrate with park brake	• Not consistent in torque performance
Disk Brake	• Resist high temperature • Brake pads are easily replaceable • The condition of brake pads can be checked without much dismantling of brake system • High stability • Easy to install • Can use cable instead of hydraulic power • Cheap	• More force is needed be applied as the brakes are not self-emerging • Pad wear is more • High actuation effort
Hydraulic Brake	• Equal braking action on all wheels. • Increased braking force • Simple in construction • Low wear rate of brake linings • Flexibility of brake linings • Increased mechanical advantage • Better stopping power than the other Brakes	• Whole braking system fails due to leakage of fluid from brake linings • More maintenance than other system • Presence of air inside the tubing ruins the whole system.
Band Brake	• Simple • Compact • Rugged • Generate high force with a light input force	• Band brakes are prone to grabbing or chatter and loss of brake force when hot

2.4 TORQUE TRANSMISSION CAPACITY OF OLD AND NEW DISK CLUTCHES

Torque transmission capacity of clutch is determined by two approaches (theories).

- uniform pressure theory
- uniform wear theory

In uniform pressure theory it is assumed that the pressure on friction lining is same from inner radius to outer radius, whereas in uniform wear theory it is assumed that the pressure is more at inner edge of lining and goes on reducing towards outer lining.

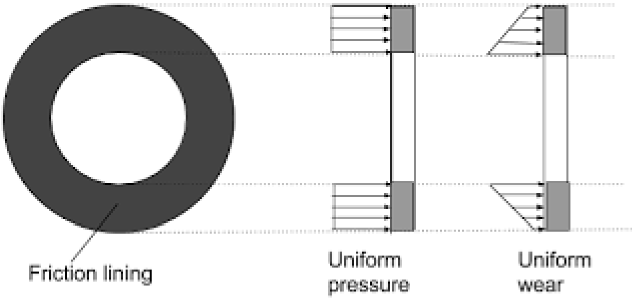


FIGURE 2.16 : OLD AND NEW DISK CLUTCHES

Power transmitted by clutch is given by,

Pow = ωT

$$= \frac{2\pi NT}{60} \text{ watt}$$

Where, N = speed of driving shaft in rpm

T = torque transmitted by clutch in Nm

2.4.1 TORQUE TRANSMISSION UNDER UNIFORM PRESSURE

This theory is applicable to new clutches. In new clutches employing a number of springs, the pressure can be assumed as uniformly distributed over the entire surface area of the friction disk. With this assumption, the intensity of pressure between disks is regarded as constant

2.4.2 TORQUE TRANSMISSION UNDER UNIFORM WEAR

This theory is based on the fact that wear is uniformly distributed over the entire surface area of friction disk. This assumption can be used for worn out clutches/old clutches. The axial wear of the friction disk is proportional to frictional work.

When the clutch plate is new and rigid, the wear at the outer radius will be more, which will reduce pressure at the outer edge due to rigid pressure plate. This will change pressure distribution. During running condition, the pressure distribution is adjusted constantly.

The uniform-pressure theory is applicable only when the friction lining is new. When the friction lining is used over a period of time, wear occurs. Therefore, the major portion of the life of friction lining comes under uniform-wear criterion. Hence, in the design of clutches, the uniform wear theory is used.

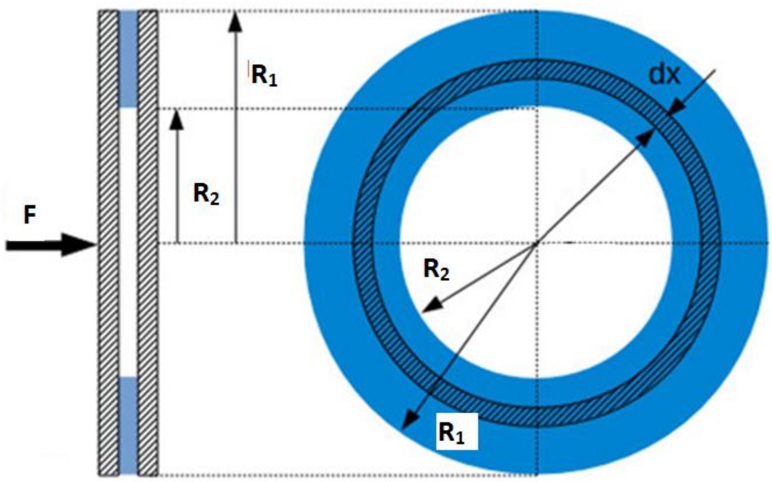


FIGURE 2.17 : PLATE CLUTCH

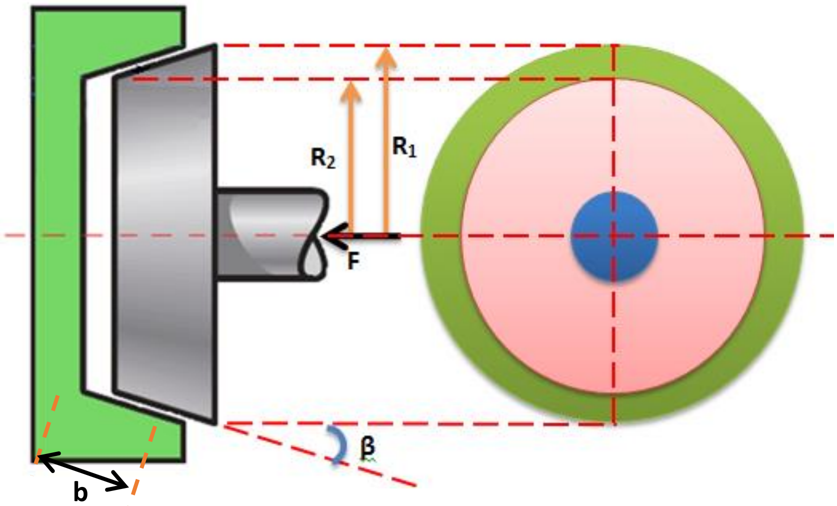


FIGURE 2.18 : CONE CLUTCH

If the clutch is engaged when one member is stationary and other rotating, then the cone faces will tend to slide on each other in the direction of an element of the cone. This will resist the engagement and then force.

$$\text{Force width, b} = \left[\frac{R_1 - R_2}{\sin\beta}\right]$$

$$\text{Outer radius, } R_1 = R_M + \left[\frac{\text{b Sin}\beta}{2}\right]$$

$$\text{Inner radius } R_2 = R_M - \left[\frac{\text{b Sin}\beta}{2}\right]$$

$$\text{Mean radius, } R_M = \frac{R_1 + R_2}{2}$$

$$\text{Axial force, F} = \pi D_M p b \sin\beta$$

Where,

F = total actuating force /axial force, N

T = Torque transmitted by friction, Nmm

μ = coefficient friction between the surfaces

p = normal/axial intensity of pressure, N/mm^2

R_1 = outer radius,mm

R_2 = inner radius, mm

R_m = mean radius, mm

b = face width, mm

β = cone pitch/face/ Semi cone angle, °

$n = n_1 + n_2 - 1$

n_1 = number of plates in driving shaft

n_2 = number of plates in driven shaft

n = number of pair of active surfaces

D_M = Mean diameter, mm

TYPES OF CLUTCHES	UNIFORM PRESSURE THEORY (NEW CLUTCH)	UNIFORM WEAR THEORY (OLD CLUTCH)
PLATE	$F = \pi p({R_1}^2 - {R_2}^2)$	$F = 2\pi p R_2(R_1 - R_2)$
	$T = \frac{2}{3}\mu F\left[\frac{{R_1}^3 - {R_2}^3}{{R_1}^2 - {R_2}^2}\right]n$	$T = \frac{1}{2}\mu F(R_1 + R_2)$ n
CONE	$F = \pi p({R_1}^2 - {R_2}^2)$	$F = 2\pi p R_2(R_1 - R_2)$
	$T = \frac{2\mu F}{3\sin\beta}\left[\frac{{R_1}^3 - {R_2}^3}{{R_1}^2 - {R_2}^2}\right]$	$T = \frac{\mu F}{\sin\beta}\left[\frac{R_1 + R_2}{2}\right]$

Example 1:

The inner and the outer radii of a single plate clutch are 40 mm and 80 mm respectively. Determine the maximum and minimum pressure when the axial force is 3 kN

Solution:

Given,
R_2 = 40mm
R_1 = 80mm
p_{max} = ?
p_{min} = ?
F = 3kN
single plate

Assume clutch with wear theory,

The maximum pressure will be at the inner radius,

$$F = 2\pi p R_2(R_1 - R_2)$$

3k = 2(π)p(40)(80-40)

$\therefore$ **p_{max}** = $0.2984 N/mm^2$

= **$\underline{298.4 kN/m^2}$** #

The minimum pressure will be at the outer radius,

$$F = 2\pi p R_1(R_1 - R_2)$$

3k = 2(π)p(80)(80-40)

∴ p_{min} = 0.1492N/mm^2= **149.2kN/m^2** #

Example 2:

Maximum pressure intensity between a single plate clutch is 85 kN/m^2 at 900rpm The outer and inner diameter of the plate are 360 mm and 263 mm. Both the sides of the plate are effective and the coefficient of friction is 0.25. Determine the axial force to engage the clutch and power needed to transmit the required torque.

Solution:

Given,
p = 85kN/m^2
N = 900rpm
D_1= 360mm,
R_1 =0. 18m
D_2= 263mm,
R_2 = 0.1315m
μ = 0.25
n=2
F = ?
Pow =?

$\Big\}$ single plate

In case of power transmission through a clutch, it is safer to use the expressions obtained by uniform wear theory.

F,

$$\boldsymbol{F} = 2\pi p R_2(R_1 - R_2)$$

$= 2\pi(85k)(0.1315)(0.18-0.1315)$

= **3406.17N** #

Pow,

Pow = $\frac{2\pi NT}{60}$

$T = \frac{1}{2}\mu F(R_1 + R_2)$ n

$T = \frac{1}{2}(0.25)(3406.17)(0.18 + 0.1315)2$

= 265.255Nm

∴ **Pow** = $\frac{2\pi NT}{60}$ = $\frac{2\pi(900)(265.255)}{60}$ = **25kW** #

Example 3:

A multi-plate friction clutch has three contact surfaces is used to rotate a machine from a shaft rotating at a uniform speed of 250 rpm. The disc type clutch has both of its sides effective, the coefficient of friction being 0.3. The outer and the inner diameters of the friction plate are 200 mm and 120 mm respectively. Assuming uniform wear of the clutch, the intensity of pressure is not to be more than 100 kN/m^2. Calculate the power that can be transmitted using uniform pressure and wear theory.

Solution:

Given,
n=3
N = 250rpm
μ = 0.3
D_1= 200mm,
R_1 =0. 1m
D_2= 120mm,
R_2 = 0.06m
p = 100kN/m^2
Pow =?

$\Big\}$ multi plate

Pow = $\frac{2\pi NT}{60}$

Uniform Pressure Theory,

$$T = \frac{2}{3}\mu F\left[\frac{{R_1}^3 - {R_2}^3}{{R_1}^2 - {R_2}^2}\right]n$$

$$F = \pi p({R_1}^2 - {R_2}^2)$$

$$F = \pi(100k)(0.1^2 - 0.06^2) = 2010.62 \text{ N}$$

$$T = \frac{2}{3}(0.3)(2010.62)\left[\frac{0.1^3 - 0.06^3}{0.1^2 - 0.06^2}\right](3)$$

= 147.781 Nm

∴ **Pow** = $\frac{2\pi(250)(147.781)}{60} = \mathbf{3.869kW}$ #

Uniform Wear Theory,

$$T = \frac{1}{2}\mu F(R_1 + R_2)\ \mathrm{n}$$

$$F = 2\pi p R_2(R_1 - R_2)$$

$$F = 2\pi(100k)(0.06)(0.1 - 0.06) = 1507.96\ \mathrm{N}$$

$$T = \frac{1}{2}(0.3)(1507.96)(0.1 + 0.06)(3) = 108.573\ \mathrm{Nm}$$

$$\therefore \textbf{Pow} = \frac{2\pi(250)(108.573)}{60} = \mathbf{2.842kW}\ \#$$

Example 4:

A multi-plate disc clutch transmits 55 kW of power at 1800 rpm. Coefficient of friction for the friction surfaces is 0.1. Axial intensity of pressure is not to exceed 160 kN/m^2. The internal radius is 80 mm and is 0.7 times the external radius. Find the number of plates needed to transmit the required torque.

Solution:

Given,
Pow = 55kW
N = 1800rpm
μ = 0.1
p = 160kN/m^2
R_2= 0.08m
R_2 =0.7R_1
$\therefore$ R_1 = 0.08/0.7)
= 0.1143m
n =?

$\left.\right\}$ multi plate

Assume clutch with wear theory,

$$\text{Pow} = \frac{2\pi NT}{60}$$

$$55k = \frac{2\pi(1800)T}{60}$$

$\therefore$ T = 291.784Nm

$$F = 2\pi p R_2(R_1 - R_2)$$

$$F = 2\pi(160k)(0.08)(0.1143 - 0.08) = 2758.57\ \mathrm{N}$$

$$T = \frac{1}{2}\mu F(R_1 + R_2)\ \mathrm{n}$$

$$291.784 = \frac{1}{2}(0.1)(2758.57)(0.1143 + 0.08)\ \mathrm{n}$$

$\therefore$ **n** = 10.888 ≈ **<u>11 number of surfaces</u>** #

Example 5:

A conical clutch has an included angle of 120°. The outer and inner diameter are 80 mm and 20 mm respectively. Calculate the force required to press the two halves together if it is to transmit 200kW at 600rev/min. The coefficient of friction is 0.3. Use both the uniform wear and the uniform pressure theory.

Solution:

Given,
B = 120°
D_1= 0.08m, R_1=0.04m
D_2= 0.02m, R_2=0.01m
F=?
Pow = 200kW
N = 600rpm
μ = 0.3

$\left.\right\}$ cone plate

$$\text{Pow} = \frac{2\pi NT}{60}$$

$$200k = \frac{2\pi(600)T}{60}$$

$\therefore$ T = 3183.098Nm

Uniform Pressure Theory,

$$T = \frac{2\mu F}{3\sin\beta}\left[\frac{{R_1}^3 - {R_2}^3}{{R_1}^2 - {R_2}^2}\right]$$

$$\mathbf{3183.098 = \frac{2(0.3)\mathit{F}}{3(\mathit{sin}120)}\left[\frac{0.04^3 - 0.01^3}{0.04^2 - 0.01^2}\right]}$$

$\therefore$ F = <u>328.172kN</u> #

Uniform Wear Theory,

$$\mathbf{T = \frac{\mu F}{\sin\beta}\left[\frac{R_1 + R_2}{2}\right]}$$

$$\mathbf{3183.098 = \frac{(0.3)F}{\sin 120}\left[\frac{0.04 + 0.01}{2}\right]}$$

$\therefore$ F = <u>367.552kN</u> #

Example 6:

A cone clutch is to transmit 7.5 KW at 600 rpm. The face width is 50mm, mean diameter is 300mm and the face angle 15°. Assuming co efficient of friction as 0.2, determine the axial force necessary to hold the clutch parts together and the normal pressure on the cone surface.

Solution:

Given,
Pow = 7.5kW
N = 600rpm
b = 0.05m
D_M = 0.3m, R_M = 0.15m
β = 15°
μ = 0.2
F = ?
p = ?

cone plate

Assume clutch with wear theory,

$$F = 2\pi p R_2(R_1 - R_2)$$

Pow = $\frac{2\pi NT}{60}$

7.5k = $\frac{2\pi(600)T}{60}$

$\therefore$ T = 119.366Nm

$$T = \frac{\mu F}{\sin\beta}\left[\frac{R_1 + R_2}{2}\right]$$

R_M = $\frac{R_1+R_2}{2}$=0.15m

$$119.366 = \frac{(02)F}{\sin 15}(0{,}15)$$

$\therefore$ F =<u>1029.808 N</u> #

$$F = 2\pi p R_2(R_1 - R_2)$$

R_1 = R_M + $\left[\frac{b\ \mathrm{Sin}\beta}{2}\right] = 0.15 + \left[\frac{0.05\ \mathrm{Sin}15}{2}\right] = 0.1565m$

R_2 = R_M - $\left[\frac{b\ \mathrm{Sin}\beta}{2}\right] = 0.15 - \left[\frac{0.05\ \mathrm{Sin}15}{2}\right] = 0.1435\mathrm{m}$

Thus, $\quad 1029.808 = 2\pi p(0.1435)(0.1565 - 0.1435)$

$\therefore$ **P = $\underline{87.858kNm^{-2}}$ #**

1. The following data is for a conical clutch.

Inside dimeter	30mm
Outside diameter	110 mm
Coefficient of friction	0.23
Axial force	800 N
Included angle	80^{o}
Speed	1000 rev/min

Calculate the torque and power that can be transmitted without slipping using

a) The uniform pressure theory
b) The uniform wear theory

2. The following data is for a conical clutch.

Inside dimeter	20mm
Outside diameter	120 mm
Coefficient of friction	0.3
Included angle	100^{o}
Speed	3000 rev/min

Calculate the axial force needed to allow the transmission 800 watts without slipping using

a) The uniform pressure theory
b) The uniform wear theory

3. A friction cone clutch has to transmit a torque of 200 N-m at 1440 rpm. The longer diameter of the cone is 350mm. The cone pitch angle is 6.25°the force width is 65mm. the coefficient of friction is 0.2. Determine i) the axial force required to transmit the torque. ii) The average normal pressure on the contact surface when maximum torque is transmitted.

4. An engine developing 30 KW at 1250 rpm is filted with a cone clutch. The cone face angle of 12.5°. The mean diameter is 400 rpm μ = 0.3 and the normal pressure is not to exceed 0.08 N / mm^2. Determine the axial force and torque to engage the clutch.

5. The following data is for a multi-plate clutch.

Number of contact surfaces	5
Speed	2000 rev/min
Outside diameter	150 mm

Inside dimeter	80mm
Coefficient of friction	0.25
Axial force	600 N

a) Calculate the maximum power that the clutch can transmit without slipping based on constant wear theory.
b) Calculate the maximum power that the clutch can transmit without slipping based on constant pressure theory.

6. A multi-plate clutch must transmit 20 kW of power without slipping at 4000 rpm. The coefficient of friction is 0.28. The inner and outer diameters are 80 mm and 160 mm respectively. The axial force applied to the plates is 460 N. Determine the number of plates required using

   a) The uniform pressure theory
   b) The uniform wear theory

7. A multi-plate clutch has three contact surfaces and transmits power at 1500 rpm. The coefficient of friction is 0.4. The inner and outer diameters are 30 mm and 150 mm respectively. The axial force applied to the plates is 400 N. Calculate the torque and power that can be transmitted without slipping using:

   a) The uniform pressure theory
   b) The uniform wear theory

Previous Years' Question Papers

Jun 17

a. Discuss THREE(3) functions of clutches in engineering 6M

b. Draw these types of brake:

   i. Disc brake
   ii. Bend type break

6 M

Jun 17 (PK)

a. Discuss THREE(3) advantages of clutches in engineering 6M

Dis 16

a. Describe the function of clutch. 4M

b. List THREE (3) advantages of using the braking system below;

   i. Bend type
   ii. Clutch type

6M

Jun 16

a. Identify THREE (3) advantages of clutch in engineering. 6M

b. Draw bent type (mechanical) brake. 5M

Dis 15

a. State the functions of a clutch. 4M

b. State THREE (3) requirements in the design of clutch. 6M

c. The mean diameter of a contact surface of a clutch cone is 320 mm and the width of the surface is 50mm. The value for the coefficient of friction for the clutch is 0.33. The angle between the surface and the axial of the clutch is 15^o. If the normal pressure that can be apply to the surface and the clutch is limited to $85kN/m^2$, determine:

   i. The maximum power that can be transmit at the velocity of 1500rpm. 9M

   ii. The minimum axial force that should be apply to make sure that the clutch is remain in it entry direction. 6M

Jun 15

a. State the function of clutch. 5M

b. The following data is for multiplayer clutch

No of contact surfaces : 5

Speed rev/min : 2000

Outside diameter mm : 150

Inside diameter mm : 80

Coefficient of friction : 0.25

Axial force R : 600N

c. Calculate the maximum power that the clutch can transmit without slipping, based on constant wear theory. 10M

d. Calculate the maximum power that the clutch can transmit without slipping, based on constant pressure theory. 10M

Dis 14

a. List FOUR (4) groups of clutch classification. 4M

b. Differentiate between clutch system and brake system based on initial and final conditions. 8M

c. A clutch plate consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 100 and 200mm respectively. The coefficient of friction is 0.2 and the permissible intensity of pressure is $1 N/mm^2$. Assuming uniform wear theory, calculate the power transmitting capacity of the clutch at 750rpm. 9M

d. List FOUR (4) factors that affects brake performance. 4M

CHAPTER 3:
CHAIN TRANSMISSIONS

OBJECTIVES:

In the end of this chapter, student should know to:

Explain the concept of chain transmission.

3.1 INTRODUCTION

This topic introduces chain drives to students and the calculation involved to determine the transferred power.

3.2 DEFINITION

- Chain drive is a way of transmitting mechanical power from one place to another.
- It is often used to convey power to the wheels of a vehicle, particularly bicycles and motorcycles. It is also used in a wide variety of machines besides vehicles.
- It transmits power by means of tensile forces, and is used primarily for power transmission and conveyance systems

3.3 FUNCTION

In general the use of chains for engineering can be divided into 3 categories.

- The chain to lift the load, it can be moved at a maximum speed of 15 m / min.
- Chain conveyer for product transfer, it can be used up to a maximum of 120 m / min.
- Power transfer chain can operate at a maximum speed of 900 m / minute

The advantages of using chain transmission:

- Force can be transferred in long distance.
- Low noise for toothed operation.
- High bending reliability for power transfer using sprocket.
- High heat resistance for chain.
- Easy to maintain.

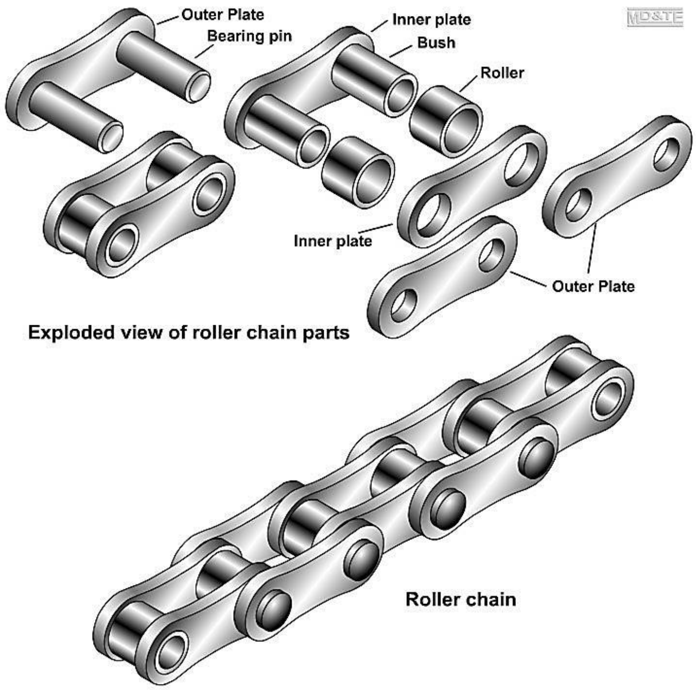
The disadvantages of using chain transmission:

- Drive belts can often slip (unless they have teeth) which means that the output side may not rotate at a precise speed.
- Some work gets lost to the friction of the belt against its rollers.
- Teeth on toothed drive belts generally wear faster than links on chains.
- Roller chain drives suffer the potential for vibration.

3.4 TYPES OF DRIVE CHAIN

3.4.1 ROLLER CHAINS

Roller chain or bush roller chain is the type of chain drive most commonly used for transmission of mechanical power on many kinds of domestic, industrial and agricultural machinery, including conveyors, wire and tube-drawing machines, printing presses, cars, motorcycles, and bicycles. It consists of a series of short cylindrical rollers held together by side links. It is driven by a toothed wheel called a sprocket. It is a simple, reliable, and efficient means of power transmission.



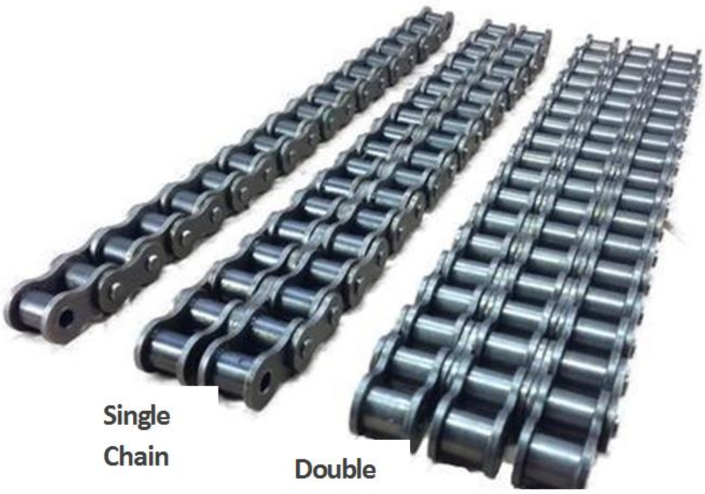
Exploded view of roller chain parts

Roller chain

Based on the figure above, it shows the part of roller chains and the connection involved. The inner link consists of two inner plates into which, two bushes are pressed and two rollers, which rotate on the bushes. The outer link consists of two outer plates and two bearing pins.

The rollers which rotate on the bushes run with little friction on the teeth of the chain wheel, as there is a constant change in contact area. The grease film between rollers and bushes contributes towards silent running and absorbs shocks.

The roller chains can be used in multiple strand. Drive using a multiple smaller pitch chain will run more quietly than a single chain with a larger pitch. Multiple chains are more sensitive to misalignment of the chain wheels

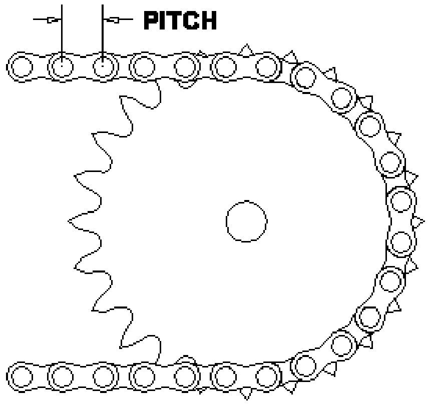


Single Chain

Double Chain

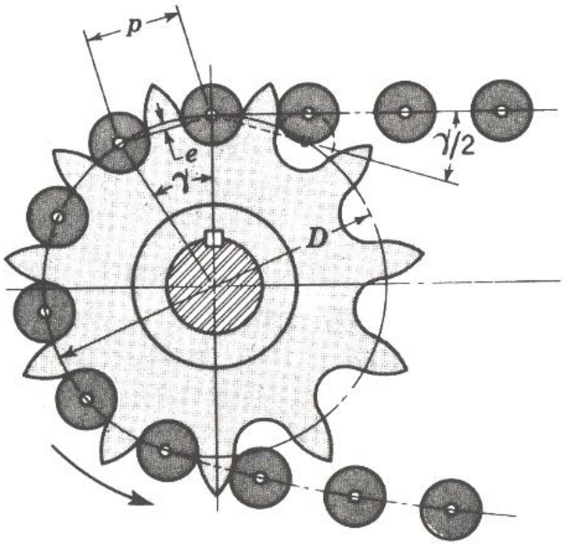
Triple Chain

3.4.2 SPROCKET



A sprocket or sprocket-wheel is a profiled wheel with teeth, or cogs, that mesh with a chain, track or other perforated or indented material. The name 'sprocket' applies generally to any wheel upon which radial projections engage a chain passing over it. It was made of carbon for small size and iron for the big size.

3.4.3 ENGAGEMENT OF A CHAIN AND SPROCKET



Pitch, p chain is a linear distance between the center or midpoint of ball chain with ball chain next. A Sprocket driving a chain and rotates in a counterclockwise direction. Denoting the chain pitch by p, the pitch angle by γ, and the pitch diameter of the sprocket by D, using trigonometry we can get equation:

$$\sin\frac{\gamma}{2} = \frac{p/2}{D/2} \qquad \text{So,} \quad D = \frac{p}{\sin\left(\frac{\gamma}{2}\right)}$$

From the figure, we know that $\gamma = \frac{360}{N}$, therefore the diameter of the sprocket, D :

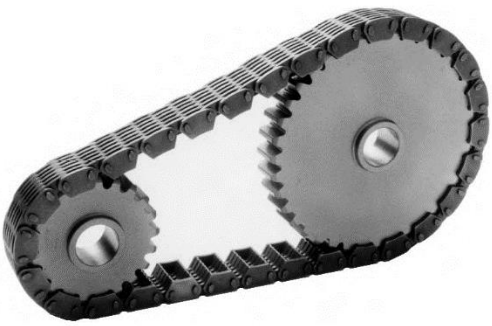
$$\boxed{D = \frac{p}{\sin\frac{180}{Z}}}$$

Where, p = chain pitch, mm

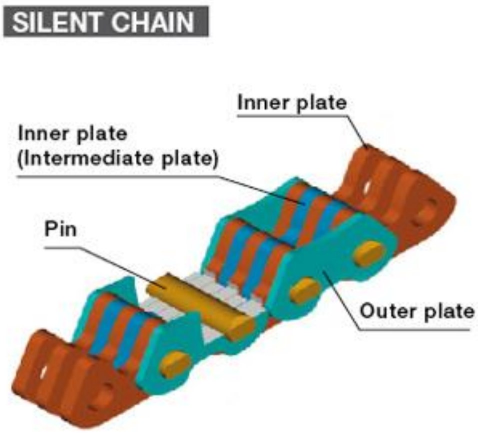
Z_1 = number of teeth for drive sprocket

Z_2 = number of teeth for driven sprocket

3.4.4 SILENT CHAINS



A silent chain is essentially an assemblage of gear racks, each with two teeth, pivotally connected to form a closed chain. The links are pin-connected, flat steel plates with straight teeth. Silent chain consists of a series of toothed link-plates assembled on pin connectors, permitting smooth joint articulation. Silent Chain consists of many links positioned close to each other. This type of chain is capable of carrying more power per chain area than standard roller chain. The plates of the chain fit into the sprocket very closely and the chain runs very smoothly with very little vibration.

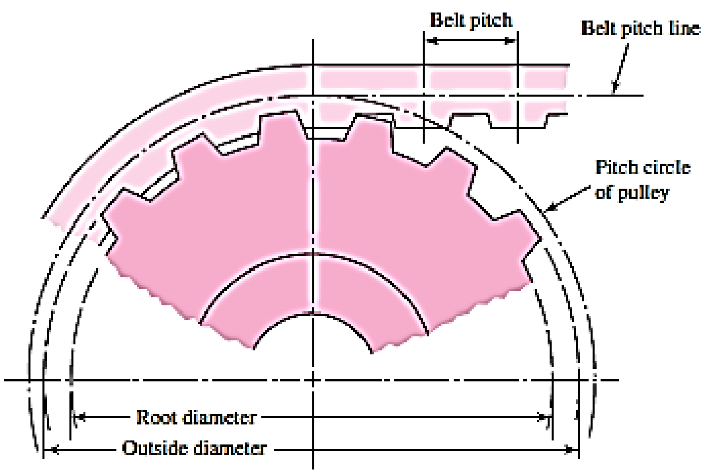


These chains can be found in a wide variety of demanding industrial applications and similar designs are used in 4 wheel drive automobiles and NASCAR racing engines.

In conveying applications, silent conveying chains provide a heat resistant, flat, durable, non-slip, conveying surface which can be customized to fit a wide range of industrial applications.

3.4.5 TOOTHED BELT

A toothed belt is made of a rubberized fabric coated with a nylon fabric, and has steel wire within to take the tension load. It has teeth that fit into grooves cut on the periphery of the pulleys. Toothed belts are non-slipping mechanical drive belts. They are made as flexible belts with teeth moulded onto their inner surface. The belts run over matching toothed pulleys or sprockets. When correctly tensioned, these type of belts have no slippage, and are often used to transfer motion for indexing or timing purposes. They are often used in chains or gears, so there is less noise and a lubrication bath is not necessary.



Toothed belts are used widely in mechanical devices, including sewing machines, photocopiers and many others. A major use of toothed belts is as the timing belt used to drive the camshafts within an automobile or motorcycle engine.

As toothed belts can deliver more power than a friction-drive belt, they are used for high-power transmissions.

3.4.6 ADVANTAGES OF CHAIN DRIVE OVER THE BELT AND GEAR

- Unlike a belt drive, the chain driver does not slip, so no power loss makes it more efficient.
- More compact than conveyor belt. For specific capacity, chain drivers are narrower than conveyor drives and the sprocket diameter is smaller than sheaves on the belt drive.
- Chain drivers are more practical at slow speeds than are more efficient at high temperatures.

- Generally easier to install than conveyor belt.
- The chain does not depend on the oil, grease or sunlight impairment and is generally resistant to abrasion.
- The chain may operate in wet conditions.
- Chain stretching in normal operating conditions is low and less adjustment is needed than conveyor belt.
- Only two gear wheels and one chain are required to transmit rotational motion at a certain distance versus gear range which requires a lot of gear to be arranged and connected with each other to transmit motion.
- The chain driver is generally simpler or easier and costs are lower than the gear drive and can be used for shaft centers at varying distances.
- The chain is suitable for the use of reversing.
-

3.5 METHOD OF INSTALLATION AND CUSTODY FOR THE CHAIN

3.5.1 METHOD OF INSTALLATION OF CHAIN:

1. Shaft alignment

Make sure that all shafts are parallel and level. Check alignment with a spirit level. The shafts should be supported by sufficiently strong bearings to avoid any displacement during operation.

2. Installation of Sprockets

Align the sprockets exactly on the shafts. Check with a straight edge of a string held against the sides of the sprocket face. Improper alignment of sprockets will cause abnormal wear on the chain link plates and on the sides of the sprocket teeth. Check occasionally during operation for such wear

3. Mounting of Chain

Wrap the chain around the sprockets and bring the two ends together on one sprocket to connect them with a connecting link.

4. Chain Tension

The chain should never run with both sides tight. To check tension, turn one sprocket to tighten the lower span of the chain. Then measure the sag of the lower strand which should be about to 2 to 3% of the tangent to the sprockets. In an inclined drive the sag should be less. In vertical drives a chain tensioner must be provided for.

3.5.2 METHOD OF MAINTENANCE OF CHAIN:

1. Always checking the elasticity of the chain. Make sure it setting properly to avoid broken or slip from the sprocket.
2. Always put grease / oil on the chain for lubrication.
3. Always observe the sound of the chain.
4. Always be cleaned / put cover to avoid from dirt.

3.6 CALCULATE VALUE RELATED TO THE CHAIN DRIVE

3.6.1 VELOCITY

The chain velocity, *V* is defined as the number of feet coming off the sprocket per unit time. Thus the chain velocity in feet per minute is:

$$V = \frac{Npn}{12}$$

The velocity ratio of the chain, $i = \dfrac{\text{input speed}}{\text{output speed}}$

Therefore, $i = \dfrac{N_1}{N_2} = \dfrac{Z_2}{Z_1}$

Where p = chain pitch, mm

Z_1 = number of drive sprocket teeth

Z_2 = number of driven sprocket teeth

N_1 = speed of drive sprocket, rpm

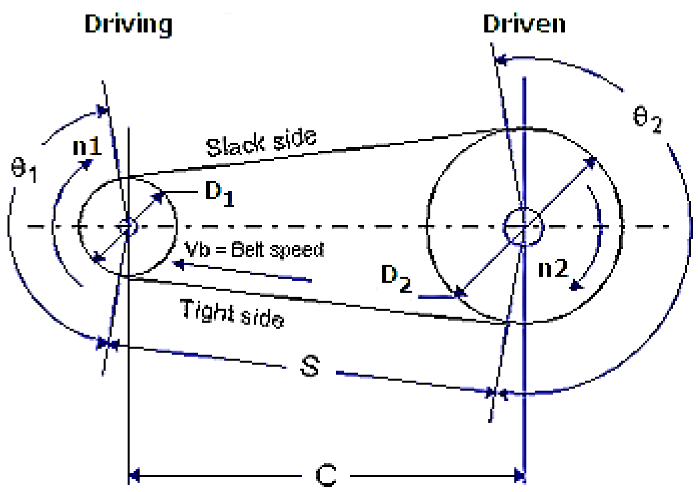
N_2 = speed of driven sprocket, rpm

The chain velocity in milimeter per second can be determine as :

$$V = r\omega$$

Where $r = radius\ of\ sprocket$, mm

And angular velocity, $\omega = \dfrac{2\pi N}{60}$ where N = Sprocket speed, rpm



Note: D1 and D2 are pitch diameters

3.6.2 WRAP ANGLE OF CHAIN DRIVE

The wrap angle, θ can be determine as:

$$Warp\ angle\ for\ drive\ sprocket, \theta_1 = 180° - 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right]$$

$$Warp\ angle\ for\ driven\ sprocket, \theta_2 = 180° - 2\sin^{-1}\left[\frac{D_1 - D_2}{2C}\right]$$

Where, D_1 = diameter of the driven sprocket, mm

D_2 = diameter of the drive sprocket, mm

3.6.3 LENGTH OF CHAIN

The length of chain, L in millimeter can be determined as:

$$Length\ of\ chain, \boldsymbol{L} = \boldsymbol{2C} + \frac{\boldsymbol{Z_1 + Z_2}}{\boldsymbol{2}} + \frac{(\boldsymbol{Z_2 - Z_1})^2}{\boldsymbol{4\pi^2 C}}, \text{mm}$$

For the length of chain, L in pitch, the nominal center distance will be divided by pitch. Therefore, it can determine as:

$$Length\ of\ chain, \boldsymbol{L} = \frac{\boldsymbol{2C}}{\boldsymbol{p}} + \frac{\boldsymbol{Z_1 + Z_2}}{\boldsymbol{2}} + \frac{(\boldsymbol{Z_2 - Z_1})^2}{\boldsymbol{4\pi^2 \frac{C}{p}}}, \text{no of pitch}$$

Where. p = chain pitch, mm
Z_1 = number of drive sprocket teeth
Z_2 = number of driven sprocket teeth
C = nominal center distance (distance between the center of drive and driven sprocket), mm

Example 1:
A chain drive is used to transmit 10 hp of power to the driven sprocket to move a load. Input speed is 450 rpm and the output speed needed is 125 rpm. Given that the service factor of chain is 0.7 and the chain pitch is 7.5mm. The number of drive sprocket is 12 and nominal center distance is 300 mm. Determine:-

a) The design power
b) The correspondent number of teeth for driven sprocket
c) The pitch diameter for each sprocket
d) The velocity of the chain
e) The length of chain
f) The wrap angle for each sprocket

Solution:

a) $Design\ power, H_r = service\ factor, K_s\ x\ output\ power, H_{out}$
$= 0.7\ x\ 10$
$= \boldsymbol{7\ hp}$

b) $Z_1 = number\ of\ teeth\ for\ drive\ sprocket$
$Z_2 = number\ of\ teeth\ for\ driven\ sprocket$

$$ratio\ of\ speed: \frac{input\ speed}{output\ speed} = \frac{Z_2}{Z_1}$$

$$\frac{450}{125} = \frac{Z_2}{12}$$

$$Z_2 = 43.2$$

Thus, number of teeth for driven sprocket is **43**

c) $$\text{Pitch diameter for drive sprocket}, D_1 = \frac{\text{pitch}}{\sin(180/Z_1)}$$

$$= \frac{7.5}{\sin(\frac{180}{12})} = \mathbf{28.98}\boldsymbol{mm}$$

$$\text{Pitch diameter for driven sprocket}, D_2 = \frac{\text{pitch}}{\sin(180/Z_2)}$$

$$= \frac{7.5}{\sin(\frac{180}{43})} = \mathbf{102.75}\boldsymbol{mm}$$

d) $$V = r\omega$$

$$= \frac{2\pi N}{60}\ x\ \frac{D}{2}$$

$$= \frac{2\pi\ x\ 450}{60}\ x\ \frac{28.98}{2} = \mathbf{682.83}\ \boldsymbol{mm/s}$$

e) $$Length\ of\ chain, \boldsymbol{L} = \boldsymbol{2C} + \frac{\boldsymbol{Z_1 + Z_2}}{\boldsymbol{2}} + \frac{(\boldsymbol{Z_2 - Z_1})^2}{\boldsymbol{4\pi^2 C}}$$

$$= 2(300) + \frac{12 + 43}{2} + \frac{(43 - 12)^2}{4\pi^2(300)}$$

$$= \mathbf{627.5811}\ \boldsymbol{mm}$$

f) $$Warp\ angle\ for\ drive\ sprocket, \theta_1 = 180° - 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right]$$

$$= 180° - 2\sin^{-1}\left[\frac{102.75 - 28.98}{2(300)}\right]$$

$$= \mathbf{165.88°}$$

$$Warp\ angle\ for\ drive\ sprocket, \theta_2 = 180° + 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right]$$

$$= 180° + 2\sin^{-1}\left[\frac{102.75 - 28.98}{2(300)}\right]$$

$$= \mathbf{194.12°}$$

3.7 SELECTION OF ROLLER CHAIN DRIVES

The following data should be taken into consideration while selecting roller chain drives:

i. Horsepower to be transmitted
ii. RPM of the driving and driven sprocket (speed ratio)
iii. Load classification
iv. Space limitations if any
v. Driven machine

vi. Source of power

If the pitch centre distance and number of teeth on both driving and driven sprockets are known, you can use the following formula, tables and charts to calculate chain lengths.

3.7.1 SELECTION PROCEDURE

For maximum service life, smooth operation and optimum performance, the following points should be considered, while determining the number of teeth in the pinion.

a. As most drives have an even number of pitches in the chain, the use of a pinion with an odd number of teeth ensures even distribution of chain and wheel tooth wear.
b. Pinions for normal, stead drives should generally not have less than 17 teeth, the reason being that a chain forms a polygon around the pinion. When the pinion speed is constant, the chain speed is subject to regular cyclic variation. The percentage of cyclic variation becomes less marked as the number of teeth increases – and in fact becomes insignificant for the majority of applications when the number of teeth in the pinion exceeds 17.
c. A minimum of 23 teeth is recommended on moderate shock drives where the speed of the pinion exceeds 50 % of the maximum rated speed, and for heavy shock drives where the speed of the pinion exceeds 25% of the maximum rated speed.
d. The pinion should be heated to HV 10- 550 for smooth drives where the pinion speeds exceeds 70% of the maximum speed and operates under full horsepower rating. For heavy shock drives, the pinion be treated in all cases.

3.7.2 DETERMINE THE CLASS OF LOAD

If the shock loads are expected, then first determine the class of load on the basis of the drives equipment (refer table below)

Load Classifications

UNIFORM LOAD	MODERATE SHOCK LOAD	HEAVY SHOCK LOAD
Centrifugal pumps, Agitator for liquids, Conveyors, Fans-Uniform Load, Generators, Machines all types with uniform non-reversing loads	Reciprocating pumps, Wood working Machines Grinders, Conveyors– irregular Load, Mixers and Machines all types with moderate shock and non-reversing loads	Presses, Earth moving equipment Shears, Cranes & Hoists, Reciprocating and Shaker type conveyors, Crushers, Reciprocating feeders, Machines – all types with severe impact shock loads or variation and reversing service

TABLE 3. 1 : LOAD CLASSIFICATION

3.7.3 ESTABLISH THE DESIGN HORSEPOWER

Establish the design horsepower by multiplying the specified horsepower value with the service factor given in Table below:

SERVICE FACTOR, K_s			
Type of Driven Load	TYPE OF INPUT POWER		
	Internal Combustion Engine with Hydraulic Drive	Electric Motor or Turbine	Internal Combustion Engine with Mechanical Drive
Uniform	1.0	1.0	1.2
Moderate Shock	1.2	1.3	1.4
Heavy Shock	1.4	1.5	1.7

TABLE 3. 2 : SERVICE FACTOR, K_s

3.7.4 FACTOR OF CHAIN

a. Tooth Correction Factor, K_1

The use of a tooth factor further modifies the final power selection. The choice of a smaller diameter sprocket will reduce the maximum power capable of being transmitted since the load in the chain will be higher.

Z_1	K_1	Z_1	K_1
11	0.53	23	1.35
12	0.62	24	1.41
13	0.7	25	1.46
14	0.78	27	1.57
15	0.85	29	1.68
16	0.92	31	1.77
17	1	35	1.95
18	1.05	40	2.15
19	1.11	45	2.37
20	1.18	50	2.51
21	1.26	55	2.66
22	1.29	60	2.8

TABLE 3. 3 : TEETH CORRECTION FACTOR, K_1 BASED ON THE NUMBER OF TEETH IN THE DRIVE SPROCKET, Z_1

b. Multiple Strands Factor, K_2

Selection of multi – strand chains will become necessary if available space is limited or high speeds call for a chain with lower pitch. The strand factors are given in Table 3.4. To facilitate selection of multi – strand chains, multiply the horsepower rating for single strand chains by corresponding strand factor.

Multiple Strand Factor, K_2	
Number of strands	**Multiple Strand Factor, K_2**
1	1.0
2	1.7
3	2.5
4	3.3
5	3.9
6	4.6
8	6.2
10	7.5

TABLE 3. 4 : MULTIPLE STRAND FACTOR, K_2

3.7.5 CALCULATE THE SELECTION POWER

Multiply the power to be transmitted by the factors obtained.

Selection POWER, H_r = POWER to be transmitted x K_1 x K_2

Therefore, $$H_r = H_p \times K_1 \times K_2$$

This selection power can now be used with the given:

SPROCKET SPEED (RPM)	ANSI CHAIN NUMBER						
	25	**35**	**40**	**41**	**50**	**60**	**80**
50	0.05	0.16	0.37	0.20	0.72	1.24	2.88
100	0.09	0.29	0.69	0.38	1.34	2.31	5.38
150	0.13	0.41	0.99	0.55	1.92	3.32	7.75
200	0.16	0.54	1.29	0.71	2.50	4.30	10.00
300	0.23	0.78	1.85	1.02	3.61	6.20	14.50
400	0.30	1.01	2.40	1.32	4.67	8.03	18.70
500	0.37	1.24	2.93	1.61	5.71	9.81	22.90
600	0.44	1.46	3.45	1.90	6.72	11.60	27.00
700	0.50	1.68	3.97	2.18	7.73	13.30	31.00
800	0.56	1.89	4.48	2.46	8.71	15.00	35.00
900	0.62	2.10	4.98	2.74	9.69	16.70	39.90
1000	0.68	2.31	5.48	3.01	10.70	18.30	37.70
1200	0.81	2.73	6.45	3.29	12.60	21.60	28.70
1400	0.93	3.13	7.41	2.61	14.40	18.10	22.70
1600	1.05	3.53	8.36	2.14	12.80	14.80	18.60
1800	1.16	3.93	8.96	1.79	10.70	12.40	15.60
2000	1.27	4.32	7.72	1.52	9.23	10.60	13.30
2500	1.56	5.28	5.51	1.10	6.58	7.57	9.56
3000	1.84	5.64	4.17	0.83	4.98	5.76	7.25

SPROCKET SPEED (RPM)	ANSI CHAIN NUMBER						
	100	120	140	160	180	200	240
50	5.52	9.33	14.40	20.90	28.90	38.40	61.80
100	10.30	17.40	26.90	39.10	54.00	71.60	115.00
150	14.80	25.10	38.80	56.30	77.70	103.00	166.00
200	19.20	32.50	50.30	72.90	101.00	134.00	215.00
300	27.70	46.80	72.40	105.00	145.00	193.00	310.00
400	35.90	60.60	93.80	136.00	188.00	249.00	359.00
500	43.90	74.10	115.00	166.00	204.00	222.00	0.00
600	51.70	87.30	127.00	141.00	155.00	169.00	
700	59.40	89.00	101.00	112.00	123.00	0.00	
800	63.00	72.80	82.40	91.70	101.00		
900	52.80	61.00	69.10	76.80	84.40		
1000	45.00	52.10	59.00	65.60	72.10		
1200	34.30	39.60	44.90	49.90	0.00		
1400	27.20	31.50	35.60	0.00			
1600	22.30	25.80	0.00				
1800	18.70	21.60					
2000	15.90	0.00					
2500	0.40						
3000	0.00						

TABLE 3.5 : THE RATED HORSEPOWER, H_P REFER TO DRIVING SPROCKET SPEED (RPM)

ANSI Chain Number	Pitch (mm)	Width (mm)	Roller Diameter (mm)
25	6.35	3.18	3.30
35	9.52	4.76	5.08
41	12.70	6.35	7.77
40	12.70	7.94	7.92
50	15.88	9.52	10.16
60	19.05	12.70	11.91
80	25.40	15.88	15.87
100	31.75	19.05	19.05
120	38.10	25.40	22.22
140	44.45	25.40	25.40
160	50.80	31.75	28.57
180	57.15	35.71	35.71
200	63.50	38.10	39.67
240	76.70	47.63	47.62

TABLE 3. 6 : CHAIN SPECIFICATION REFER TO CHAIN NUMBER

ISO Chain Number	Pitch, P (mm)	Roller Diameter ϕ_1, (mm)	Width B_1, (mm)	Transverse Pitch P1 (mm)	Breaking Load For Single Strand Chain (kN)
06 B	9.525	6.35	5.72	10.24	10.7
08 B	12.70	8.51	7.75	13.92	18.2
10 B	15.875	10.16	9.65	16.59	22.7
12 B	19.05	12.07	11.68	19.46	29.5
16 B	25.40	15.88	17.02	31.88	65.0
20 B	31.75	19.05	19.56	36.45	98.1
24 B	38.10	25.40	25.40	48.36	108.9
28 B	44.45	27.94	30.99	59.56	131.5
32 B	50.80	29.21	30.99	58..55	172.4
40 B	63.50	38.10	38.10	72.29	272.2

TABLE 3.7: CHAIN SPECIFICATION REFER DIMENSION AND BREAKING LOADS OF ROLLER CHAINS (ISO CHAIN TABLE)

Pinion Speed (rpm)	Power (kW)				
	06 B	08 B	10 B	12 B	16 B
50	0.14	0.34	0.64	1.07	2.59
100	0.25	0.64	1.18	2.01	4.83
200	0.47	1.18	2.19	3.75	8.94
300	0.61	1.70	3.15	5.43	13.06
500	1.09	2.72	5.01	8.53	20.57
700	1.48	3.66	6.71	11.63	27.73
1000	2.03	5.09	8.97	15.65	34.89
1400	2.73	6.81	11.67	18.15	38.47
1800	3.44	8.10	13.03	19.85	-
2000	3.80	8.67	13.49	20.57	-

TABLE 3. 8: CHAIN SPECIFICATION REFER POWER RATING FOR SIMPLE ROLLER CHAINS (ISO CHAIN TABLE)

3.7.6 FINAL SELECTION OF CHAIN

From the rating chart, select the smallest pitch of simple chain to transmit the selection power at the speed of the driving sprocket N_1.This normally results in the most economical drive selection. If the selection power is now greater than that shown for the simple chain, then consider a multiplex chain of the same pitch size as detailed in the rating charts.

Example 2:
A roller chains used to transmit 90 hp from a 17 teeth sprocket to a driven sprocket at a speed 0f 300 rpm. The operation involve heavy shocks internal combustion engine with mechanical drive. The center distance of sprocket is about 35 pitch and velocity ratio 3: 1. Find:

a) The correspondent number of teeth for driven sprocket
b) Design horse power
c) Chain number and number of strand roller chain
d) The length of roller chain in number of pitch

Solution:

a) $N_1 = number\ of\ teeth\ for\ drive\ sprocket$

$N_2 = number\ of\ teeth\ for\ driven\ sprocket$

$$ratio\ of\ speed : \frac{input\ speed}{output\ speed} = \frac{N_2}{N_1}$$

$$\frac{3}{1} = \frac{N_2}{17}$$

$$N_2 = 51$$

Thus, number of teeth for driven sprocket is **51.**

b) For heavy shocks internal combustion engine with mechanical drive, refer table 3.2:
$K_s = 1.7$

So, Design horse power, Hr = Power x K_S
= 90 hp x 1.7
= 153 hp.

c) Given $Z_1 = 17$, so $K_1 = 1.0$ (refer table 3.3)
$H_r = H_p x K_1 x K_2$
$153 = H_p x 1.0 x K_2$

Try with the highest number of strands given (refer table 3.4)

For 10 number of strands, $K_2 = 7.5$
$153 = H_p x 1.0 x 7.5$
$H_p = 20.4$ hp.

By refer table 3.5, for 300 rpm sprocket speed, the nearest and greater than 20.4 hp is 27.70 hp.

Therefore, the suitable chain number is 100.

$H_r = H_p \times K_1 \times K_2$

For $K_2 = 7.5$ (10 strands), $H_r = 27.7 \times 1 \times 7.5$
$= 207.75$ hp

For $K_2 = 6.2$ (8 strands), $H_r = 27.7 \times 1 \times 6.2$
$= 166.2$ hp

For $K_2 = 4.6$ (6 strands), $H_r = 27.7 \times 1 \times 4.6$
$= 127.42$ hp

So the nearest and greater than 153 horsepower is 166.2 hp

Therefore the suitable chain specification used is Chain Number 100, **8 strands.**

d) $$Length\ of\ chain, \boldsymbol{L} = \boldsymbol{2C} + \frac{\boldsymbol{Z_1 + Z_2}}{\boldsymbol{2}} + \frac{(\boldsymbol{Z_2 - Z_1})^2}{\boldsymbol{4\pi^2 C}}$$

$$L = 2(35) + \frac{17 + 51}{2} + \frac{(51 - 17)^2}{4\pi^2(35)}$$

= **104.84 pitch.**

1. State the definition of chain pitch
2. State **FOUR (4)** advantages of chain drive in power transmission mechanism
3. A chain drive is used to transmit 10 hp of power to the driven sprocket to move a load. Input speed is 450 rpm and the output speed needed is 125 rpm. Given that the service factor of chain is 0.7 and the chain pitch is 7.5mm. The number of drive sprocket is 12 and nominal center distance is 300 mm. Determine:-
   a. The design power
   b. The correspondent number of teeth for driven sprocket
   c. The pitch diameter for each sprocket
   d. The velocity of the chain
   e. The length of chain
   f. The warp angle for each sprocket

4. A chain drive is used to connect a 10 kW, 1400 rpm electric motor to centrifugal pump running at 720 rpm. The operation involves moderate shocks. In order to reduce the polygonal effect, a driving sprocket with 17 teeth is used. Refer to the ISO chain table attached.
   a. Calculate the power rating of the chain
   b. Select the suitable roller chain to be used for the operation
   c. Refer to table given. State the actual specification of a chain
   d. Determine the pitch circle diameters of driving and driven sprocket

Previous Years' Question Papers

JUNE 17 & DEC 17

Q2

a. A chain drive is used to transmit power between a motor with the speed of 1000 rpm. The necessary velocity of the load is between 240 to 250 rpm and the power that is needed to drive is 13 kW. Given the safety factor is 1.4, pitch is 15 mm, the number of teeth of drive sprocket is 20 and the nominal center distance is 600 mm.
   Calculate:-
   i. The actual power
   ii. The correspondence number of teeth for driven sprocket. 7M

b. Analyse the reason of chain transmission used in a bicycle (not gear or belt mechanism) 6M

DEC 16

Q3

a. List 4 advantages and 4 disadvantages of chain drive in power transmission. 8M
a. Explain the procedure of roller chain drive installation. 7M

DEC 15

Q3

a. Explain 5 advantages of chain drives compared to gear drives. 10M

b. It is required to design a chain drive to connect a 15kW, 1800rpm electric motor to a centrifugal pump running at 800rpm. The service conditions involve moderate shocks. In order to reduce the polygonal effect, a driving sprocket is used with 17 teeth (Refer ISO chain table attached).

   i. Select a proper roller chain. 4M
   ii. List its dimension. 3M
   iii. Calculate the pitch circle diameters of the driving and driven sprockets. 4M
   iv. Calculate the length of chain by assuming the centre distance between the sprocket wheel as 50p (50 x pitch). 4M

JUNE 15

Q3

a. State the definition of chain drive. 3M

b. A chain drives to transmit power 30hp of driven sprocket to move a load. Input speed is 850rpm and output speed need is 200rpm. Given that the service factor of chain is 1.4 and the pitch is 15mm. The number of teeth of drive sprocket is 23 and nominal center distance is 700mm, determine:

   i. The design power.
   ii. Number of teeth on the sprocket
   iii. The pitch diameter for each sprocket
   iv. Chain length
   v. The wrap angle for driven sprocket 22M

DEC 14

Q3

a. Define chain pitch. 2M

b. State 4 advantages and 4 disadvantages of chain drive. 8M

c. A chain drive is used to transmit power between a motor with a speed of 1000rpm and a load. The necessary velocity of the load is between 240 to 250rpm and the power that needed to drive it is 13kW. Given that safety factor is 1.4, pitch is 15mm, the number of teeth of drive sprocket is 20 and the nominal center distance is 600mm. Determine:

   i. The actual power
   ii. The correspondent number of teeth for driven sprocket.
   iii. The pitch diameter for each sprocket
   iv. The length of chain
   v. The warp angle for each sprocket 10M

CHAPTER 4: SHAFTS AND AXLES

4

OBJECTIVES:

In the end of this chapter, student should know to:

- Explain the principle of shaft in power transmission system.
- Explain principle of bearing.

4.1 INTRODUCTION

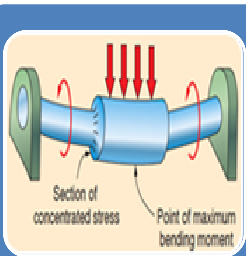
This topic introduces the benefits of shaft in power transfer and its use in key shaft. Emphasis is given on the methods of choosing bearings based on a bearing catalogue. It also includes inspection and care of the devices.

4.2 DEFINITION

Axles are classified as either live or dead. The live axle is used to transmit power. The dead axle serves only as a support for part of the vehicle while providing a mounting for the wheel assembly. Many commercial trucks and truck-tractors have dead axles on the front, whereas practically all passenger vehicles use independent front-wheel suspensions and have no front axles. Axles are connected within vehicles to perform two important functions:

i) they transmit torque from variance to wheel through planetary gear arrangement, and
ii) they maintain the position of the wheels comparative to each other and to the body of the vehicle.

The shafts are installed in the tire's wheel well near the differentials and stretch across the bottom of the vehicle. Often during operation, the axle shafts are subjected to heavy torque due to loads or sudden acceleration and therefore, they are manufactured from different grades of hardened steels.



Shaft

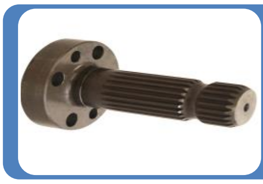
- **A long cylindrical device such as a rod or pole. On a wheel, the shaft extends from the centre of the wheel along its axis.**
- **Used to connect other component of a drive unit train, can't be connected directly.**
- **Shafts are carries of torque.**
- **Subject to torsion and shear stress.**

Axle

- An axle is a central shaft supports gear, wheel, rotor etc. subjected to torsion.
- 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.
- Bearings or bushings are provided at the mounting points where the axle is supported.

Spindle

- A short shaft, usually of small diameter, usually rotating, e.g. valve spindle for gate valve, but consider also the HEADSTOCK SPINDLE of a lathe, which is quite large and usually has a hole right through its centre



Stub shaft

- A shaft which is integral with an engine, motor or prime mover and is of suitable size, shape and projection to allow its easy connection to other shafts.

Line shaft

- A shaft connected to a prime mover which transmits power to a number of machines – now
mostly superseded by machines having individual motors.

Jack shaft

- A short shaft used to connect a prime mover to a machine or another shaft. May also be a short shaft placed as an
intermediate shaft between a prime mover and driven machine.

Flexible shaft

- Permits the transmission of power between two shafts (e.g. motor shaft and machine shaft) whose rotational axes are at an angle or where the angle between the shafts may change.

4.3 **KEYS**

KEY is a metal plate used to connect the **shaft and pulley** or **shaft and gear** so that both can rotate with the **same rotational speed** for allow power transmitted.

Perhaps the most widely used method of torque transfer is by the use of keys. Key **FUNCTION** is used

i) To transmit the torque from shaft to hub of the mating element and vice versa
ii) on shafts to secure the rotating gears
iii) prevent relative movement between one shaft and another component part that attach to it whereas it could support torque load.
iv) To prevent axial motion between two elements except in case of feather key or splined connection.

v) The key is plugged in by inserting into the keyway cut parallel to the shaft and hub pulley/gear.
vi) The key fits simultaneously into both grooves, locking them together.
vii) The key is subjected to direct shear.

Types of key

- Sunk keys
  - Rectangular/Flat sunk key
  - Square sunk key
  - Parallel sunk key
  - Gib-head key
  - Feather key
  - Woodruff key
- Saddle keys
  - Flat saddle key
  - Hollow saddle key
- Tangent keys
- Round/Dowel keys
- Splines

FIGURE 4.2: TYPES OF KEYS

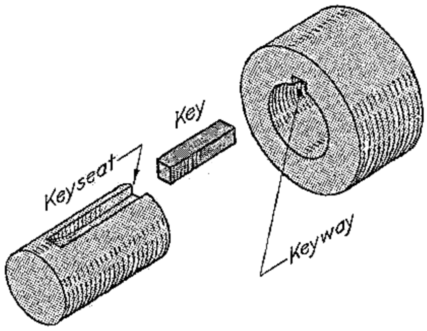


FIGURE 4.1: MECHANICAL SHAFT KEY

Advantages of key

- prevents relative rotation between the two parts and may enable torque transmission
- reduce maintenance cost

Rectangular sunk keys

- Rectangular sunk keys are shown in figure4.3. They are the simplest form of machine keys and may be either straight or slightly tapered on one side. The parallel side is usually fitted into the shaft.

Gib head sunk keys

- The gib head keys are ordinary sunk keys tapered on top with a raised head on one side so that its removal is easy

Feather keys

- A feather key is used when one component slides over another. The key may be fastened either to the hub or the shaft and the keyway usually has a sliding fit.

Woodruff keys

- A woodruff key is a form of sunk key where the key shape is that of a truncated disc. It is usually used for shafts less than about 60 mm diameter and the keyway is cut in the shaft using a milling cutter . It is widely used in machine tools and automobiles due to the extra advantage derived from the extra depth.

Flat key

- A flat key is used for light load because they depend entirely on friction for the grip. The sides of these keys are parallel but the top is slightly tapered for a tight fit. Theses keys have about half the thickness of sunk keys.

Saddle key

- A saddle key, is very similar to a flat key except that the bottom side is concave to fit the shaft surface. These keys also have friction grip and therefore cannot be used for heavy loads. A simple pin can be used as a key to transmit large torques. Very little stress concentration occurs in the shaft in these cases.

Tangent Key

- A tangent keys are fitted in paie at right angles. Each key is to withstand torsion in one direction only. These are used in large heavy duty shafts. If cross section of tangent is square, it is also called Kennedy Key.

Spline shaft and hub

- A spline shaft is used when the hub is required to slide along the shaft. These shafts are used mostly for sliding gear application as in automotive gear box and propeller shaft of aircraft.

FIGURE 4.3: TYPE OF KEYS AND DESCRIPTIONS

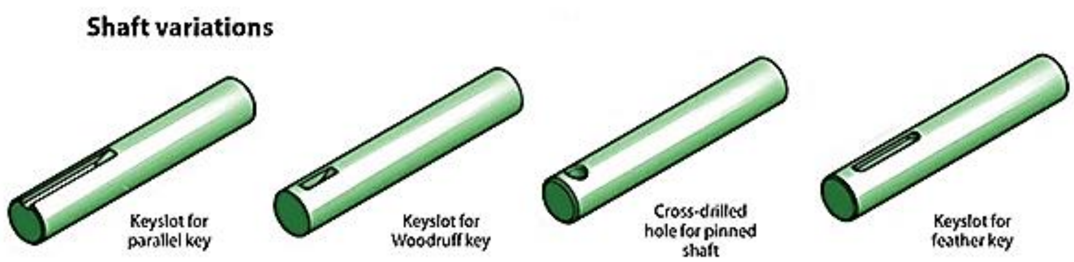


FIGURE 4.4: KEYSLOT FOR DIFFERENT TYPES OF KEYS

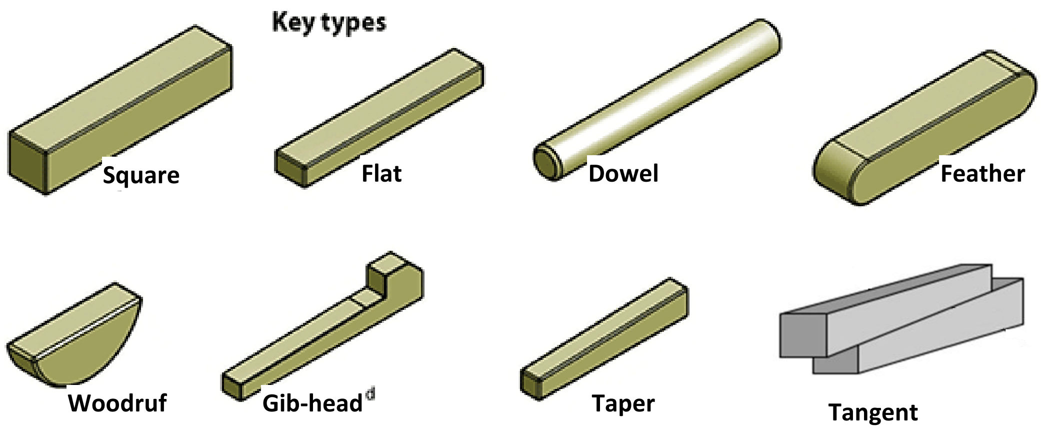
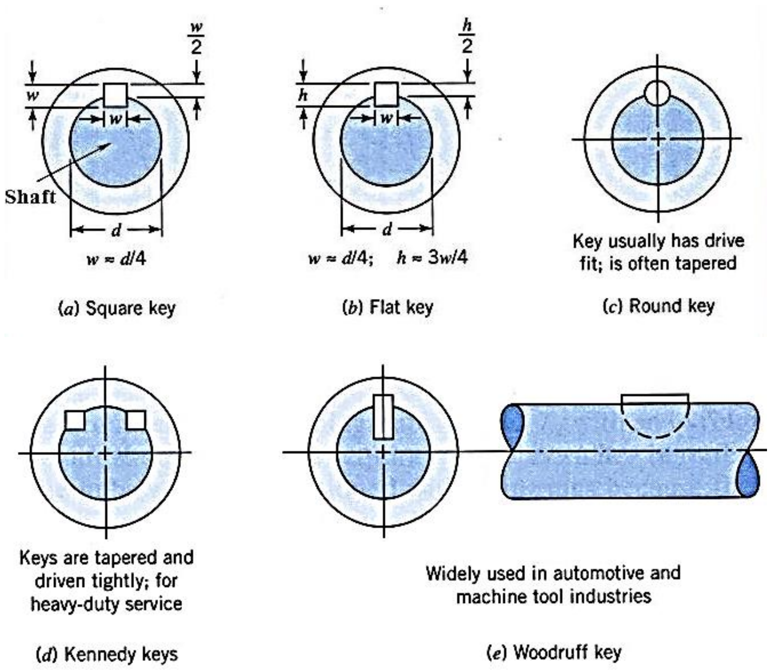
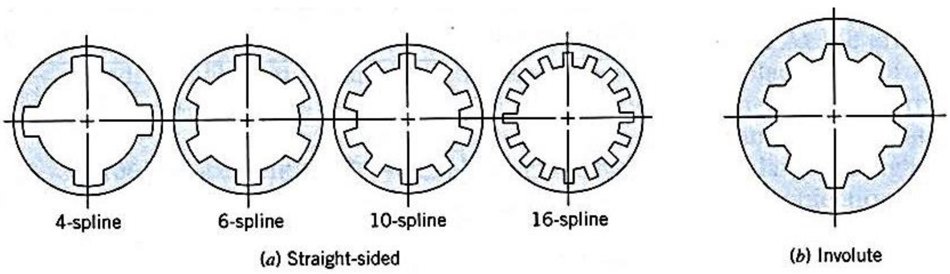


FIGURE 4.5: SKETCH OF BASIC KEYS



Splines may be considered as using multiple keys, integral with the shaft. Best way to transmit large torque.



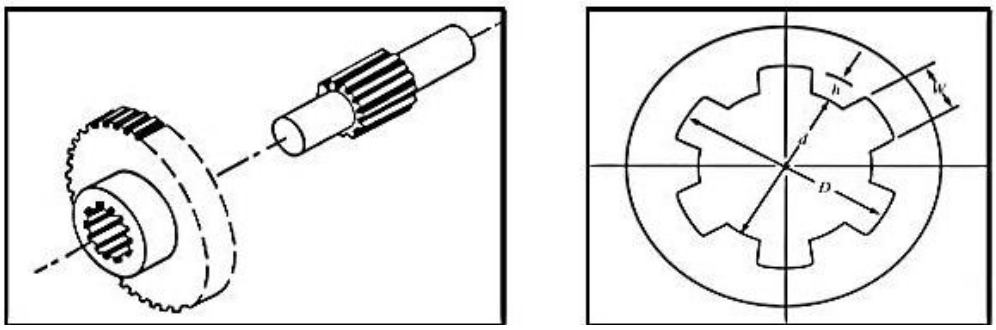


FIGURE 4.6: SPLINE KEY

Key and keyways are one of the most important techniques used for the coupling purpose. These are commonly used in shaft-hub connections. Despite knowing the importance of this a very little research work has been reported in this field. The failure of key and keyways occurred due to the stress concentration in certain areas of machine element. Following are the solutions that can be taken to solve the problem:

1. Increase fillet radius
2. Add additional notches

3. Cutting additional holes
4. Suitable liquid coding

4.3.1 Stress And Shear Analysis Of Sunk Keys

When a key is used in transmitting torque from a shaft to a rotor or hub, the following two types of forces act on the key:

a. Forces,F_1 due to fit of the key in its keyway, as in a tight fitting straight key or in a tapered key driven in place. These forces produce compressive/crushing stresses in the key which are difficult to determine in magnitude.
b. Forces,F due to the torque transmitted by the shaft. These forces produce shearing stresses in the key

A key has two failure mechanisms:

1. it can be sheared off, and
2. it can be crushed due to the compressive bearing forces.

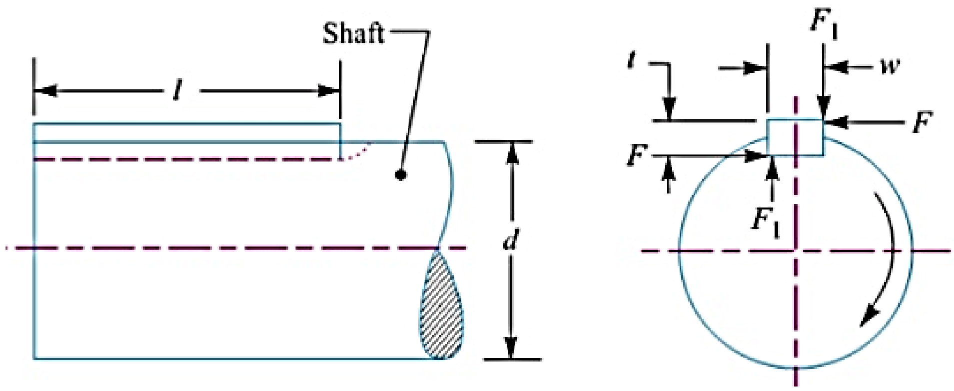


FIGURE 4.7: STRESS ANALYSIS OF SUNK KEYS

4.3.2 Strength Of A Sunk Key

A key connecting the shaft and hub is shown in Figure 4.6.

Let T = Torque transmitted by the shaft, Nmm

F = Tangential force acting at the circumference of the shaft, N

d = Diameter of shaft, mm

l = Length of key, mm

w = Width of key, mm

t = Thickness of key, mm

τ_S = Shear stresses for the material of key, N/mm^2

σ_c = Compressive/crushing stresses for the material of key, N/mm^2

A little consideration will show that due to the power transmitted by the shaft, the key may fail due to shearing or crushing. Considering shearing of the key, the tangential shearing force acting at the circumference of the shaft, F = Area resisting shearing × Shearing stress = l × w × τ

$$\boxed{F = l\, w\, \tau}$$

∴ Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = F \times r \quad \text{, thus}$$

$$\boxed{T = l\, w\, \tau\, r}$$

Considering crushing of the key, the tangential crushing force acting at the circumference of the shaft,

F = Area resisting crushing × Crushing stress = $l \times \frac{t}{2} \times \sigma_c$

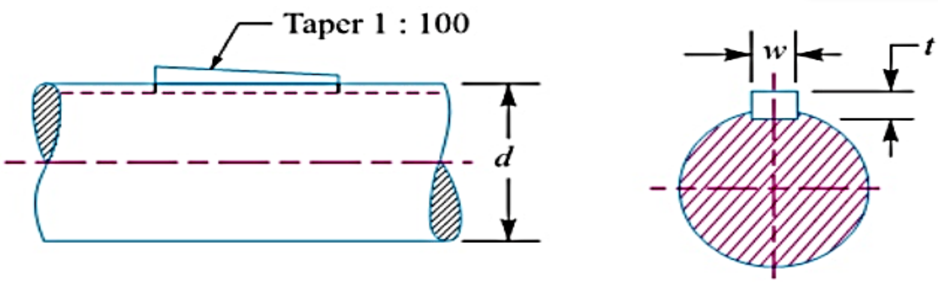
∴ Torque transmitted by the shaft,

$$T = F \times r = l \times \frac{t}{2} \times \sigma_c \times r$$

The key is equally strong in shearing and crushing, if

$$l.w.\tau.\ r = l\ .\ \frac{t}{2}\ .\ \sigma_c\ .\ r$$

$$\boxed{T = \sigma_c \frac{t}{2}\, l\, r} \quad \text{or} \quad \boxed{\frac{w}{t} = \frac{\sigma_c}{2\tau}}$$



The usual proportions of this key are,

w=d/4 **t=2w/3**

4.3.3 Length of The Key

In order to find the length of the key to transmit full power of the shaft,

- The Shearing strength of key is equal to the torsional shear strength of the shaft

Shearing strength of key,

$$\boxed{T = l\, w\, \tau_K\, r}$$

torsional shear strength of the shaft,

$$\boxed{T = \frac{\pi}{16} \tau_S\, d^3}$$

$$l\ w\ \tau_K\ r = \frac{\pi}{16}\tau_S\ d^3$$

when the key **material is same as that of the shaft**, then $\tau_K = \tau_S$

$$l = 1.571d$$

where,

τ_K = shear stress of key, N/mm^2

τ_S = shear stress of shaft, N/mm^2

Example 1:

A 45 mm diameter shaft is made of steel with a yield strength of 400 MPa. A parallel key of size 14 mm wide and 9 mm thick made of steel with a yield strength of 340 Mpa is to be used. Find the required length of key, if the shaft is loaded to transmit the maximum permissible torque. Use maximum shear stress theory and assume a factor of safety of 2.

Solution:

Given,
d = 45mm,
r=22.5mm
w = 14mm
t = 9mm
σ_S = 400MPa
σ_K = 340MPa
l = ?
SF = 2

According to maximum shear stress theory:
The maximum shear stress for the shaft

$$\tau_{max} = \frac{\sigma_S}{2(SF)} = \frac{400}{2(2)} = 100\text{N/mm}^2$$

and maximum shear stress for the key,

$$\tau_K = \frac{\sigma_K}{2(SF)} = \frac{340}{2(2)} = 85\text{N/mm}^2$$

We know that the maximum torque transmitted by the shaft and key,

$$T = \frac{\pi}{16}\tau_S\ d^3 = \frac{\pi}{16}(100)(45)^3 = 1.789\ \text{MN/mm}^2$$

First of all, let us consider the failure of key due to shearing.

We know that the maximum torque, T transmitted,

$$1.789\ M = l\ w\ \tau_K\ r$$

$$1.789M = l\ (14)(85)(22.5)$$

$$\therefore\ \underline{l = 66.82\text{mm}}\ \#$$

Now considering the failure of key due to crushing, We know that the maximum torque transmitted by the shaft and key,

$$1.789M = \sigma_c\ \frac{t}{2}\ l\ r$$

$$\sigma_c = \frac{\sigma K}{SF} = \frac{340}{2} = 170\text{MPa}$$

$$1.789M = (170)(\tfrac{9}{2})\,(l)(22.5)$$

$\therefore$ <u>l = 103.94mm</u> #

Taking the larger of the two values, we have

<u>l = 103.94mm ≈104mm</u> #

Example 2:

A motor transmit 15 kW at 960 rpm has a mild steel shaft. The shaft diameter is 40mm. The shaft and key made from same material with allowable shear stress 58 N/mm^2 and allowable stress 112 N/mm^2. Find other key dimensions if,

a) w = d/4 and t = 2/3 W

b) The motor attached by gear has 30 mm hub.

Solution:

Given,
Pow = 15kW
N = 960rpm
d = 40mm,
r=20mm
τ =58N/mm^2
σ = 112N/mm^2

a) l when, w = d/4 and t = 2/3 W,

$w = \frac{40}{4} = 10\text{mm}$

$t = \frac{2}{3}(10) = 6.667\text{mm}$

we know, $T = l\, w\, \tau_K\, r$ and

$T = \sigma c\, \frac{t}{2}\, l\, r$

$\omega = \frac{2\pi N}{60} = \frac{2\pi(960)}{60} = 100.531$ rad/s

Pow = Tω, $\therefore$ 15k =T (100.531)

Thus, T = 149.208 Nm =149.209 kNmm

$\therefore$149.208k = l_1 (10)(58)(20)

$\therefore$ <u>L_1 = 12.863mm</u>#

$\therefore$149.208k = (112) $\left(\frac{6.667}{2}\right)$ l_2 (20)

$\therefore$ <u>L_2 = 19.982mm</u>#

b) w and t, when l = 30mm

$T = \sigma c\, \frac{t}{2}\, l\, r$

$\therefore$149.208k = (112) $\left(\frac{t}{2}\right)$(30)(20)

$\therefore$ <u>t = 4.441mm</u>#

$T = l\, w\, \tau_K\, r$

$\therefore$149.208 = (30)(w)(58)(20)

$\therefore$ <u>w = 4.29 x 10^{-3}mm</u>#

4.4 **BEARING**

Bearings are machine elements which are used to support a rotating member called as shaft. It transmits the load from a rotating member to a stationary member known as frame or housing.

4.4.1 CLASIFICATION OF BEARING

Bearing are classified depending upon the load. Bearing are also classified depending upon the type of contact.

i) Sliding contact bearing

- journal bearing
- plane bearing

ii) Antifriction bearing

- ball bearing
- roller bearing

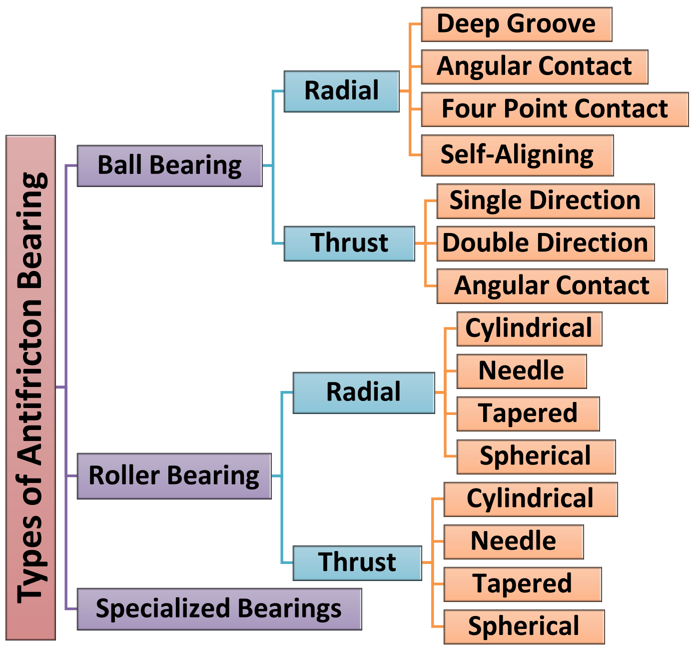


FIGURE 4.8: TYPES OF ANTI-FRICTION BEARING

4.4.2 FUNCTION OF BEARING

- A bearing permits relative motion between two machine members while minimizing frictional resistance.
- A bearing consists of an inner and outer member separated either by a thin film of lubricant or a rolling element.
- A bearing bears the load.
- It locates the moving parts in correct position.
- It provides free motion to the moving part by reducing friction.

4.4.3 TERMINOLOGY

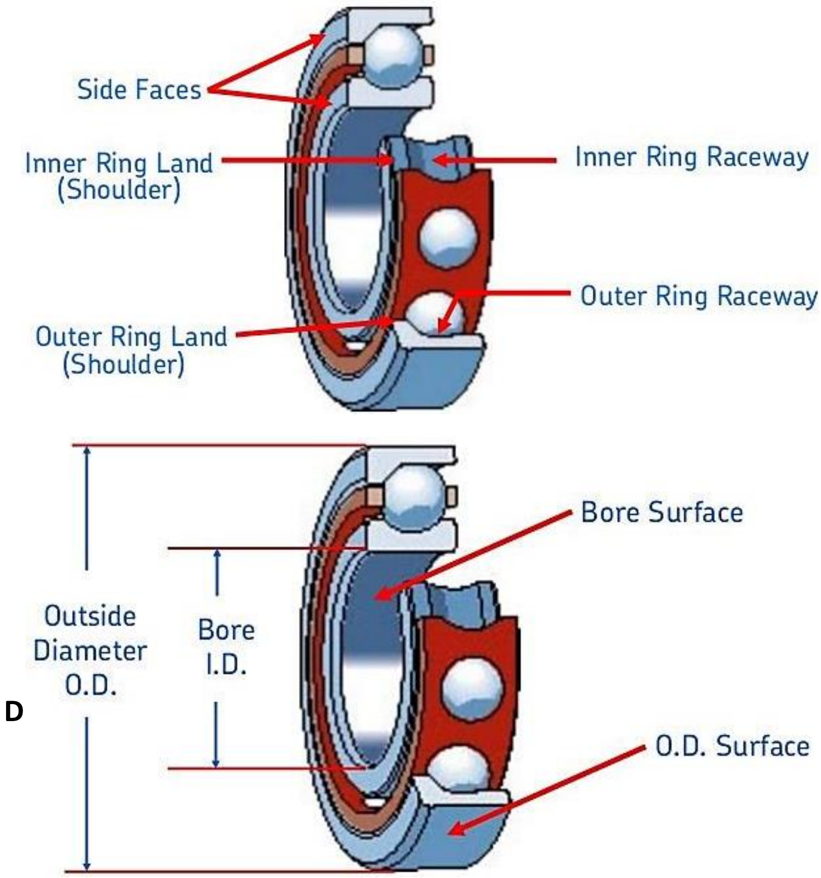


FIGURE 4.9: BALL BEARING TERMINOLOGY

Bearings types / Characteristics	Deep groove ball bearings	Angular contact ball bearings	Cylindrical roller bearings	Needle roller bearings	Tapered roller bearings	Self-aligning roller bearings	Thrust ball bearings
Load carrying capacity (Radial load, Axial load)			●				
High speed rotation ●	☆☆☆☆	☆☆☆☆	☆☆☆☆	☆☆☆	☆☆☆	☆☆	☆
Low noise/vibration ●	☆☆☆☆	☆☆☆	☆	☆			☆
Low friction torque ●	☆☆☆☆	☆☆☆	☆				
High rigidity ●			☆☆	☆☆	☆☆	☆☆☆	
Allowable misalignment for inner/outer rings ●	☆					☆☆☆	★
Non-separable or separable ●			○	○	○		○

❶ ☆The number of stars indicates the degree to which that bearing type displays that particular characteristic.
★Not applicable to that bearing type.
❷ ○Indicates both inner ring and outer ring are detachable.
❸ Some cylindrical roller bearings with rib can bear an axial load.

TABLE 4.1 : TYPES AND PERFORMANCE OF ROLLING BEARINGS

Ball bearings
- Ball bearings are extremely common because they can handle both radial and axial/thrust loads, but can only handle a small amount of weight.
- They are found in a wide array Of applications, such as roller blades and even hard drives.

Roller Bearings
- Roller bearings are designed to carry heavy loads - the primary roller is a cylinder, which means the load is distributed over a larger area, enabling the bearing to handle larger amounts of weight.
- This structure, however, means the can handle primarily radial loads, but is not suited to thrust loads.

Roller Thrust Bearings
- Roller thrust bearings, much like ball thrust bearings, handle thrust loads.
- The difference, however, lies in the amount of weight the bearing can handle: roller thrust bearings can support significantly large

Specialized Bearings
- There are, of course, several kinds of bearings that are manufactured for specific applications, such as magnetic bearings and giant roller bearings.
- Magnetic bearings are found in high-speed devices no moving parts - this stability enables it to support devices that move unconscionably fast.

FIGURE 4.10: BEARING TYPES AND DESCRIPTIONS

4.4.4 BEARING LIFE

Bearing life is defined as the number of rotations or the number of hours at a fixed speed in which bearings can withstand before failure occurs. The service life of the rolling bearings or balls is based on the basic equation:

where,

L_H = life in working hours, hrs

C@C_{10} = basic dynamic load rating, N

P = equivalent radial load (dynamic), N

a = 3 for ball bearings

= 10/3 for roller bearings

N = *speed, rpm*

$$L_H = \frac{1x10^6}{60\ N}\ \frac{C}{P}^{\ a}$$

4.4.5 Bearing Load : Radial Load and Axial/ Thrust Load.

When both dynamic radial loads and dynamic axial loads act on a bearing at the same time, the hypothetical load acting on the center of the bearing which gives the bearings the same life as if they had only a radial load or only an axial load is called the dynamic equivalent load. For radial bearings, this load is expressed as pure radial load and is called the dynamic equivalent radial load. For thrust bearings, it is expressed as pure axial load, and is called the dynamic equivalent axial load.

Consider,

F_a = axial@thrust load,N

F_r = radial load, N

P = equivalent radial load (radial + polarization), N

The relationship between F_a, F_r and P is given as;

$$\mathbf{P} = X_i F_r + Y_i F_a$$

Where,

i=1 if $\frac{F_a}{F_r} \leq e$ and i=2 if $\frac{F_a}{F_r} > e$

Equivalent Dynamic Bearing Load, P is defined as a fixed radial load or fixed axial load which if applied, would affect bearing life is just like an actual load and rotation.

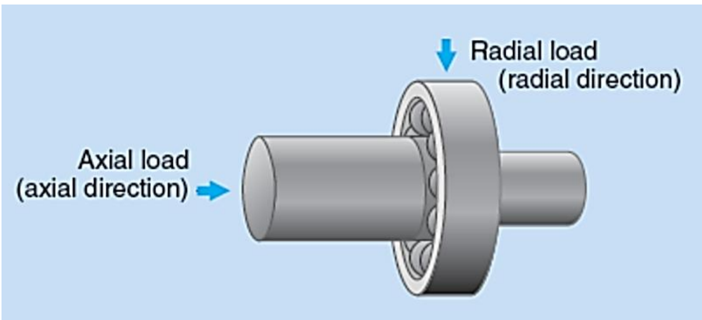


FIGURE 4.11: LOAD ON BEARING

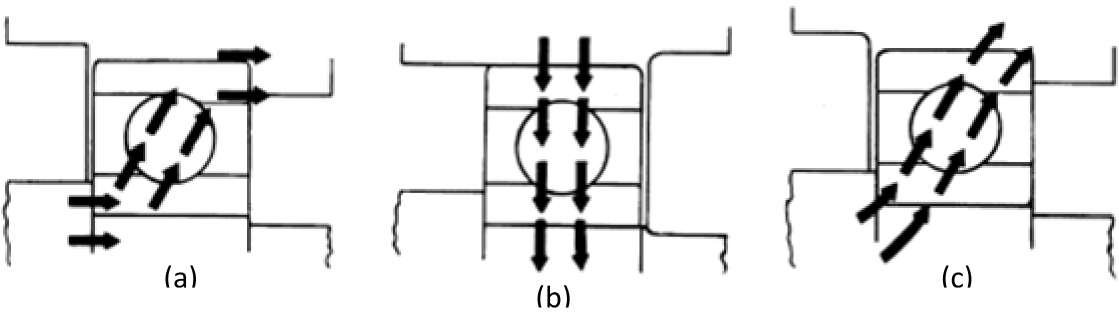


FIGURE 4.12: (A) THRUST LOAD, (B) AXIAL LOAD, (C) ANGULAR LOAD

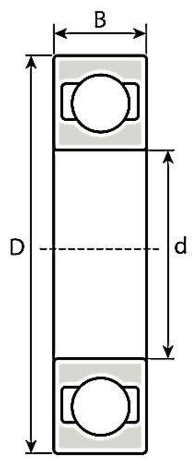
$\frac{F_a}{C_O}$	e	$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$	
		X_1	Y_1	X_2	Y_2
0.014	0.19	1	0	0.56	2.3
0.021	0.21				2.15
0.028	0.22				1.99
0.042	0.24				1.85
0.056	0.26				1.71
0.070	0.27				1.63
0.084	0.28				1.55
0.110	0.30				1.45
0.170	0.34				1.31
0.280	0.38				1.15
0.420	0.42				1.04
0.560	0.44				1.00

*Use 0.014 if $F_a/C_O < 0.014$

TABLE 4.2: VALUES OF X_I AND Y_I FOR RADIAL BEARINGS.

SINGLE ROW DEEP GROOVE BALL BEARINGS

Principal.dimensions.			Basic.load.ratings.		Designation Number
			dynamic	*static*	
d	D	B	C	C_0	
mm			kN		
40	52	7	4.94	3,45	61808
	62	12	13.8	10	61908
	68	9	13.8	9,15	16008
	68	15	17.8	11,6	6008
	80	18	32.5	19	6208
	90	23	42.3	24	6308
	110	27	63.7	36,5	6408
45	58	7	6.63	6,1	61809
	68	12	14	10,8	61909
	75	10	16.5	10,8	16009
	75	16	22.1	14,6	6009
	85	19	35.1	21,6	6209
	100	25	55.3	31,5	6309



Principal.dimensions.			Basic.load.ratings.		Designation Number	Principal.dimensions.			Basic.load.ratings.		Designation Number
d	D	B	dynamic	static		d	D	B	dynamic	static	
			C	C_0					C	C_0	
mm			kN			mm			kN		
50	120	29	76.1	45	6409		110	12	28.6	27	16115
	72	12	14.6	11,8	61910		115	13	30.2	27	16015
	80	10	16.8	11,4	16010		115	20	41.6	33,5	6015
	80	16	22.9	16	6010		130	25	68.9	49	6215
	90	20	37.1	23,2	6210		160	37	119	76.5	6315
	110	27	65	38	6310	80	190	45	153	114	6415
55	130	31	87.1	52	6410		100	10	13	15	61816
	72	9	9.04	8,8	61811		110	16	25.1	20,4	61916
	80	13	16.5	14	61911		125	14	35.1	31.5	16016
	90	11	20.3	14	16011		125	22	49.4	40	6016
	90	18	29.6	21,2	6011		140	26	72.8	55	6216
	100	21	46.2	29	6211		170	39	130	86.5	6316
	120	29	74.1	45	6311	85	200	48	163	125	6416
60	140	33	99.5	62	6411		110	13	19.5	20.8	61817
	78	10	11.9	11.4	61812		120	18	31.9	30	61917
	85	13	16.5	14,3	61912		130	14	35.8	33.5	16017
	95	11	20.8	15	16012		130	22	52	43	6017
	95	18	30.7	23,2	6012		150	28	87.1	64	6217
	110	22	55.3	36	6212		180	41	140	96.5	6317
	130	31	85.2	52	6312		210	52	174	137	6417
	150	35	108	69,5	6412	90	115	13	19.5	22	61818
65	85	10	12.4	12,7	61813		125	18	33.2	31.5	61918
	90	13	17.4	16	61913		140	16	43.6	39	16018
	100	11	22.5	16,6	16013		140	24	60.5	50	6018
	100	18	31.9	25	6013		160	30	101	73.5	6218
	120	23	58.5	40,5	6213		190	43	151	108	6318
	140	33	97.5	60	6313	95	225	54	186	150	6418
70	160	37	119	78	6413		120	13	19.9	22.8	61819
	90	10	12.4	13,2	61814		130	18	33.8	33.5	61919
	100	16	23.8	21,2	61914		145	16	44.8	41.5	16019
	110	13	29.1	25	16014		145	24	63.7	54	6019
	110	20	39.7	31	6014		170	32	114	81.5	6219
	125	24	63.7	45	6214	100	200	45	159	118	6319
	150	35	111	68	6314		125	13	19.9	24	61820
75	180	42	143	104	6414		140	20	42.3	41	61920
	95	10	12.7	14,3	61815		150	16	46.2	44	16020
	105	16	24.2	19,3	61915		150	24	63.7	54	6020

Principal.dimensions.			Basic.load.ratings.		Designation Number	Principal.dimensions.			Basic.load.ratings.		Designation Number
d	D	B	dynamic C	static C_0		d	D	B	dynamic C	static C_0	
mm			kN			mm			kN		
	180	34	127	93	6220	140	280	58	229	216	6326
105	215	47	174	140	6320		175	18	39	46,5	61828
	130	13	20.8	19,6	61821		190	24	66.3	72	61928
	145	20	44.2	44	61921		210	22	80.6	86,5	16028
	160	18	54	51	16021		210	33	111	108	6028
	160	26	76.1	65,5	6021		250	42	165	150	6228
	190	36	140	104	6221	150	300	62	251	245	6328
110	225	49	182	153	6321		190	20	48.8	61	61830
	140	16	28.1	26	61822		210	28	88.4	93	61930
	150	20	43.6	45	61922		225	24	92.2	98	16030
	170	19	60.2	57	16022		225	35	125	125	6030
	170	28	85.2	73,5	6022		270	45	174	166	6230
	200	38	151	118	6222	160	320	65	276	285	6330
	240	50	203	180	6322		200	20	49.4	64	61832
120	150	16	29.1	28	61824		220	28	92.3	98	61932
	165	22	55.3	57	61924		240	25	99.5	108	16032
	180	19	63.7	64	16024		240	38	143	143	6032
	180	28	88.4	80	6024		290	48	186	186	6232
	215	40	146	118	6224	170	340	68	276	285	6332
130	260	55	208	186	6324		215	22	61.8	78	61834
	165	18	37.7	43	61826		230	28	93.6	106	61934
	180	24	65	67	61926		260	28	119	129	16034
	200	22	83.2	81,5	16026		260	42	168	173	6034
	200	33	112	100	6026		310	52	212	224	6234
	230	40	156	132	6226		360	72	312	340	6334

TABLE 4.3: SKF SINGLE ROW DEEP GROOVE BALL BEARINGS

4.4.6 LUBRICATION OF BEARINGS

Lubrication of rolling contact bearings is important because it:

i) Prevents corrosion.
ii) As an anti-friction layer within moving component in bearing.
iii) To reduce the heat generated
iv) To protect the bearing components from dirt and moisture.

Lubrication may be achieved by

i) Drip, splash or mist (oil only).
ii) Lubricators (oil or grease).

iii) Oil circulation within a gearbox by entrainment and splashing, etc.

4.4.7 METHODS OF INSPECTION AND MAINTENANCE OF BEARINGS

- Should always be checked to make sure no dust or impurities.
- Always grease or lubricated for a long life expectancy.
- Maintained over the life time.
- Always replace the old to the new grease.
- Always heard sounds rates.
- Always cleaned and fitted the bearing cover.

Example 3:

A single row, deep groove ball bearing 60 mm inside diameter and design number 6212 is used in a power transmission system at 2100 rpm of speed. The radial load is 4500N and axis load is 3640N. By using bearing table attached, calculate:

a) Outer diameter of bearing.

b) Equivalent bearing load.

c) Bearing life in hours.

Solution:

Given,
d = 40mm,
D.N = 6212
N =2100rpm
F_r = 4500N
F_a = 3640N

a) D,

From table 4.3, d=60mm & D.N=6212,

D = 110mm#

b) P,

P = $X_i F_r + Y_i F_a$

From table 4.3, d=60mm & D.N=6212,

⇒ C = 55.3kN and C_O = 36kN

$$\frac{F_a}{F_r} = \frac{3640}{4500} = 0.809$$

$$\frac{F_a}{C_o} = \frac{3640}{36k} = 0.1011$$

From table 4.2, $F_a/C_O = 0.1011 \approx 0.11$, ⇒ e = 0.3

$\therefore$ i=2, $\frac{F_a}{F_r} > e$ ⇒ X = 0.56, Y = 1.45

Thus, **P** = (0.56)(4500) + (1.45)(3640) **= 7798 N** #

c) L_H

$$L_H = \frac{1x10^6}{60\ N}\left(\frac{C}{P}\right)^a = \frac{1x10^6}{60(2100)}\left(\frac{55.3k}{7798}\right)^3 \ \underline{= 2830.448\ Hrs}\ \#$$

Example 4:

A single row, deep groove ball bearing have 120mm inner diameter and designation number 6024 (SKF catalogue) used at an axial flow compressor with rotation speed 1600rpm. The radial load is 2500N and the axial load 1500N. The compressor operates 8hours every day. Find:

a) The equivalent load for bearing

b) The life in years for the bearing

Solution:

Given,
d = 120mm,
D.N = 6024
N =1600rpm
F_r = 2500N
F_a = 1500N
Operate
= 8hrs/day

a) P,

P = $X_i\ F_r + Y_i\ F_a$

From table 4.3, d=120mm & D.N=6024,

⇒ C = 88.4kN and C_O = 80kN

$$\frac{F_a}{F_r} = \frac{1500}{2500} = 0.6$$

$$\frac{F_a}{C_o} = \frac{1500}{80k} = 0.01875$$

From table 4.2, F_a/C_O = 0.01875 ≈ 0.021, ⇒ e = 0.21

$\frac{F_a}{F_r} > e$, ∴ i=2, ⇒ X = 0.56, Y = 2.15

Thus, **P** = (0.56)(2500) + (2.15)(1500) $\underline{= \mathbf{4625\ N}}$ #

b) L_H

$$L_H = \frac{1x10^6}{60\ N}\left(\frac{C}{P}\right)^a = \frac{1x10^6}{60(1600)}\left(\frac{88.4k}{4625}\right)^3 = 72736.157\ Hrs = 72736.157\ Hrs$$

In years, ... (Assuming 300 working days per year)

$$\boldsymbol{L_H} = \frac{72736.157}{8(300)} = \underline{\mathbf{30.307\ yrs}}\#$$

1. State FOUR (4) functions of bearing lubrication.
2. Define the life of bearing
3. Name and sketch FOUR (4) types of keys
4. Specify FTVE (5) methods of inspection and maintenance of bearings.
5. State the function of:
    a. Bearings
    b. Keys
6. A rigid type of coupling is used to connect two shafts transmitting 30 kW at 100 rpm. The shaft and keys are made of C45 steel with allowable shear stress 40 N/mm^2. Design the shaft.
7. A single row angular contact ball bearing number 6310 is used for an axial flow compressor. The bearing is to carry a radial load of 2500 N and an axial or thrust load of 1500 N. Determine the rating life of the bearing.
8. A single row, deep groove ball bearing 110 mm inside diameter and 170 mm outside diameter, design number 6022 is used in a power transmission system at 1500 rpm of speed. The radial load is 2600 N and 2280N of thrust load and the bearing is rotating inner ring. By using table S(b), determine:
    a. Equivalent bearing load
    b. Life of the bearing in years
9. Single row deep groove type bearings have internal diameter, d = 100 mm from the designation number is 61820, applied to an axial flow compressor. If radial load is 2500 N and axial load is 1500 N determine;
    a. Equivalent bearing load
    b. Life of the bearing in years

Previous Years' Question Papers

Jun 17

a. Explain THREE (3) functions of lubricants on bearing 6M

b. Illustrate these type of keys in power transmissions
   - i) Sunk key
   - ii) Gib head key
   - iii) Woodruff key
   - iv) Tangent key

8M

c. A single row, deep groove ball bearing 70 mm inside diameter and design number 6314 is used in a power transmission system at 2300 rpm of speed. The radial load is 5200N and axis load is 4410N. By using a table, determine the equivalent bearing load. 7M

DIS 16

a. Explain the meaning and function of key. 8M

b. List and sketch Four (4) types of key. 8M

c. A square key has 50mm length and 8mm thick. The shaft and a key has allowable shear stress is $50N/mm^2$. Calculate a suitable shaft diameter. 7M

d. Choose one type of bearing that you know and show clearly the appropriate use and function of the bearing. 5M

JUN 16

a. Define the following terms:
   - i) Life of bearing 2M
   - ii) Equivalent Load 2M

b. Keys connecting shaft to pulley hubs are commonly used to achieve reliable no-slip power in transmission system.
   - i) Discuss **TWO (2)** functions of the key. 4M
   - ii) List **SEVEN (7)** steps in the shaft design. 4M

c. Determine **FIVE (5)** functions of lubrication to the bearing. 10M

Dis 15

d. Explain **FIVE (5)** methods of inspection and maintenance of bearings. 10M

e. A single row, deep groove ball bearing 60 mm inside diameter and design number 6212 is used in a power transmission system at 2100 rpm of speed. The radial load is 4500N and axis load is 3640N. By using bearing table attached, calculate:

   i) Outer diameter of bearing. 2M

   ii) Equivalent bearing load. 9M

   iii) Bearing life in hours. 4M

   iv)

Jun 15

a. Explain the function of key. 5M

b. List **FOUR (4)** types of key. 8M

c. A square key is 50 mm length and 8 mm thick. The shaft and a key have allowable shear stress of 50 N/mm^2. Calculate the suitable shaft diameter. 12M

Dis 14

a. Explain FOUR (4) functions of bearing lubrication. 8M

b. Define the following terms:

   i) Life of bearing. 2M

   ii) Equivalent load. 2M

c. State **THREE (3)** functions of key. 6M

d. List **SEVEN (7)** types of key. 7M

CHAPTER 5: ARM CONNECTOR

OBJECTIVES:

In the end of this chapter, student should know to:

- Explain the principle of connector arm

5.1 INTRODUCTION

This topic introduces various types of connector arm and its role in power transmission.

5.2 CONNECTOR ARM

Connector arms are often used on robot components, on power transmission mechanisms in the industry and as an additional component of easy harmonic movement.

5.2.1 DEFINITION

It is defined as a mechanism of **lever connection** or **workpiece** to **jump** or **move** something. For lever connection it uses some hardware adhesives such as screws, pins, rivets and pin slots so as to support the connecting arm to do their work.

5.2.2 FUNCTION OF CONNECTOR ARM/LINKAGES:

To produce rotating, oscillating and reciprocating motion from rotation of crank and vice versa.

5.2.3 USEFULNESS AND IMPORTANCE OF CONNECTOR ARM:-

- Used as **HAND** movements.
- Act as a worker to **GET** / **TAKE** a load.
- Used as **HANDS LIFTING** and **LOWERING** loads.
- Used as a hand **DRAWN** in engineering work.
- As a **CIRCULAR** motion **EXCHANGER** to **LINEAR** motion or vice versa as in the internal combustion engine.
- Sending greater power such as toggle connector arm.

5.2.4 TYPES OF CONNECTING ARMS

Connecting linkages
- mechanisms built to connect two or more and as an agent of power transfer.

Pivot linkages
- the pivot connecting arm is designed for horizontal movement, parallel inner and outer cylinders and single-shaped

Toggle action linkages
- works with a pin in the middle of the joint to support force actions.

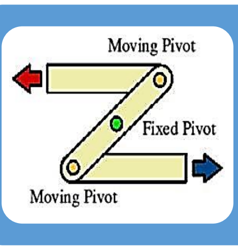
5.2.5 IDENTIFY THE USE OF THE CONNECTOR ARM IN THE POWER TRANSMISSION MECHANISM

- Connection allows a work piece to move in the desired direction and can act on time set in accordance with the required adjustment.
- For reverse movements, the connecting arm is pinned in the middle as a stationary point, while the first arm (power transmission) will push the second arm (the arm is driven) towards the opposite. It works when we set the pivot point in the middle of the connection, then the movement of the entrance is not the same as the outer movement.
- For parallel motion, the large rod above will move left and the small rod below will move to the right. All the rods will move in line with each other.
- For cylinder / rod (crank and cylinder) power transmission through arm movement using shaft components or piston cylinders moving forward or backwards and moving in sliding conditions.
- For toggle action, toggles act as a grip that operates in a pivot and lever connection

5.2.6 APPLICATION OF LINKAGES

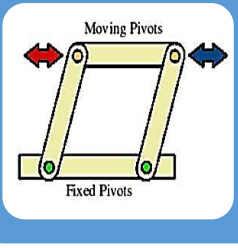
1. Tool Box Drawer
2. Bolt Cutter
3. Windshield Wipers
4. Desk Lamp
5. Bicycle Brakes
6. Crankshaft
7. Piston-Cylinder and Connecting Rod
8. Robotic Arm
9. Mechanical Jack
10. Automobile Suspension

5.2.7 CONNECTOR ARM/LINKAGE WORKS IN VARIOUS SHAPES AND DIRECTIONS:



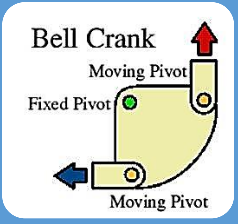
Reverse-motion linkage

- A reverse-motion linkage changes the direction of motion. In the diagram, the linkage looks a little like a "**Z**". The central rod moves around a central fixed pivot. By pulling (or pushing) the linkage in one direction, it creates an exact opposite motion in the other direction.



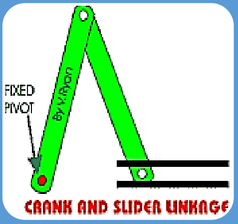
Parallel-motion linkage

- A parallel-motion linkage creates an identical parallel motion. The linkage looks a little like an "**n**". This time, it is the two side rods that move around two central fixed pivots, while the top of the "**n**" moves freely. By pulling (or pushing) the linkage in one direction, it creates an identical parallel motion at the other end of the linkage.



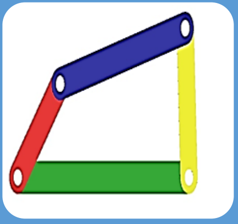
Bell-crank linkage

- A bell-crank linkage changes the direction of movement through 90°. A bell-crank linkage tends to look a little like an "**L**" or, a mirror image of an "**L**". By pulling (or pushing) the linkage in one direction, it creates a similar motion at the other end of the linkage. For example, a bell-crank linkage could be used to turn a vertical movement into horizontal movement, as in a **bicycle braking system**.



Slider Crank linkage

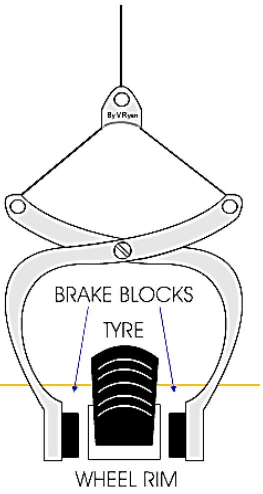
- The Slider-crank mechanism is used to transform rotational motion into translational motion by means of a rotating driving beam, a connection rod and a sliding body.



Toggle motion linkage

- Toggle motion linkage mechanisms are four-bar linkages that are dimensioned so that they can fold and lock. The linkage is dimensioned so that the linkage reaches a toggle position just before it folds. Toggle mechanisms are used as clamps.

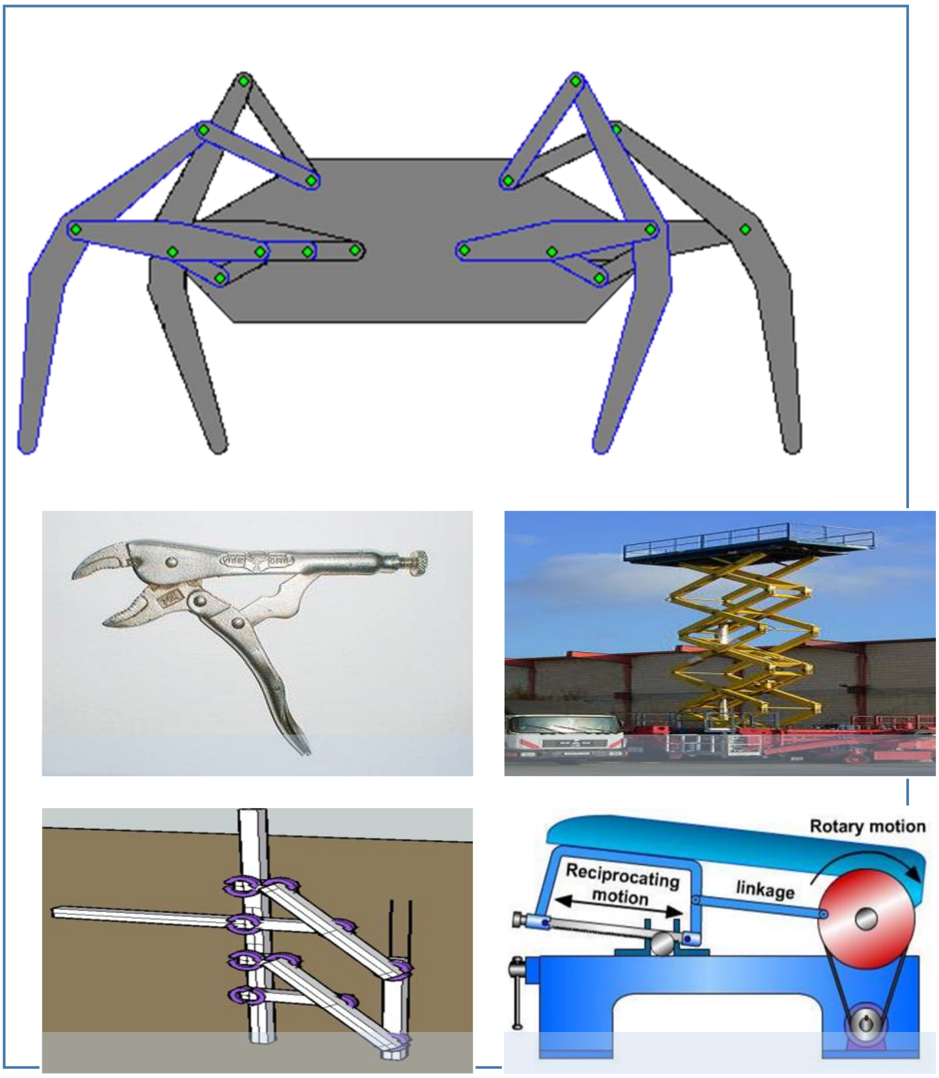
5.2.7.1 *BICYCLE BRAKING MECHANISM.*



A bicycle brake is designed to change the direction of a movement or force by 90° by using bell crank linkage.

As the brake lever on the bike is pulled, the cable moves upwards and forces the brake blocks against the rim of the wheel. A bell-crank linkage changes the direction of movement through 90°. By pulling (or pushing) the linkage in one direction, it creates a similar motion at the end of the linkage. Brake blocks will clamp the wheel rim and reduce the speed due to friction

Example of Linkages Motion

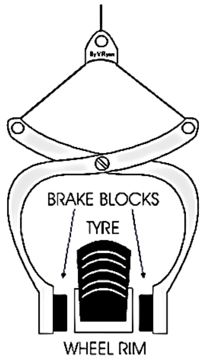


1. List **THREE (3)** types of arm connector. 3M
2. Explain **SIX (6)** functions of arm connector. 12M
3. List **THREE (3)** type of arm connector movement. 3M
4. Explain the operational of **TWO (2)** types of arm connector. 4M
5. Explain briefly and sketch **TWO (2)** types of linkages. 8M
6. State and explain **THREE (3)** types of linkages. 9M
7. Differentiate between **THREE (3)** types of motion linkages. 9M

Dis & Jun 17

c. List FOUR (4) examples of application of linkages. 4M

d. According to the picture below, explain the operation of Bell Crank Linkage.



8M

Jun 15

a. Define

i. Connector arm 2M

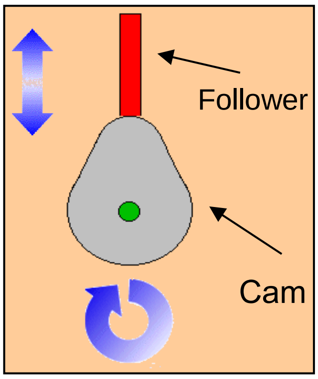
CHAPTER 6: CAM

OBJECTIVES:

In the end of this chapter, student should know to:

- Explain the principle of Cam
- Describe types of cam followers

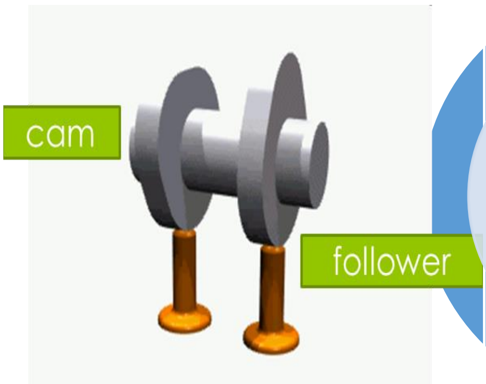
6.1 INTRODUCTION



This topic introduces types of cam, followers and its related terminologies. This topic also contains analysis of motion and calculation on speed and acceleration in followers.

6.2 CAM

Cams are a mechanical part which impart a prescribed motion to another part by direct contact. It may remain stationary, translate or rotate Cam can easily produce complicated output motions which are otherwise difficult to achieve. The driver is called the cam and the driven member is called the follower. A follower is a device that been driven by a cam and the motion of it may translate or oscillate. A familiar example is the camshaft of an automobile engine, where the cams drive the push rods (the followers) to open and close the valves in synchronization with the motion of the pistons. The cams are widely used for operating the inlet and exhaust valves of internal combustion engines, automatic attachment of machineries, paper cutting machines, spinning and weaving textile machineries, feed mechanism of automatic lathes etc.



A cam system consists of four elements

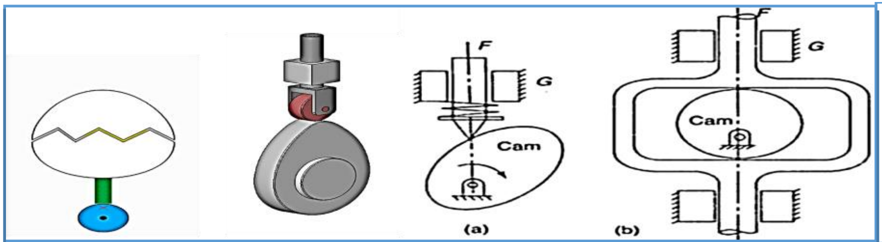
- Cam
- Follower
- Follower system
- Drive

6.3 CLASSIFICATION OF CAMS

Cams are classified in three ways:

a. In terms of their shape, such as wedge, radial, cylindrical, globoidal, conical, spherical, or three-dimensional;
b. In terms of the follower motion, such as dwell-rise-dwell (DRD), dwell-rise-returndwell (DRRD), or rise-return-rise (RRR); or
c. In terms of the follower constraint, which is accomplished by either positive drive or spring load as mentioned previously.

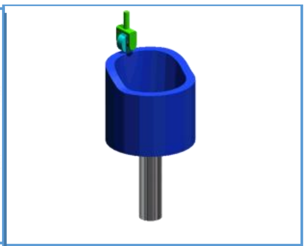
6.3.1 According to the Shape



1) Plate/Radial/Disc Cams

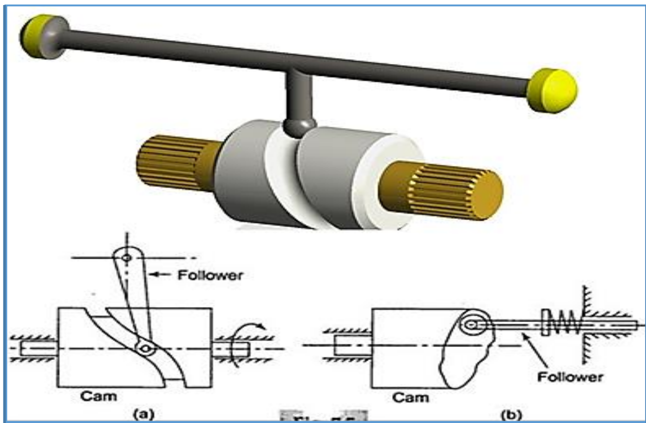
A cam in which the follower moves radially from the centre of rotation of the cam is known as a radial or a disc cam **(Fig. (a) and (b)]**.

Radial cams are very popular due to their simplicity and compactness.



2) End Cams

In this case the end of the cylindrical cam has the profile machined on the end..



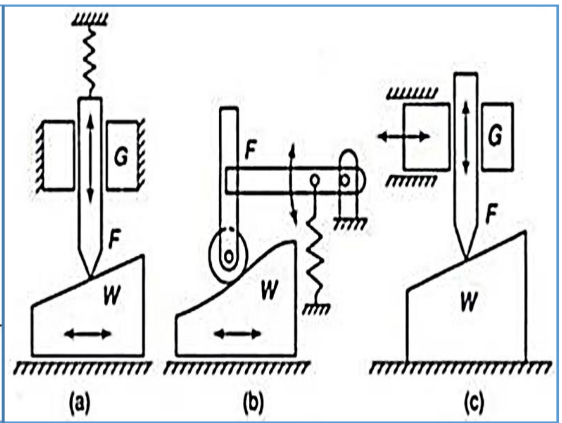
3) Cylindrical/Barrel Cams

In a cylindrical cam, a cylinder which has a circumferential contour cut in the surface, rotates about its axis.

The follower motion can be of two types as follows: In the first type, a groove is cut on the surface of the cam and a roller follower has a constrained (or positive) oscillating motion [Fig.(a)].

Another type is an end cam in which the end of the cylinder is the working surface (b).

A spring-loaded follower translates along or parallel to the axis of the rotating cylinder.



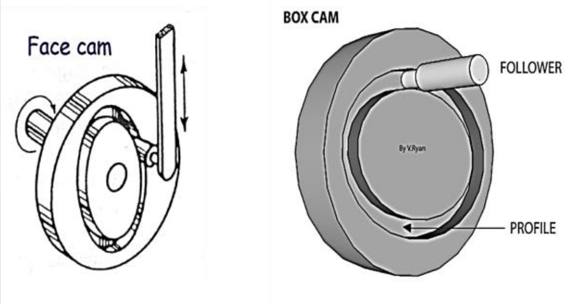
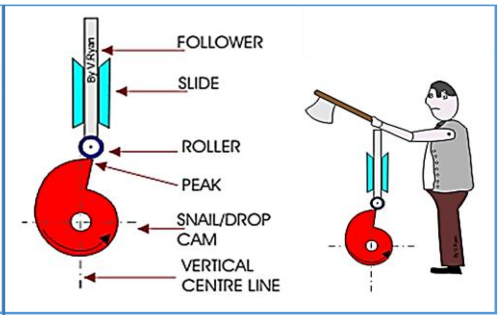
4) Wedge and Flat Cams

A wedge cam has a wedge **W** which, in general, has a translational motion.

The follower **F** can either translate **Fig.(a)** or oscillate **Fig.(b)**.

A spring is, usually, used to maintain the contact between the cam and t he follower.

In **Fig.(c)**, the cam is stationary and the follower constraint or guide **G** causes the relative motion of the cam and the follower.

	
5) Face Cams produces motion by using a follower riding on the**face** of a disk. The most common type has the follower ride in a slot so that the captive follower produces radial motion with positive positioning without the need for a spring or other **mechanism** to keep the follower in contact with the control surface.	**6) Snail / Drop Cam** Eccentric, heart shaped and pair shaped cams generally allow for a slow rise and fall of the follower. However, a snail drop cam is used where the drop or fall of the follower must be sudden.

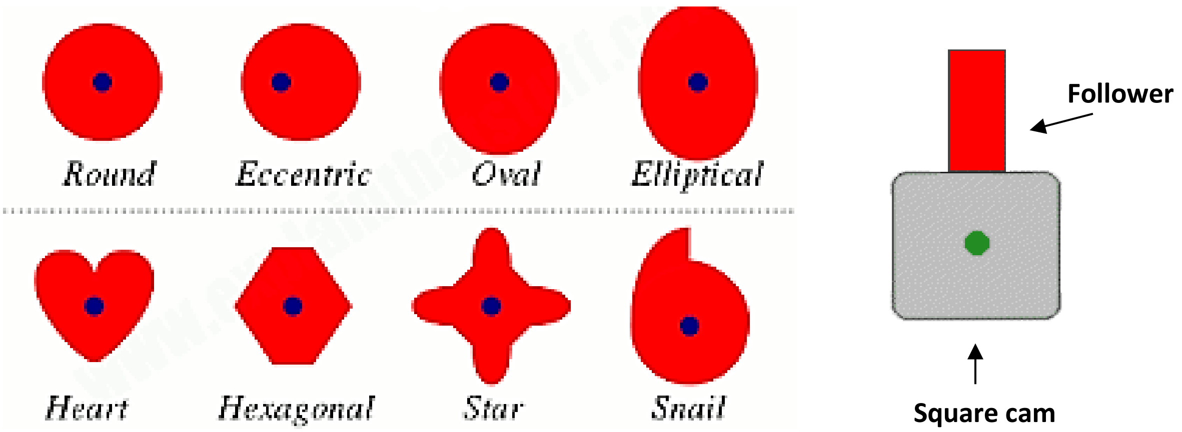
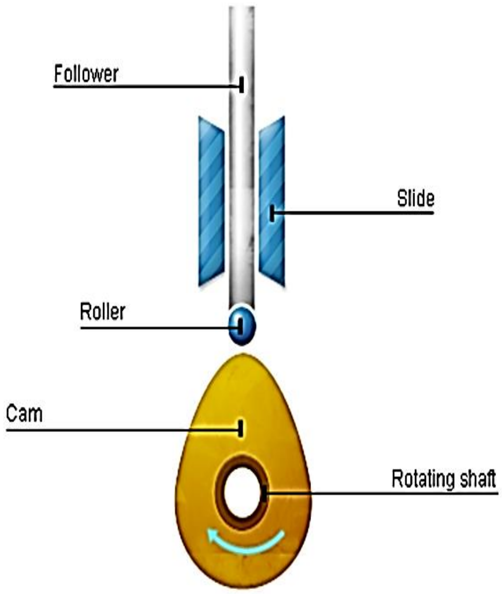


FIGURE 6.1: SOME COMMON TYPES OF CAMS

6.3.2 How a Cam Works

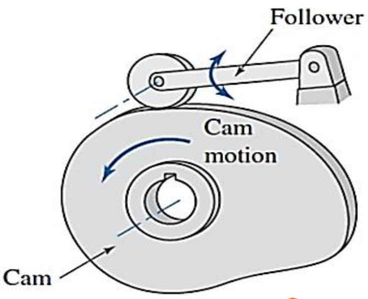


i) The diagram below shows the main components of cams consisting of cams, shafts, followers and roller.
ii) The cams will be attached to the shaft by using a key.
iii) The follower constantly pressing the cam surface attached at one end, with a roller.
iv) The diagram below shows the main components of cams consisting of cams, shafts, followers and distillers. The cams will be attached to the shaft by using the key. The followers constantly pressing the cam surface attached at one end, with a roller (roller).
v) When the shaft rotates, the roller moves according to the uneven surface of the cam, the follower will move

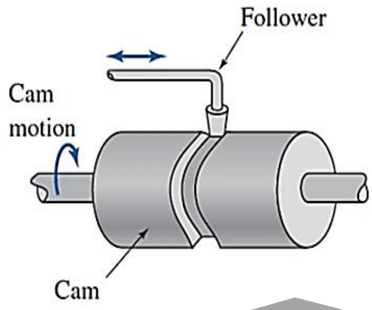
up and down according to the cam contour changes.

vi) The roller will press the cam because of the force of the spring or the weight of the follower.

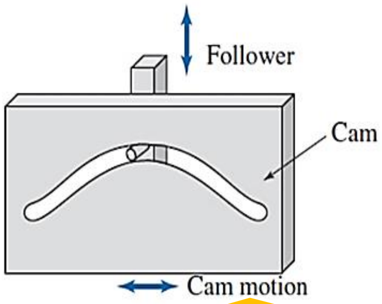
6.3.3 Cam motion



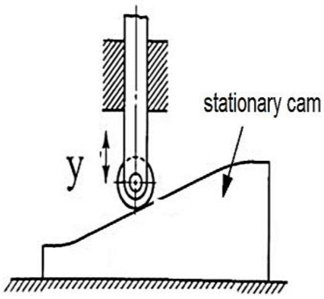
Rotating disc / plate (Radial / Disc Cam)



Rotating cylindrical (Barrel Cam)

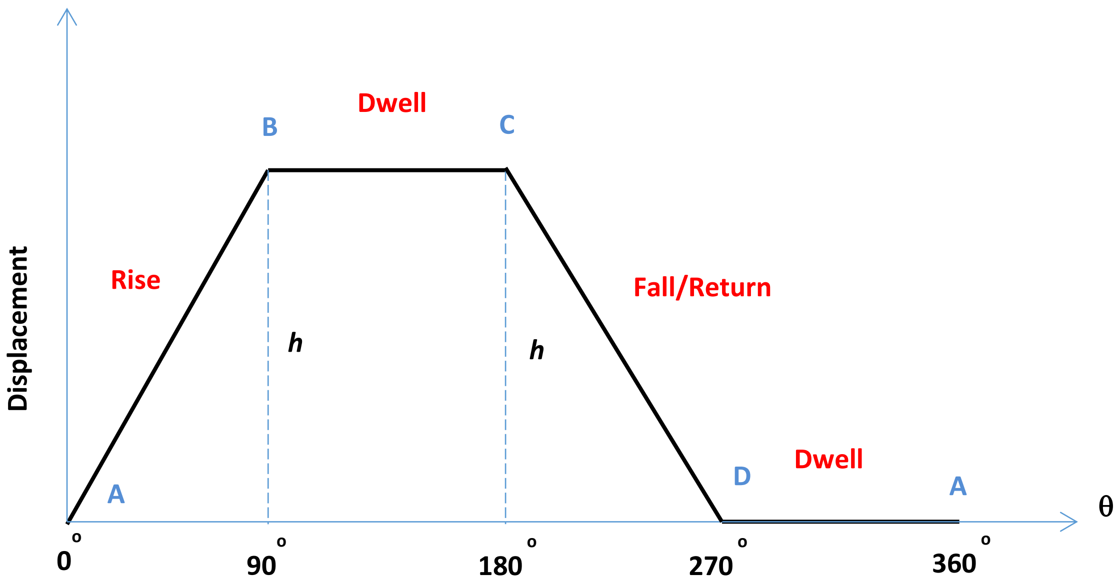


Translating (Flat Cam with Groove)



Stationary (Wedge Cam)

6.3.4 Cam Operational Based On The Displacement Diagram



Dwell is the time when followers are not moving. It is shown above as a straight horizontal line between the dwell angle.

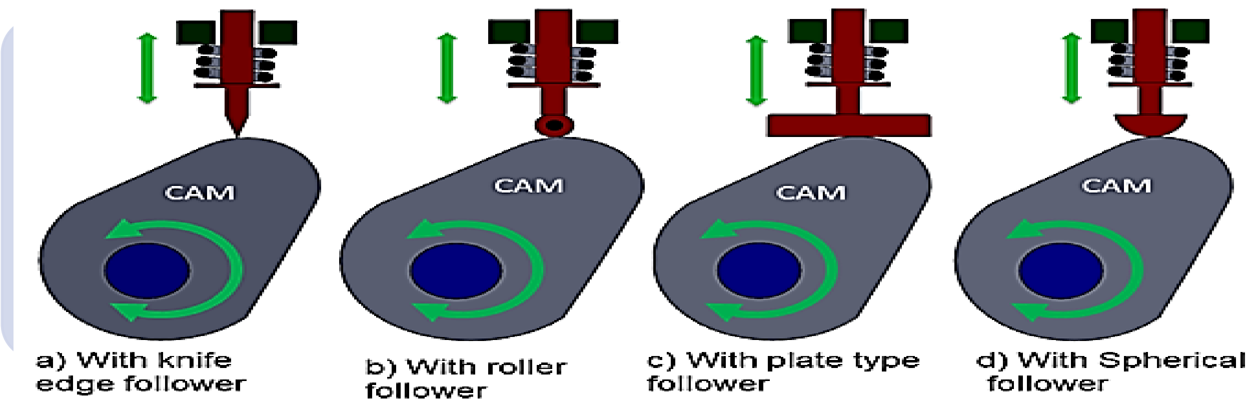
- The DISPLACEMENT DIAGRAM is the presentation of the movement layout of a cam. The displacement diagram length is the same as the working curve. The height of the displacement diagram is the same as the working curve radius.
- The displacement of the follower is plotted along the y axis and angular displacement θ of the cam is plotted along x-axis.
- From the displacement diagram, velocity and acceleration of the follower can also be plotted for different angular displacements θ of the cam.
- The displacement, velocity and acceleration diagrams are plotted for one cycle of operation i.e., one rotation of the cam.
- Displacement diagrams are basic requirements for the construction of cam profiles.
- Construction of displacement diagrams and calculation of velocities and accelerations of followers with different types of motions are discussed in the following sections.

6.3.5 Classification of Followers

A follower can be classified in three ways

1. According to the surface contact/shape
2. According to the motion of the Follower
3. According to the path of motion of the Follower

<u>According to the surface contact/shape:</u>



KNIFE EDGE
Simple form but edge wears rapidly

ROLLER
Most common
Pressure angle should be low to prevent jamming

FLAT FACE
Use with a steep cam as it will not jam
Deflection or misalignment can cause high surface stress

SPHERICAL FACE/ MUSHROOM
Radiuses face compensates for deflection or misalignment

According to the motion of the Follower:

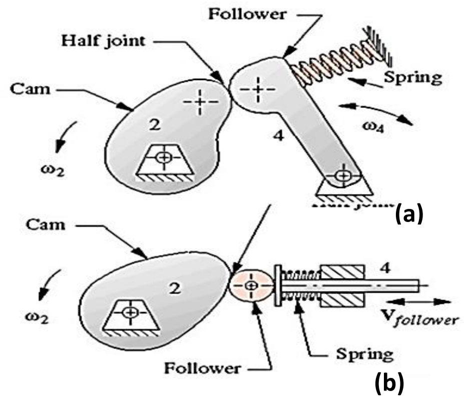


FIGURE 6.2: MOTION OF FOLLOWERS

The followers, according to its motion, are of the following two types:

1. ***Oscillating or rotating follower.***

When the uniform rotary motion of the cam is converted into predetermined oscillatory motion of the follower, it is called oscillating or rotating follower. The follower, as *shown in Fig 6.2(a), is an oscillating or rotating follower.*

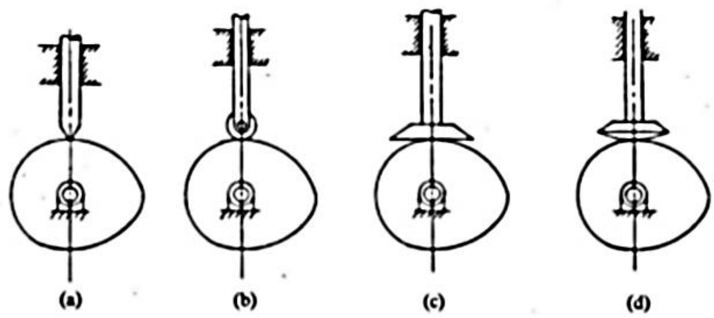
2. ***Reciprocating or translating follower.***

When the follower reciprocates in guides as the cam rotates uniformly, it is known as reciprocating or translating follower. The followers as shown in Fig 6.2(b)is reciprocating or translating followers.

According to the path of motion of the Follower

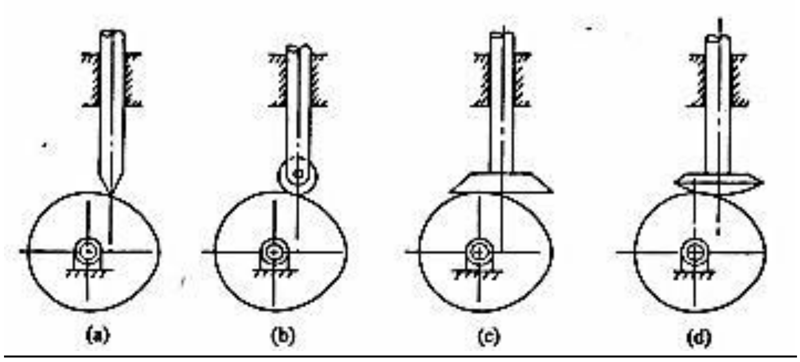
1. ***On-center / radial follower***

The lines of movement of in-line cam followers pass through the centers of the camshafts



2. ***Offset follower***

The lines of movement are offset from the centers of the camshafts



Note : In all cases, the follower must be constrained to follow the cam. This may be done by springs, gravity or hydraulic means. In some types of cams, the follower may ride in a groove.

6.3.6 Follower constraint

Constraints are needed to maintain the follower with the cam at all speed and all times. List of constraints:-

i. Pre-loaded spring
ii. Positive drive
iii. Gravity
iv. Pneumatic / hydraulic force

6.3.7 Cam nomenclature

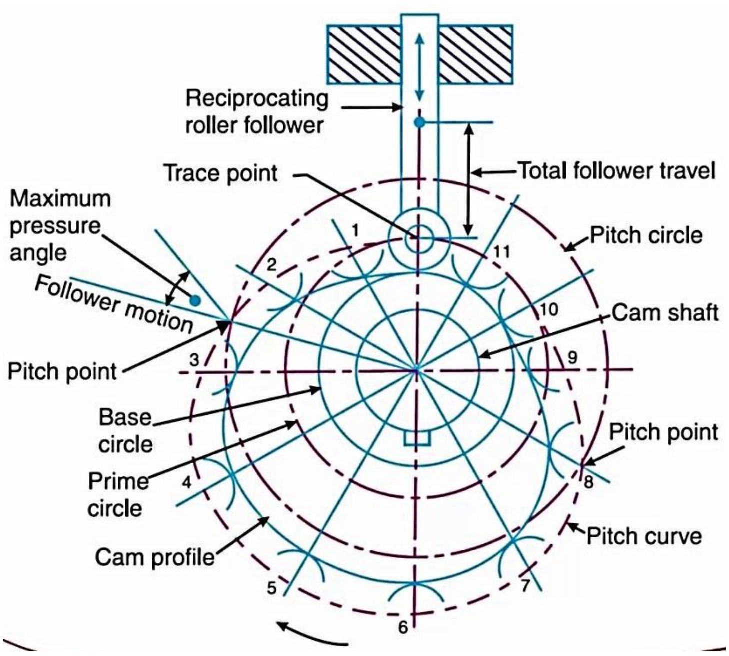


FIGURE 6.3: RADIAL CAM WITH RECIPROCATING FOLLOWER

Cam Profile

- The contour of the working surface of the cam. (groove cam has inner & outer profile)

Base Circle

- It is the smallest circle that can be drawn to the cam profile

Prime / Minor Circle

- The smallest circle drawn, tangential to the pitch curve, with its center on the axis of the camshaft.

Trace Point

- The point at the knife edge of a follower, or the center of a roller, or the center of a spherical face.

Working curve/Major Circle

- The working surface of a cam in contact with the follower. For the knife-edge follower of the plate cam, the pitch curve and the working curves coincide.
- The largest circle drawn, tangential to the pitch curve, with its center on the axis of the camshaft.

Pressure Angle

- The angle between the normal to the pitch curve and the direction of motion of the follower at the point of contact. (max 30° for radial follower)

Pitch point

- Point where is the maximum pressure angle occurred

Pitch Curve

- The path of the tracer point.

6.3.8 Classification According To Movement Of The Follower:

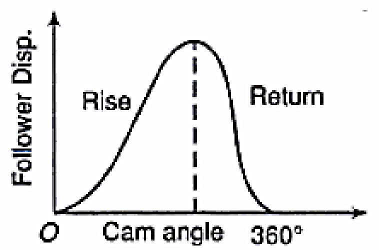
The motions of the followers are distinguished from each other by the dwells ,rises and returns they have.

Rise of a cam: The motion of the cam which tend to lift the follower is known as the rise motion.

Dwell of a cam: The rotation of the cam for which the follower is stationary at its position is known as dwell of the cam.

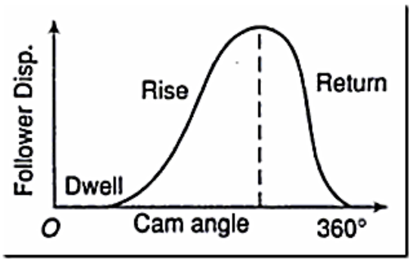
Return of a cam: The motion (rotation) of the cam for which the follower tends to move its original position is known as the return motion of the cam.

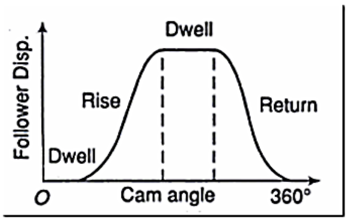
Cams are classified according to the motions of the followers in the following ways:



1. *Rise-Return-Rise* (R-R-R): In this, there is alternate rise and return of the follower with no periods of dwells (as shown in figure). It's use is very limited in the industry. The follower has a linear or an angular displacement.

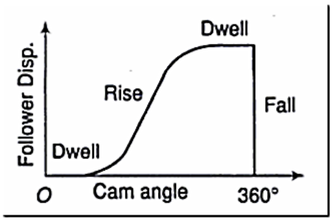
2. *Dwell-Rise-Return-Dwell (D-R-R-D):* In such a type of cam, there is rise and return of the follower after a dwell (as shown in figure). This type is used more frequently than the R-R-R type of cam.





3. *Dwell-Rise-Dwell-Return-Dwell (D-R-D-R-D):* It is the most widely used type of cam. The dwelling of the cam is followed by rise and dwell and subsequently by return and dwell as shown in Fig.

4. Return of the follower is by a fall, the motion may be known as Dwell-Rise-Dwell (D-R-D).



There are many follower motions that can be used for the rises and the returns. In this chapter, we describe a number of basic curves. Cam follower systems are designed to achieve a desired oscillatory motion. Some of the standard follower motions are as follows. They are, follower motion with,

i. Uniform velocity
ii. Modified uniform velocity
iii. Uniform acceleration and deceleration
iv. Simple harmonic motion
v. Cycloidal motion

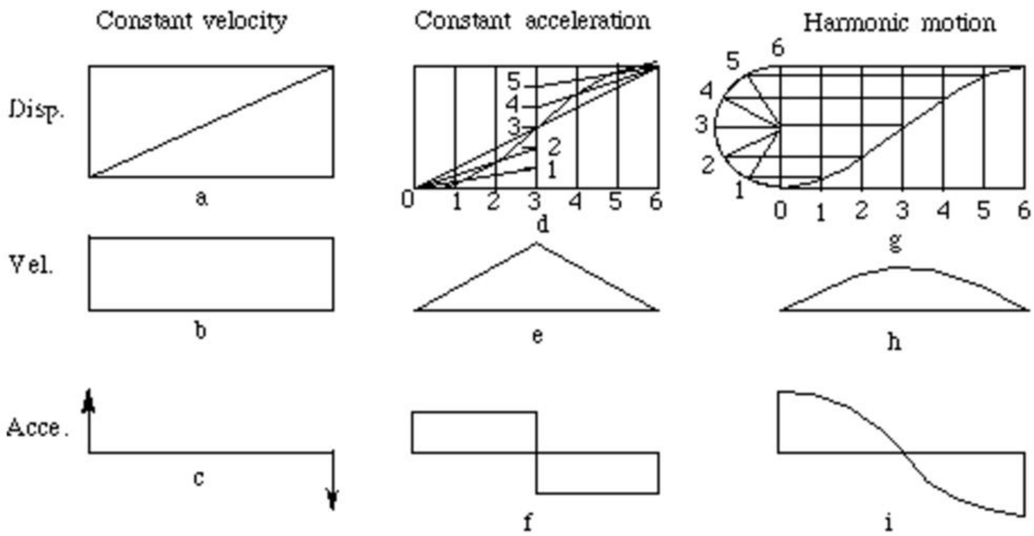
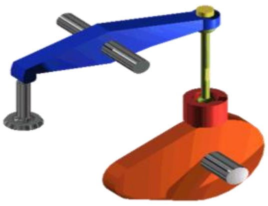


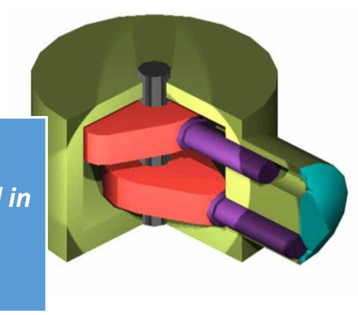
FIGURE 6.4: MOTION EVENTS

6.3.9 Examples of a Rotary cams in operation.

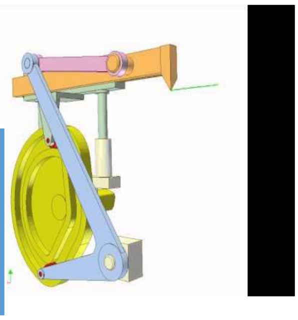
Control the movement of the engine valves.



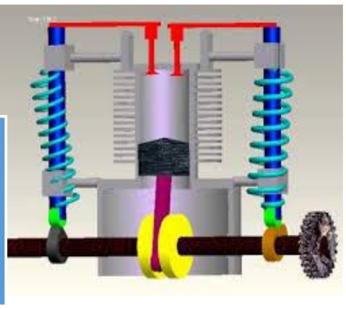
Cams used in a pump.

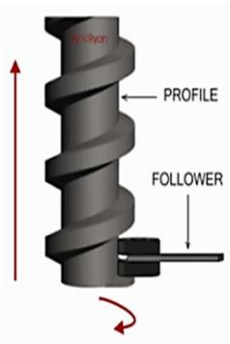


Cam mechanism of 2 followers



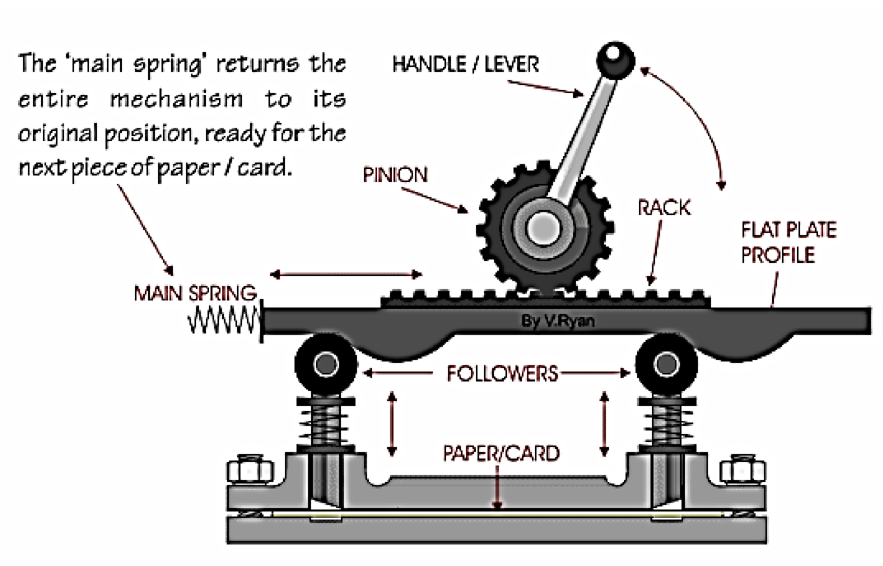
4 Stroke IC Engine Working Mechanism





The Cylindrical Cam / Barrel Cam / Unusual Cam

- As the cylinder cam profile rotates, the follower moves upwards. When the follower reaches the top, the cylinder cam rotates in the opposite direction and follower moves back down.



1. List Two (2) applications of cam follower system.
2. State the operational of the cam based on the displacement diagram.
3. State the function of Cams and give Two (2) examples where Cams is applied in industry.
4. Classify THREE (3) various types of followers and give ONE (1) example for each classification.

DIS 16

Q2

a. State THREE (3) types of cam and TWO (2) types of follower	2M
b. List FOUR (4) types of follower motion cause by the cam movement.	8M

JUN 16

a. Define the terms below:	
i) CAM	2M
ii) CAM followers	2M
b. Compare **FOUR (4)** types of CAM follower movements	8M

Dis 15

Q5

a. Identify the different type of CAM labelled A, B, C and D in Figure 5.1 4M

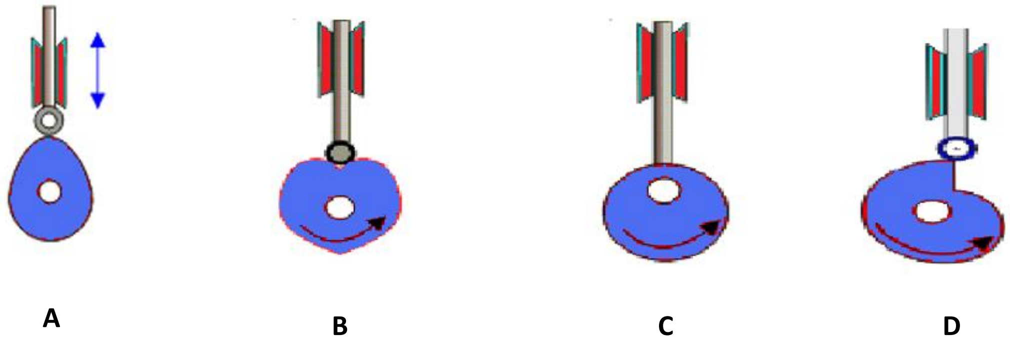


Figure 5.1: Types of CAM

b. Construct a knife-edge translating follower diagram. 12M

Dis 14

a. Define the terms below:
   i) Linkages
   ii) Cams

4M

b. Explain **THREE (3)** types of linkages. 6M

c. List **FOUR (4)** types of:
   i) Followers 4M
   ii) Cams 4M

d. Explain **THREE (3)** types of follower motion caused by the cam movement. 3M

e. List **FOUR (4)** methods used to ensure that the follower remains in contact with the cam.
4M

CHAPTER 7: GEAR

OBJECTIVES:

In the end of this chapter, student should know to:

Explain the principle of gear ratio.

7.1 INTRODUCTION

This topic explains gear networks and the use of certain gears in shaft with specific position. This topic also involves problems on gear application in machineries.

7.2 DEFINITION

FIGURE 7.1: TRANSMISSION SYSTEM

A gear is a toothed, cylindrical wheel component used for transmitting motion and power from one shaft to another. Torque is transmitted, and because the gear is rotating, power is also transmitted while it in motion.

There are at least two or more gear wheels will mesh with each other through interlocking teeth between them. This interlocking teeth on wheel to ensure no slip occur between those gear wheels. The shape of the gear wheel teeth is important in order to produce smooth transfer of the motion. Gear always work as pair or more and it never serve the purpose as a single one. Gear common functions are;

a) Change the rotation orientation
b) Change the ratation velocity
c) Transfer rotational motion to anorther axis
d) Synchronized the motion.

The motion and power transmitted by gears is kinematically equivalent to that transmitted by it frictional wheels that created by meshing gear this known as mechanical advantages of the gear wheel. This because gear wheel can create positive drive, wheres the gear teeth can be act like small lever while it in action. Gear can easily found on precise application as it most componant such as watch engine.

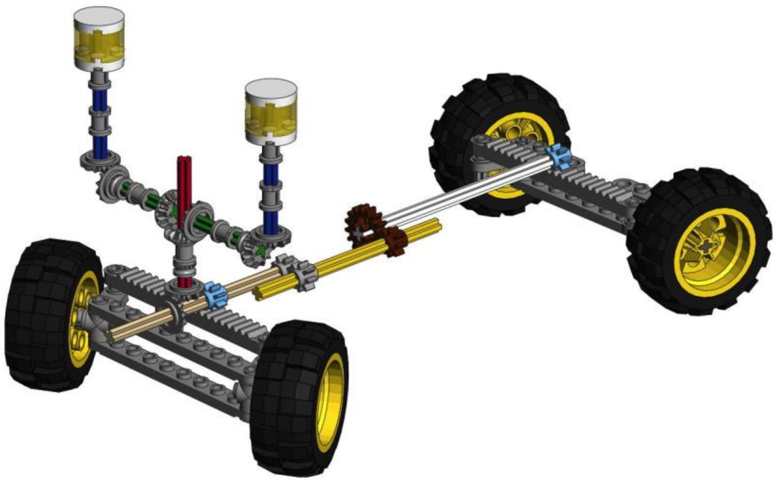


FIGURE 7. 2: STEERING SYSTEM

7.3 GEAR TYPES AND CLASSIFICATION

Gear can be classified by category type and by the orientation of axes. Gears are classified into 3 categories by the relative motion of the axes of rotation; parallel axes gears, intersecting axes gears, and nonparallel and nonintersecting axes gears.

7.3.1 CATEGORY TYPES OF GEARS

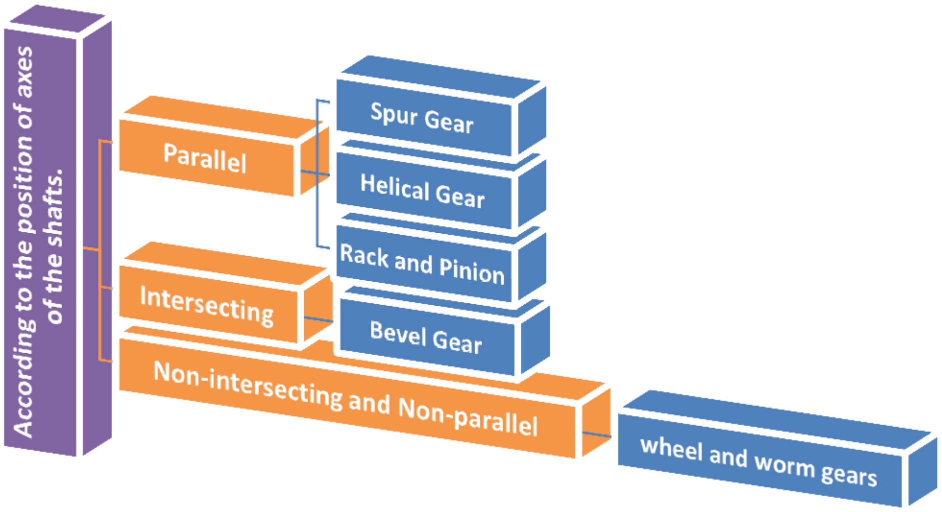


FIGURE 7.3: TYPES OF GEARS

Spur gear

FIGURE 7.4: SPUR GEAR

Spur gears have teeth parallel to the axis of rotation and are used to transmit motion from one shaft to another. These types of gear are the simplest in design, economy of manufacture and maintenance, and absence of end thrust whereas it imposes only radial loads on the shaft. Spur gears are known as slow speed gears. If noise is not a can be neglected in designing proses, spur gears can be used at almost any speed.

Helical gears

Helical gears are similar to the spur gear but it has inclined teeth to the axis of the rotation in the form of a helix, hence known as helical gear. Helical gears can be used for the same application as spur gears and beneficial from that, these types less noisy compare to the spur gears because of the gradual engagement of the teeth during meshing.

Helical gears can take higher loads than similarly sized spur gears since it can impose both radial loads and thrust load on their shaft. The angle of the helix on both the gear still the must be same in specification but opposite in direction, example a right hand pinion meshes with a left hand gear.

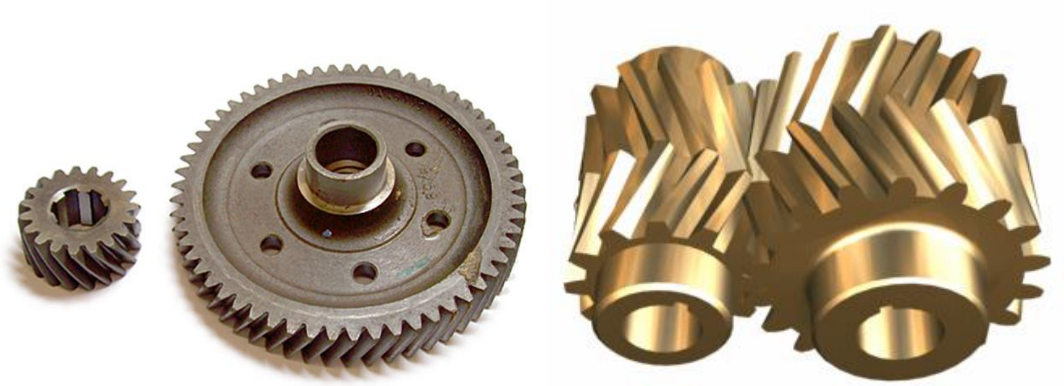


FIGURE 7.3: HELICAL GEAR

Bevel gears

Bevel gears have teeth formed on conical surface and are used mostly for transmitting motion between intersecting shafts. Straight bevel gears can be used on shafts at any angle, but right angle is the most common.

FIGURE 7.4: SPIRAL BEVEL GEAR

FIGURE 7.5: STRAIGHT BEVEL GEAR

Bevel Gears have conical blanks. The teeth of straight bevel gears are tapered in both thickness and tooth height. Another types of straight bevel is spiral bevel gear whereas the grove is curving on the conical surface

Worm gears

Worm gears resembles a screw but it use to transmit the motion through it thread loop. Commonly it used to transmit at 90^0 and it cost reducing a lot of transmitting power. The shafts of worm gears place in parallel planes and may be skewed at any angle between zero and a right angle. Mention before worm gears has screw threads.

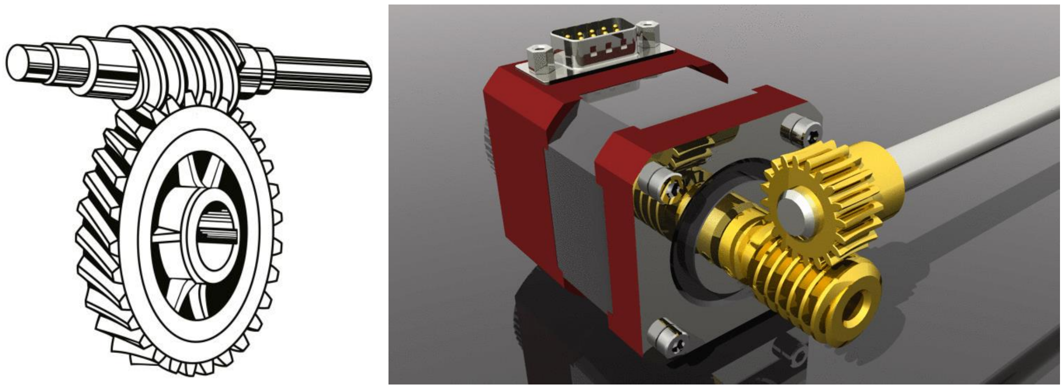


FIGURE 7.6: WORM GEAR

Due to this, worm gears are quiet, vibration free and give a smooth output. Worm gears and worm gear shafts are almost invariably at right angles.

Rack and pinion

It was a combination a rack whereas toothed bar as a driven part and a spur gear as a drive part. The main advantages of this gear type are torque can be converted to linear force by meshing a rack with a pinion: the pinion which commonly use spur gear turns; the rack moves in a straight line.

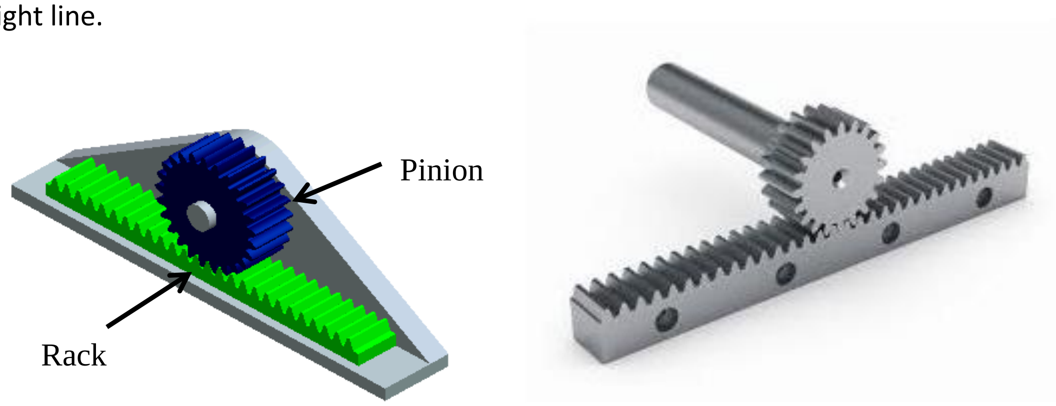
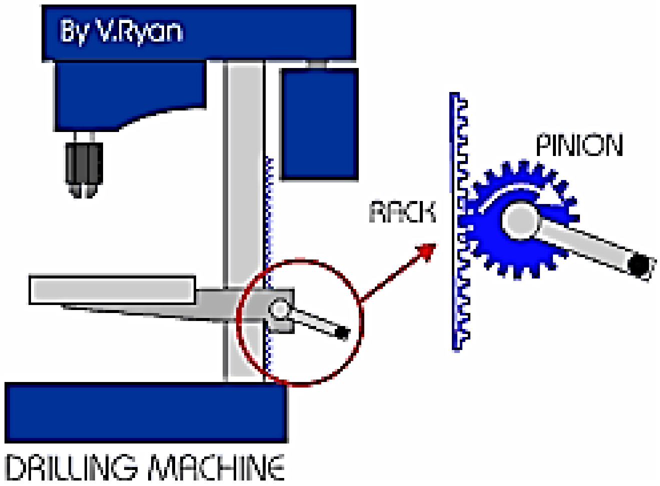
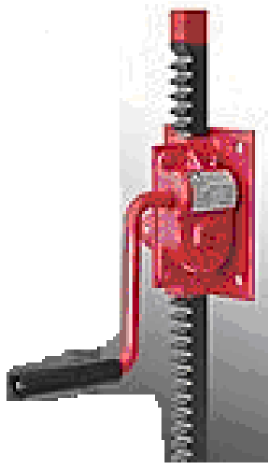


FIGURE 7.7: RACK AND PINION GEAR

Such a mechanism is used in automobiles to convert the rotation of the **STEERING** wheel into the left-to-right motion of the tie rod. Racks also feature in the theory of gear geometry, where, for instance, the tooth shape of an interchangeable set of gears may be specified for the rack (infinite radius), and the tooth shapes for gears of particular actual radii then derived from that.





Heavy Duty

Car Jack

7.3.2 GEAR CLASSIFICATION

Gear can be classified according to the position of the gear wheel shafts axes. These classifications of gear train often suggest the type of gear to be chosen. The axes of two shafts between which the motion is to be transmitted can be defined as;

a. Parallel shaft
b. Intersecting shaft
c. Non-intersecting and non-parallel shaft

This classification is use to determine the suitable type of gears to be used based on the application in which they are to be used to be in.

Parallel shaft

In this type of gearing, the axis of both the gears tends to be parallel to each other. Spur Gears, Helical Gears, and Double Helical are example gear in these categories. As shown in figure 7.3 this gear axis is parallel by each other.

Some typical application of spur and helical are automobile gearboxes, industrial gearboxes, etc. Some of the application areas of Herringbone gears are in the gearboxes used for steel rolling mills, etc.

Intersecting shaft

In this type of gearing the axis of the gears tend to be perpendicular to each other. The angle of perpendicular not at mean at 90^0 such as in figure 7.6 which this rack and pinion gear axis are in perpendicular position, but intersecting also cover for skewed axis position but not in parallel position axis and center of each axis will intersected. As in figure 7.4, bevel gears are mounted on intersecting shafts and this bevel gear can place at any desired angle, although 90° is most common. These shaft categories always are designed and manufactured in pairs and as a result, each pair is not always interchangeable.

Non-intersecting and non-parallel shaft

In this type the two gearing axis do not intersect each other, but the gear still have interacted of meshed each other to transmitting the motion. The two types of gearing that commonly fall under this category are Worm Gear and the hypoid gear.

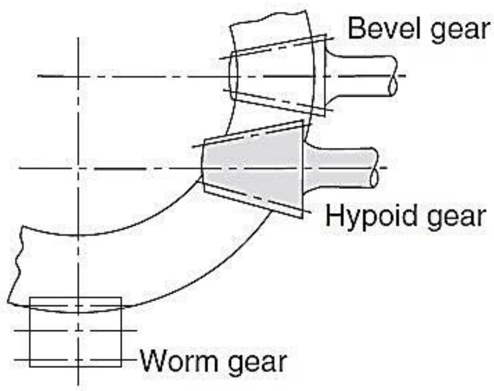


FIGURE 7.8: GEAR SHAFT POSITION

Some typical applications of worm gears are in the passenger lifts used in the buildings. Another typical application of the Hypoid gear is in the rear axle of the busses, lorries and heavy vehicles. As shown in figure 7.8, hypoid gear axis position was not intersect to the bevel gear axis, but transmitting motion still can occur.

7.4 **GEAR NOMENCLATURE**

The terminology of spur-gear teeth is illustrated in figure 7.9 these are common term use while using gear. The pitch circles of a pair of mating gears are tangent to each other. A pinion is the smaller of two mating gear and the larger is often called the gear.

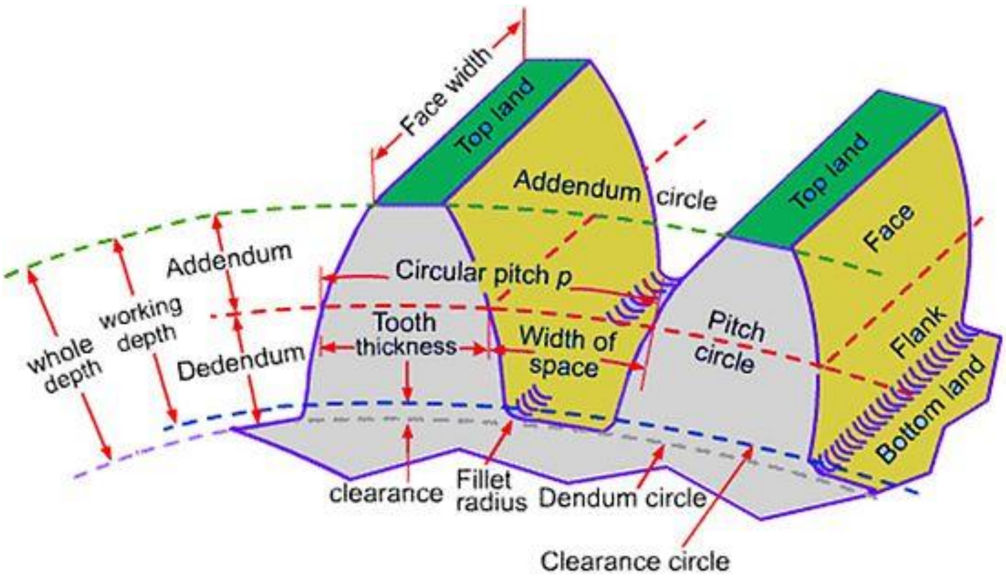


FIGURE 7.9: NOMENCLATURE OF SPUR GEAR TEETH

Pitch circle: Can roughly be defined as the circle having radius as the mean of the maximum radius (to the tip of the gear teeth) to the radius of the base of a gear tooth. However tooth proportions can vary considerably, with both root and tip adjusted to suit running conditions and manufacturing processes, making this definition somewhat unreliable.

Addendum: The tooth portion above the pitch circle (towards the tooth tip).

Dedendum: The tooth portion below the pitch circle (towards the tooth root).

Flank: The face of a gear tooth which comes in contact with the teeth of another gear. So,a flank is an important part of a gear.

Fillet: Fillets in the root region are of less importance since they don't come into contact with other gear teeth. However root fillets are of great importance with regard to tooth bending strength, and therefore power ratings. Gears with little or no fillet in the root are prone to tooth breakage, as the sharp corner acts as a **STRESS RAISER**.

Circular pitch: The sum of the width of a tooth and a space between the teeth of a gear. Circular pitch is an important parameter as it indicates the size of the tooth of a gear

Gear Relation

Module, m is the ratio of pitch circle diameter in millimeter (matric) to the number of teeth also known as in metric system

$$P = \frac{Z}{d}$$

$$p \text{ (circular pitch)} = \pi d / Z$$

$$Pp = \pi$$

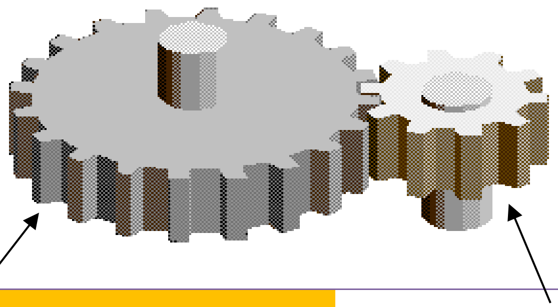
Where,
- P = diametral pitch, teeth per inch
- Z = number of teeth
- d = pitch diameter (gear diameter),mm
- p = circular pitch (inch)

Metric system/module, m = $\frac{d}{Z}$ (mm)

7.5 DRIVER AND DRIVEN

Two meshed gears always rotate in opposite directions.

Spur Gears



Driven gear

Driver gear

7.6 GEAR TRAINS

Gear trains require two or more gears meshed purposely to transmitting motion from one axis to another axis. This can be idealized as two smooth discs with their edges touching and no slip between them. Commonly gear trains have axes on each gear wheel and it relative to the frame for all gears to making up the gear trains.

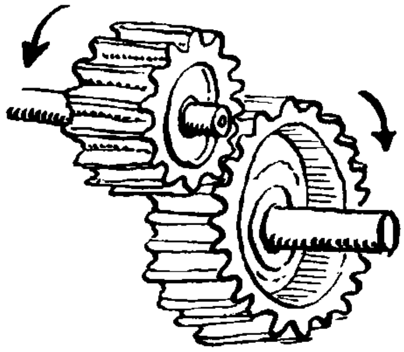


FIGURE 7.10: A PAIR OF SPUR GEAR TRAIN

GEAR RATIO

The ratio mating teeth called a gear ratio. It is a ratio of number of teeth on gear (driver) to number of teeth on the mating gear (driven) or follower. Further that a ratio of speeds of any pair of gears in mesh is the inverse of their number of teeth, therefore the formula can be write as:

$$\text{gear ratio} = \frac{z_2}{z_1} = \frac{\boldsymbol{No. of\ teeth\ on\ driven\ gear}}{\boldsymbol{No. of\ teeth\ on\ driver\ gear}} = \frac{\boldsymbol{N_1}}{\boldsymbol{N_2}}$$

Where, Z = the number of gear teeth

N = rotational speed, RPM

The value of this ratio can be positive (+ve) and negative (-ve) to indicate its **rotation** direction compare to the driver rotation. Since the speed ratio (or velocity ratio) of gear train is the ratio of the speed of the driver to the speed of the driven or follower

The constant velocity ratio between 2 gears easily archive at close connection of two gears. For large distant of two gear, 2 method can be used to transmitting the motion. First by using larger sized of gear. This method is **very inconvenient** and **uneconomical**. Second method is by providing one or more intermediate gears between drivers to driven gear. This method is **very convenient** and **economical** compare to the first method.

INTERMEDIATE GEARS

Intermediate gears are also known as **IDLER** gears, as they do not affect the speed ratio between input and the output of the system. The function of idler gear is to connect gears where a **large center distance** is required or to obtain the **desired direction** of rotation at the driven gear result such example clockwise or anticlockwise at the output of the gear compare to the input gear.

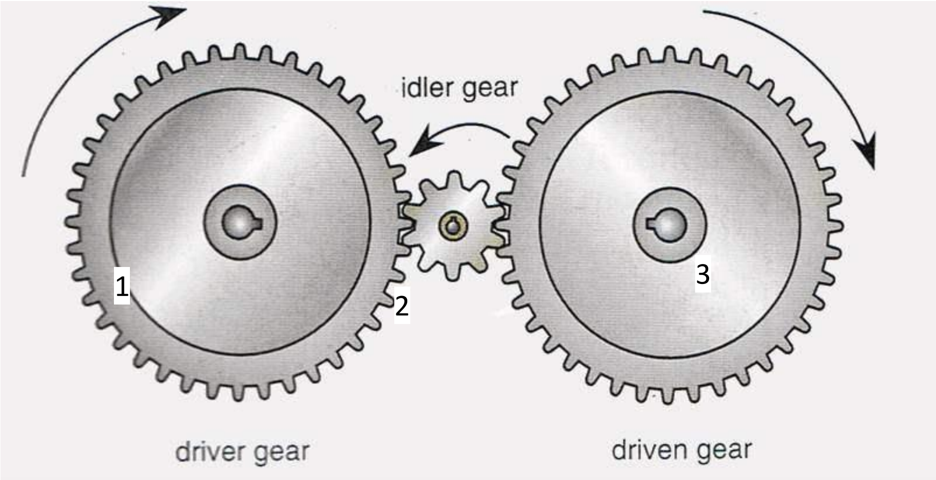


FIGURE 7.11: IDLER GEAR

It may be noted that when the number of intermediate gear are **odd**, the motion of both the gear at input and output is **same direction** and if the number of intermediate gears are **even**, the motion of the driven or follower will be in the **opposite direction** of the driver.

7.6.1 SIMPLE GEAR TRAIN

Gear trains are two or more gears meshed for the purpose of transmitting motion from one axis to another. As shown on the figure 7.10 a pair of spur gear train and figure 7.11 idler gear train, it has three gear wheels and it was an example a simple gear train when there is **only** one gear wheel on each axis of shaft. The train ratio with two gears as in figure 7.10 can be computed by formula;

$$\frac{N_1}{N_2} = \frac{Z_2}{Z_1}$$

The relations between gears are represented by their pitch circles whereas meshing teeth gear must have same pitch circle, P_c. It may be noted that the motion of the driven gear is opposite to the motion of driving gear for two wheel of gear.

Referring at figure 7.11, this simple gear train has 3 gear wheel. Wheres

N_1 = Speed of driver, rpm

N_2 = Speed of intermediate gear, rpm

N_3 = Speed of driven or follower, rpm

Z_1 = Number of teeth on driver,

Z_2 = Number of teeth on intermediate gear,

Z_3 = Number of teeth on driven or follower

Since the driving gear 1 is in mesh with the intermediate gear 2,therefore the speed ratio for these two gear is

$$\frac{N_1}{N_2} = \frac{Z_2}{Z_1}$$

The intermediate gear 2 is in mesh with the driven gear 3,therefor the speed ratio for these two gear is

$$\frac{N_2}{N_3} = \frac{Z_3}{Z_2}$$

So the total speed ratio for this simple gear train is obtain by multiplying gear train 1and 2 with gear 2 and 3. The full equation can be write as:

$$\frac{N_1}{N_2} \text{x} \frac{N_2}{N_3} = \frac{Z_2}{Z_1} \text{x} \frac{Z_3}{Z_2} \text{ or } \boldsymbol{\frac{N_1}{N_3} = \frac{Z_3}{Z_1}}$$

From above equation it also show of Law of gearing can be obtained:

$$\frac{\omega_1}{\omega_2} = \frac{N_1}{N_2} = \frac{d_2}{d_1} = \frac{Z_2}{Z_1}$$

Where,

ω = angular velocity, rad/s

N = angular velocity, rpm

d = pitch/gear diameter, mm

Z = number of gear teeth.

7.6.2 COMPOUND GEAR TRAIN

When there is more than one gear on a shaft axis, it is called a compound train of gear. The idle gears do not affect the speed ratio in a simple gear train. It works oppositely in compound gear train. These idle gear gears are useful in compound gear train because to offset the position between the driver and the driven gear.

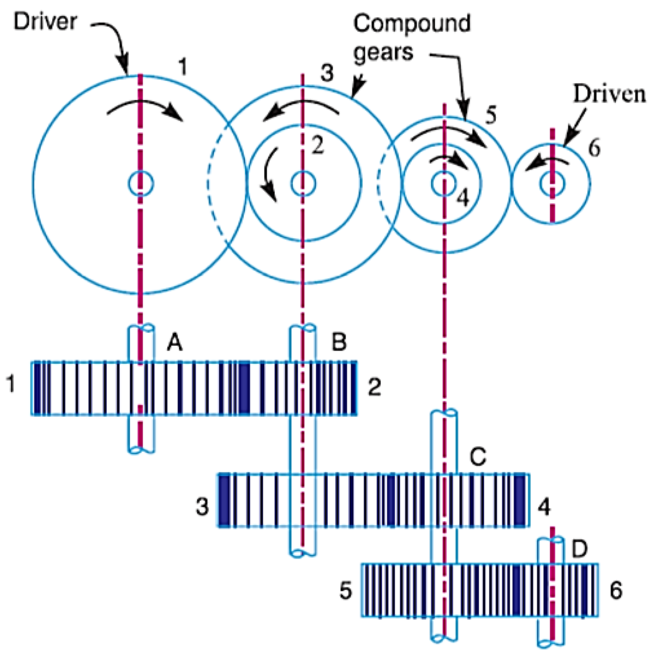


FIGURE 7.12: COMPOUND GEAR TRAIN.

In a compound train of gears, as shown in Figure 7.12, the gear 1 is the driving gear mounted on shaft A, gears 2 and 3 are compound gears which are mounted on shaft B. The gears 4 and 5 are also compound gears which are mounted on shaft C and the gear 6 is the driven gear mounted on shaft D. Let assign:

N_1 = Speed of driving gear 1, rpm

N_2 to N_6= Speed of respective gear, rpm

Z_1 = Number of teeth on driving gear 1,

Z_2 to Z_6= Number of teeth on respective gear,

Since gear 1 is in meshing with gear 2, therefore its speed ratio can be write as

$$\frac{N_1}{N_2} = \frac{Z_2}{Z_1}$$

Similarly, for gears 3 and 4, speed ratio is

$$\frac{N_3}{N_4} = \frac{Z_4}{Z_3}$$

And for gears 5 and 6, speed ratio is

$$\frac{N_5}{N_6} = \frac{Z_6}{Z_5}$$

The speed ratio of compound gear train is obtained by multiplying the equations set of gear train 1 and 2, gear train 3 and 4, and gear train 5 and 6.

$$\frac{N_1}{N_2} \text{x} \frac{N_3}{N_4} \text{x} \frac{N_5}{N_6} = \frac{Z_2}{Z_1} \text{x} \frac{Z_4}{Z_3} \text{x} \frac{Z_6}{Z_5}$$

Since gear 2 and 3 are mounted on one shaft B, therefore $n_2 = n_3$. Similarly gears 4 and 5 are mounted on shaft C, therefore $n_4 = n_5$. Then,

$$\frac{N_1}{N_6} = \frac{Z_2}{Z_1} \text{x} \frac{Z_4}{Z_3} \text{x} \frac{Z_6}{Z_5}$$

The advantage of a compound train over a simple gear train is compound gear train has much larger speed reduction from the first shaft to the last shaft can be obtained with small gears. In a simple gear train, to give a large speed reduction is by enlarge the last gear has to be very large. Usually for a speed reduction in excess of 7 to 1, a simple train is not preferred and a compound train or worm gearing is employed.

7.6.3 REVERTED GEAR TRAIN

When the axes of the first gear (*i.e*. first driver) and the last gear (*i.e*. last driven or follower) are co-axial, then the gear train is known as ***reverted gear* train** as shown in Fig. 7.12.

As can see on figure 7.13, gear 1 drives the gear 2 in the opposite direction. Since the gears 2 and 3 are mounted on the same shaft, therefore they form a compound gear and the gear 3 will rotate in the same direction as that of gear 2. The gear 3 which drives or rotates the gear 4 which is the last driven or follower in the same direction as that of gear 1. Thus we see that in a reverted gear train, the motion of the first gear and the last gear is ***like*.**

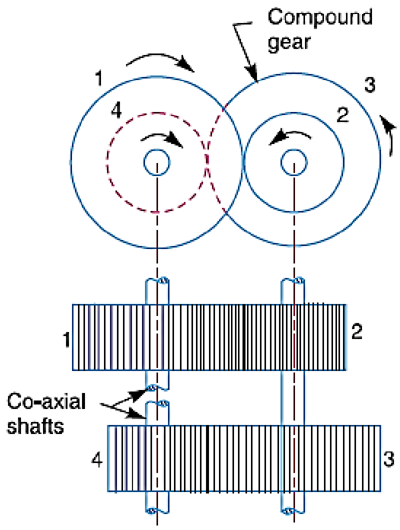


FIGURE 7.13: REVERTED GEAR TRAIN

N_1 = Speed of driving gear 1, rpm

N_2 to N_4= Speed of respective gear, rpm.

Z_1 = Number of teeth on driving gear 1,

Z_2 to Z_4= Number of teeth on respective gear.

r_1 = Pitch circle radius of gear 1,

r_2 to r_4= Pitch circle radius of respective gear.

Since

$$\text{Speed ratio} = \frac{Product\ of\ No.\ of\ teeth\ on\ drivens}{Product\ of\ No.\ of\ teeth\ on\ drivers}$$

then

$$\frac{N_1}{N_4} = \frac{Z_2}{Z_1} x \frac{Z_4}{Z_3}$$

Since the distance between the centers of the shaft of gears 1 and 2 as well as gear 3 and 4 is same, therefore

The radius of gear in reverted gear train can be write as

$$\mathrm{r}_1 + \mathrm{r}_2 = \mathrm{r}_3 + \mathrm{r}_4$$

And number of teeth for reverted gear train as

$$\mathrm{Z}_1 + \mathrm{Z}_2 = \mathrm{Z}_3 + \mathrm{Z}_4$$

7.7 GEAR EFFICIENCY CALCULATION

Efficiency is the ratio between input and output. Then the power efficiency of gear train can be written as;

$$\boldsymbol{efficicency}\ \% = \frac{\boldsymbol{power\ output}}{\boldsymbol{power\ input}} \boldsymbol{x\ 100\%}$$

Power and torque

Power is the produce of torque, T, Nm with angular velocity, ω, rads^{-1}. The power transmitted by a torque applied to a gear shaft and rotating at, *n,* rev/min is given,

$$Power, P = \frac{2\pi NT}{60}$$

Where T= torque, Nm

N = rotation speed, RPM

For ideal gear system, the input and the output powers must same, so it can write as

$$\frac{2\pi N_1 T_1}{60} = \frac{2\pi N_2 T_2}{60}$$

These can rewrite again or simplified as

$$N_1 T_1 = N_2 T_2$$

Then rearrange in the term ratio of rotation speed, *n* and torque, T , as;

$$\frac{T_2}{T_1} = \frac{N_1}{N_2}$$

Since the function of gear is to increase or decrease the torque at the end of the gear train. It will follow with the reducing speed, the torque is increase and vice versa as referring at above equation.

Example 1:

A simple train has 3 gears. Gear A is the input and has 50 teeth, rotates at 1500 RPM and torque 12 Nm. Gear C is the output and has 150 teeth. Calculate the gear ratio, input power and the output speed.

Solution:

a) Gear ratio,

$$gear\ ratio = \frac{N_A}{N_C} = \frac{Z_C}{Z_A} = \frac{150}{50} = 3$$

Given,
Z_A = 50
N_A = 1500RPM
T_A = 12Nm
Z_C = 150

b) The input power

$$P = \frac{2\pi N_A T_A}{60} = \frac{2\pi(1500)(12)}{60} = 1885W$$

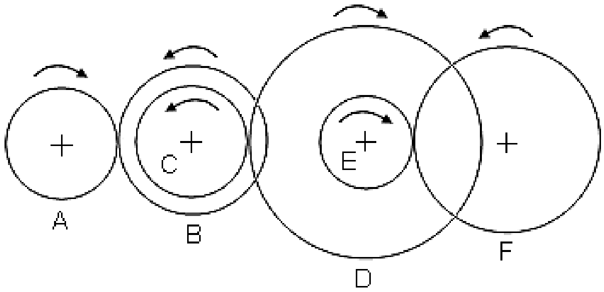
c) The output speed

$$N_C = \frac{N_A}{3} = \frac{1500}{3} = 500RPM$$

Example 2:

The gear system in a machine tool is shown in below. The motor shaft is connected to the gear A and rotates at 975 ppm. The gear wheels B, C, D and E are set on the parallel shaft rotating together. The last gear F is set on the output shaft. What is the speed of the F gear? The number of teeth per gear is as follows:

Gear	A	B	C	D	E	F
Teeth No	20	50	25	75	26	65



Solution:

$$\frac{N_A}{N_B} \text{x} \frac{N_C}{N_D} \text{x} \frac{N_E}{N_F} = \frac{Z_B}{Z_A} \text{ x } \frac{Z_D}{Z_C} \text{x} \frac{Z_F}{Z_E}$$

$$since\ , n_B = n_C\ ,\ and, n_D = n_E$$

Given,
N_A = 975 rpm

Then

$$\frac{N_A}{\cancel{N_B}} \text{x} \frac{\cancel{N_C}}{\cancel{N_D}} \text{x} \frac{\cancel{N_E}}{N_F} = \frac{Z_B}{Z_A} \text{ x } \frac{Z_D}{Z_C} \text{x} \frac{Z_F}{Z_E}$$

Simplified

$$N_F = \frac{Z_A}{Z_B} \text{ x } \frac{Z_C}{Z_D} \text{x} \frac{Z_E}{Z_F} \text{ x } N_A$$

$$N_F = \frac{20}{50} \text{ x } \frac{25}{75} \text{x} \frac{26}{65} \text{ x } 975 = \mathbf{52\ rpm}$$

1. How are the gears classified and state 5 (five) various terms used in spur gear terminology?
2. Mention four important types of gears and discuss their applications usage.
3. What condition must be satisfied in order that a pair of spur gears may have a constant velcoity ratio?
4. How to specified the size of gear?
5. Calculate the center distance between 2 gear where gear 1 have 36 and gear 2 have 60 tooth. Both gear wheel use 24 inch pitch gear drive.
6. What is advantages compund gear train compare to simple gear train?
7. A simple gear train has an input speed of 1500 RPM clockwise and an output speed of 300 RPM anticlockwise. The input power is 20 kW and the efficiency is 70%. Determine the gear ratio, input torque, output power, and output torque of the gear train.
8. Two parallel shafts, about 600 mm apart are to be connected by spur gears. One shaft is to run at 360 RPM and the other at 120 RPM. Design the gears, if the circular pitch is to be 25 mm.

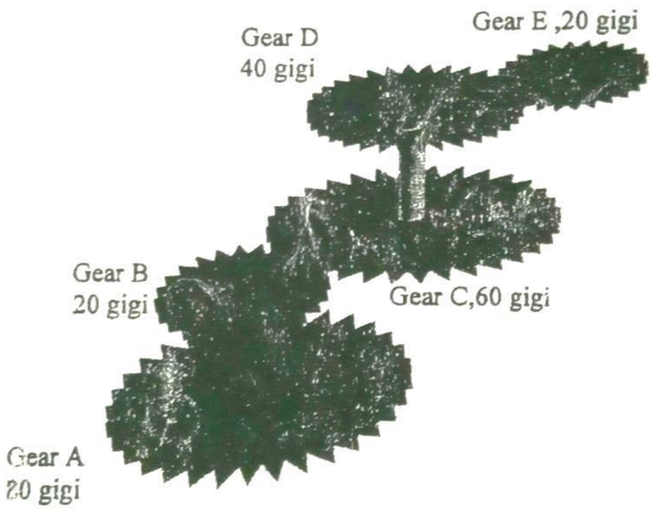
Previous Years' *Question Papers*

No.6

a. A gear in Fig below run at 1200rpm and transmits torque to gear C through gear B. The gear B is idler gear. The torque will be transmit from gear C to gear D through a shaft and then to gear E.

i) Write a gear ratio for a gear A and C. 4M

ii) Calculate speeds of gear E. 3M

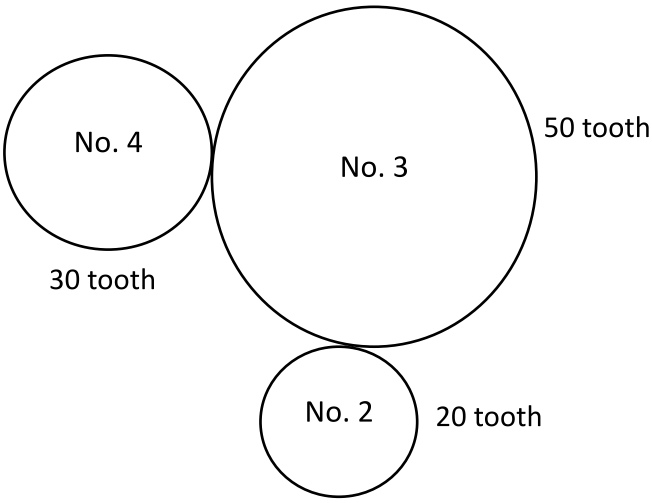


b. Pinion no. 2 in Fig. below run at 1750 rpm and transmits 2.5 kW to idler gear no.3. The teeth have a module of m 2.5 mm. Calculate;

i) Pitch diameter of gear no. 2,3 and 4 3M

ii) The speed of gear no. 4 3M

iii) Tork at gear no 2, 3 and no. 4 7M



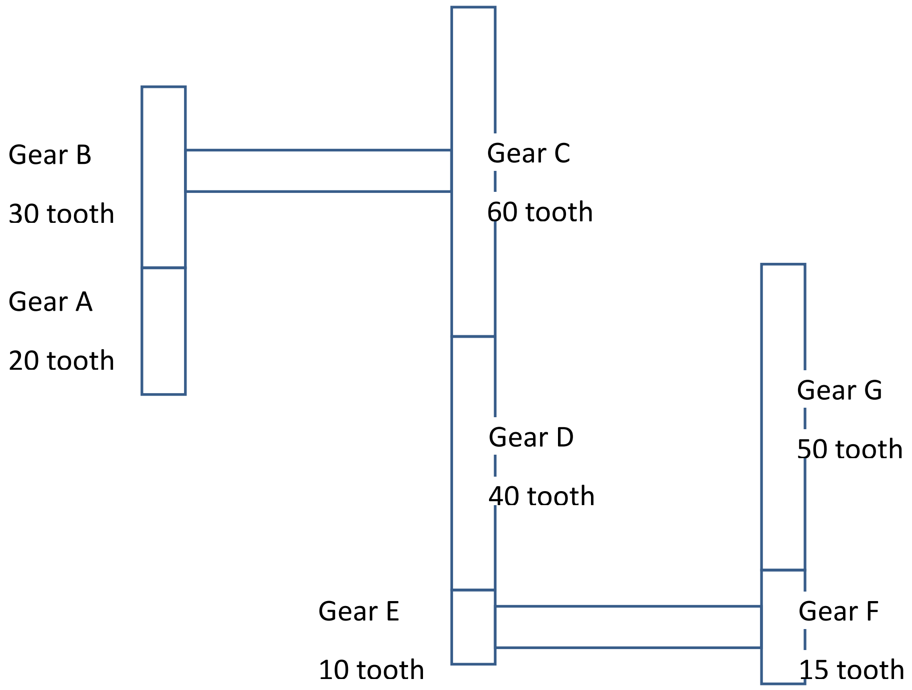
DIS 16

a. Define speed in gear ratio 2M

b. A gear A in figure below run at 3200rpm and transmits torque through gear B,C,D,E,F and G

i) Write a gear ratio for a gear A and E 6M

ii) Calculate speeds of gear G 4M



JUN 16

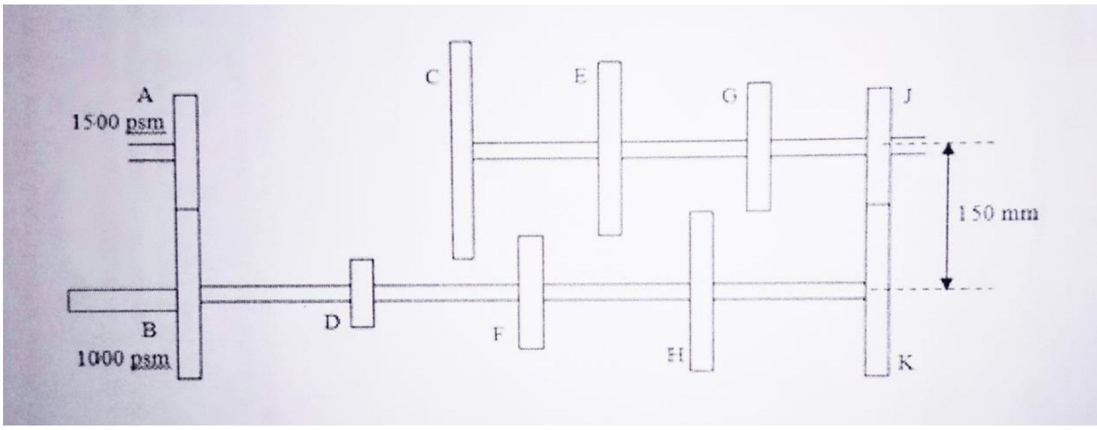
a. A 17-tooth spur pinion has a diameter pitch of 8 teeth/in, runs at 1120 rpm, and drives a gear at a speed of 544 rpm. Find ;-
   i) Gear ratio
   ii) The number of teeth on the gear (Z=DP)
   iii) The theoretical center-to-center distance

10M

b. Explain **THREE (3)** types of gear network. 6M

c. Explain worm gear 6M

d. A gear box need to be arranged at four speeds, according to the geometric series which one of them must be direct drive as shown in figure below. The drive shaft A transmits 25kW of power at the speed of 1500rpm and the speed of support shaft B is 1000 rpm. The speed of driven shaft at lowest gear which there is the couple of gear C and D is 200 rpm. The distance between the drive and support shaft is 150mm. The module for all gears is 6. Calculate the correspondent number of teeth for A, B, C, D, J and K.



8M

CHAPTER 8: POWER SCREW

OBJECTIVES:

In the end of this chapter, student should know to:

- Explain the principle of power screw
- Explain the application of Screw thread.

8.1 INTRODUCTION

A power screw is a drive used in machinery to convert a rotary motion into a linear motion for power transmission. It produces uniform motion and the design of the power screw may be such that

i. Either the screw or the nut is held at rest and the other member rotates as it moves axially. A typical example of this is a screw clamp.
ii. Either the screw or the nut rotates but does not move axially.

A typical example for this is a press. Other applications of power screws are jack screws, lead screws of a lathe, screws for vices, presses etc. Power screw normally uses square threads but ACME or Buttress threads may also be used. Power screws should be designed for smooth and noiseless transmission of power with an ability to carry heavy loads with high efficiency.

8.2 DEFINITION

Power screw is a drive used in machinery to convert a rotary motion into linear motion for power transmission.

8.2.1 FUNCTION OF POWER SCREW

Power screws are used for providing linear motion in a smooth uniform manner. They are linear actuators that transform rotary motion into linear motion to transmit power from one place to another place.

8.3 TYPES OF SCREW THREADS USED FOR POWER SCREWS

Following are the three types of screw threads mostly used for power screws :

1. Square thread.

A square thread, as shown in Fig. 8.1, is adapted for the transmission of

power in either direction.

- Strongest thread.
- No radial load.
- Hard to manufacture.
- Square threads have a much higher intrinsic efficiency than acme threads.
- Due to the lack of a thread angle there is no radial pressure, or bursting pressure, on the nut. This also increases the nut life.
- This thread is difficult in machining
- The square thread form is a common screw thread form, used in high load applications such as lead screws and jack screws.
- It gets its name from the square cross-section of the thread.
- It is the lowest friction and most efficient thread form, but it is difficult to fabricate.

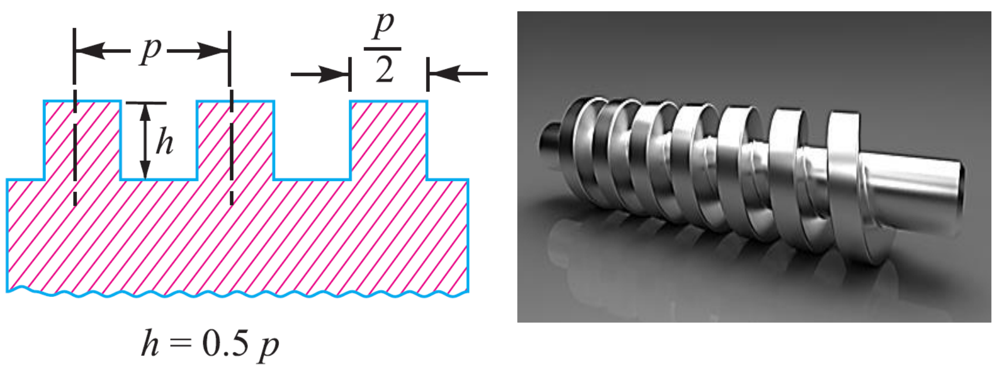


FIGURE 8.1: *SQUARE THREAD*

2. Acme thread

- The Acme thread form has a 29° thread angle with a thread height half of the pitch.
- The apex and valley are flat.
- This shape is easier to machine than is a square thread.
- The tooth shape also has a wider base which means it is stronger than a similarly sized square thread.
- This thread form also allows for the use of a split nut, which can compensate for nut wear.
- 29° included angle.
- Easier to manufacture.
- Common choice for loading in both directions.
- Faster cutting.
- Longer tool life.

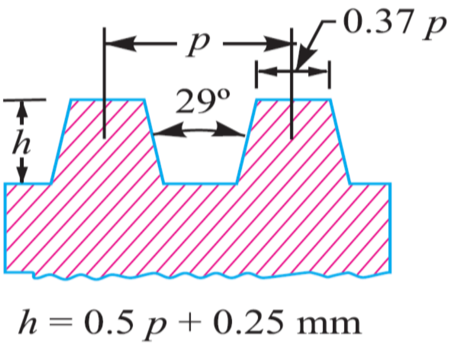


FIGURE 8.2: ACME THREAD

3. Buttress thread

- In machinery, the buttress thread form is designed to handle extremely high axial thrust in one direction.
- The load-bearing thread face is perpendicular to the screw axis or at a slight slant (usually not greater than 7°)
- The other face is slanted at 45°.
- Great strength.
- Only unidirectional loading.

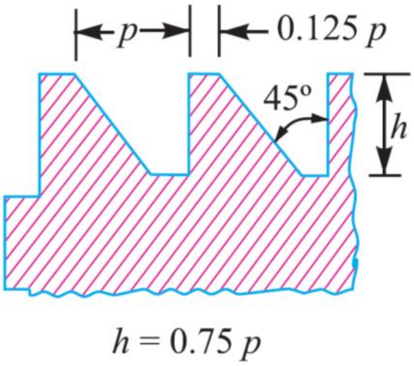


FIGURE 8.3: BUTTRESS THREAD

4. Unified/Metric Thread

The unified or metric screw threads are the worldwide most commonly used type of general-purpose screw thread. They were one of the first international standards agreed when the International Organization for Standardization was set up in 1947.

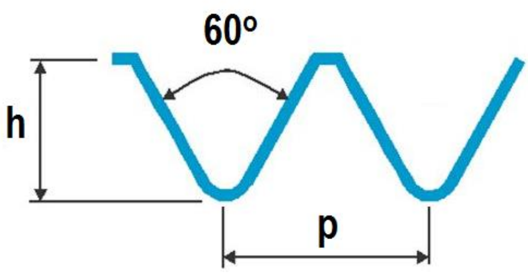


FIGURE 8.4: UNIFIED/METRIC THREAD

8.4 FORCES ACTING ON THE SCREW THREAD

A square thread power screw with a single start is shown in figure 1.

In order to analyze the mechanics of the power screw we need to consider two cases:

i. Raising the load
ii. Lowering the load.

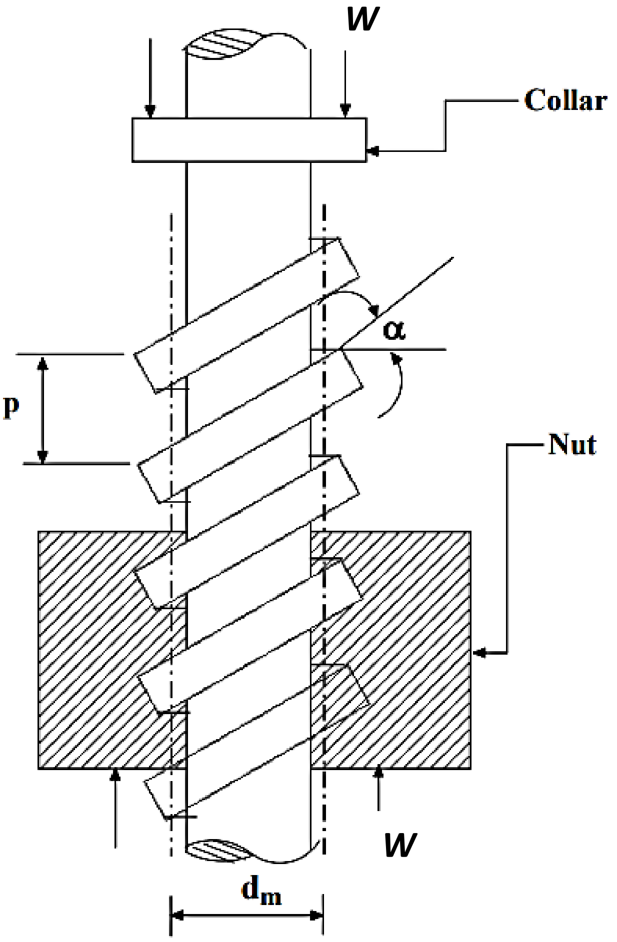


FIGURE 8.6: A SQUARE THREAD OF POWER SCREW

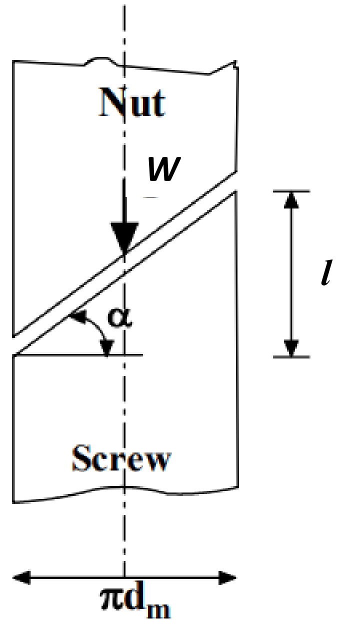


FIGURE 8.5: DEVELOPMENT OF A SINGLE THREAD

Where,

p = pitch, mm

n = number of lead start

l_S = lead, mm

α = helix/lead angle

d_m = mean diameter of thread, mm

P = axial load, N

W = load to be lifted/lowered, N

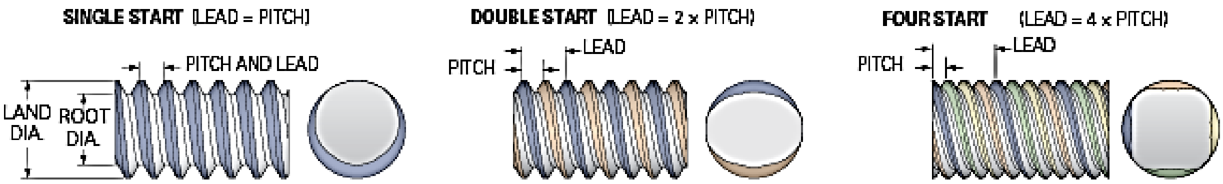
$$\tan\alpha = \frac{l_s}{\pi d_m}$$

8.5 LEAD START

The power screws with multiple threads such as double, triple etc. are employed when it is desired to secure a large lead with fine threads or high efficiency. Such type of threads are usually found in high speed actuators. Increasing the number of starts increases the lead thus increasing the translational velocity of the nut for a given fixed angular velocity of the screw.

Number of screw starts – The number of independent threads on the screw shaft; example one, two or four in the figure below.

$$\text{Lead, } l_S = \text{no. of starts x Pitch (p)}$$



Torque required to lower @ lifted a load using handle with length, l :

$$E \times l_h = T = \frac{1}{2} P d_m$$

Where,

E = Force required at the end of a lever@handle, N

l_h = Length of lever@handle, m

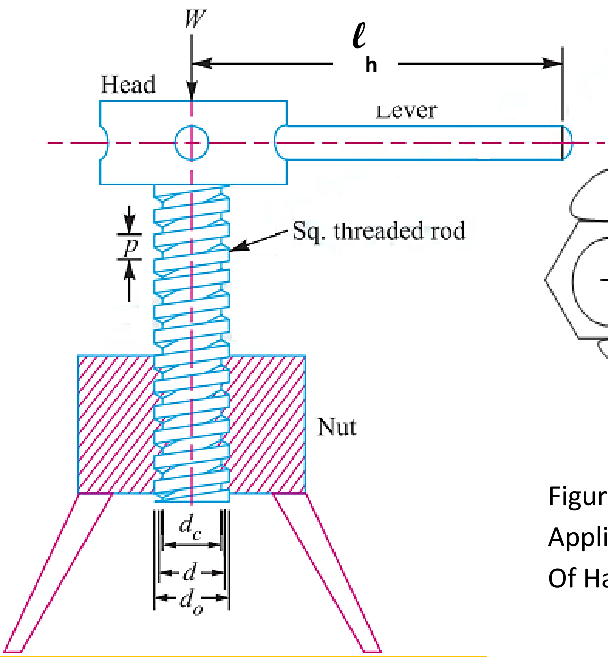


FIGURE 8.7: A JACK SCREW

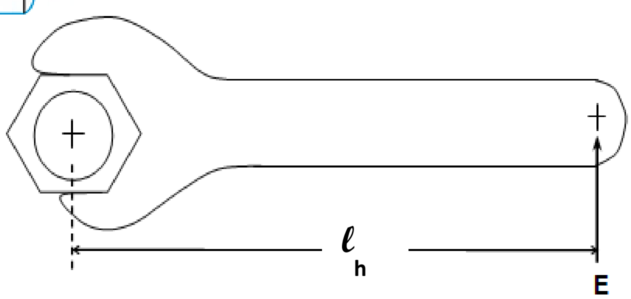
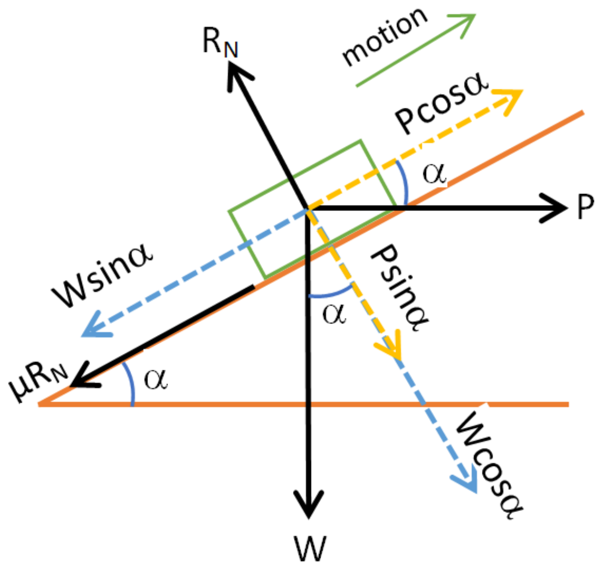


Figure 8.8: A Screw Tightened by A The Horizontal Force Applied, E And Acts At The End Of Spanner With Length Of Handle, l_h

Since the principle, on which a screw jack works is similar to that of an inclined plane, therefore the force applied on the circumference of a screw jack may be considered to be horizontal as shown in Fig.



$(F = \mu R_N)$

Resolving the forces along the plane,

$P \cos \alpha = W \sin \alpha + F = W \sin \alpha + \mu RN$ ...(i)

$R_N = P \sin \alpha + W \cos \alpha$ ....(ii)

Substituting RN in equation (i),

$P \cos \alpha = W \sin \alpha + \mu (P \sin \alpha + W \cos \alpha)$

$= W \sin \alpha + \mu P \sin \alpha + \mu W \cos \alpha$

or $P \cos \alpha - \mu P \sin \alpha = W \sin \alpha + \mu W \cos \alpha$

or $P (\cos \alpha - \mu \sin \alpha) = W (\sin \alpha + \mu \cos \alpha)$

$$\therefore \qquad P = W\ x\ \frac{(sin\alpha+\mu cos\alpha)}{(cos\alpha-\mu sin\alpha)}$$

Substituting the value of $\mu = \tan\phi$ in the above equation, we get

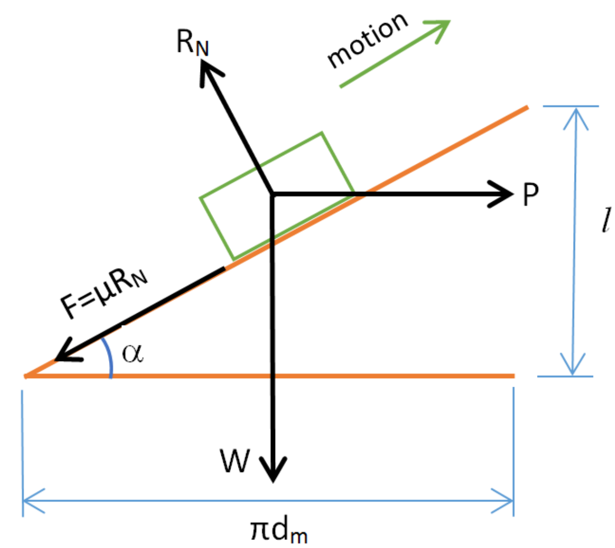
$$\text{or} \qquad P = W\ x\ \frac{(sin\alpha+\tan\phi cos\alpha)}{(cos\alpha-\tan\phi sin\alpha)}$$

Multiplying the numerator and denominator by $\cos\phi$, we have

$$P = W\ x\ \frac{(sin\alpha \cos\phi + sin\phi\ cos\alpha)}{(cos\alpha \cos\phi - sin\phi\ sin\alpha)}$$

$$= W\ x\ \frac{\sin(\alpha+\phi)}{cos(\alpha+\phi)} = W \tan(\alpha + \phi)$$

8.6 TORQUE,T OF RAISING A LOAD @ TIGHTEN A SCREW



Where,

l = lead, mm

α = helix/lead angle ,°

d_m = mean diameter of thread, mm

P = axial load, N

W = load to be lifted/lowered, N

μN = Friction force, N

μ = Coefficient of friction, between the screw and nut

ϕ = Friction angle, °

$$\boxed{\mu = \tan\phi}$$

Forces at the contact surface for raising the load

$$P_{raise} = W\tan(\phi + \alpha)$$

$$T_{raise} = \frac{Wd_m}{2}\tan(\phi + \alpha)$$

8.6.1 EFFICIENCY OF SCREW WITH SQUARE THREAD

The efficiency of a power screw is defined as the ratio between the torque required to drive the load without friction with the torque required to consider the existence of friction.

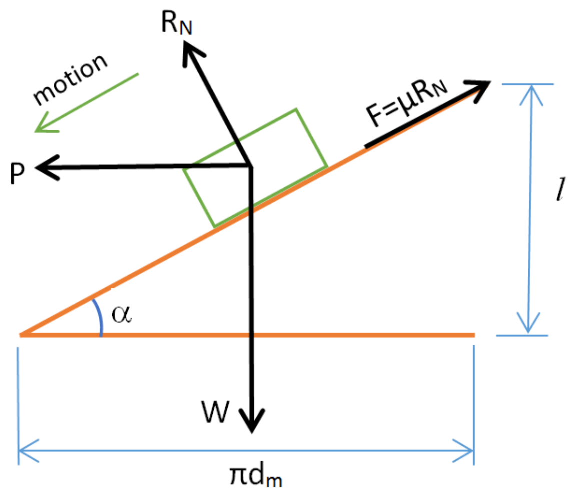
$$\text{Efficiency, } \eta_{raise} = \frac{\tan\alpha}{\tan(\alpha+\phi)} \times 100\%$$

$$\text{Max. efficiency, } \eta_{max} = \frac{1-\sin\phi}{1+\sin\phi} \times 100\%$$

8.7 TORQUE,T OF LOWERING A LOAD @ LOOSEN A SCREW.

i) **if $\phi > \alpha$, then T > 0**

- Torque required to lower the load will be positive,
- Need an effort to lower the load (**cannot come down on its own**).
- This screw known as **self locking screw**



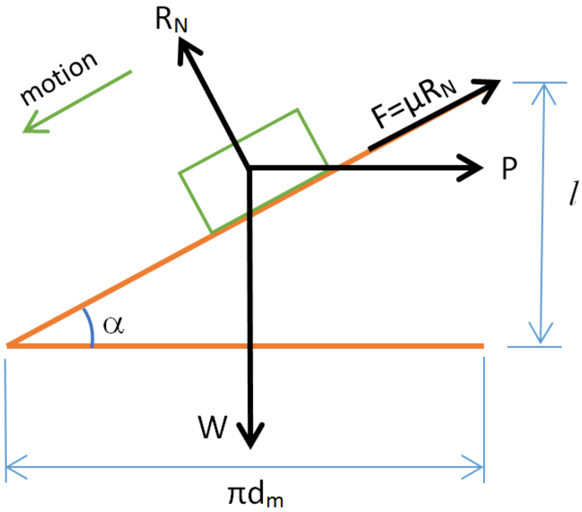
$$P_{low} = W\tan(\phi - \alpha)$$

$$T_{low} = \frac{Wd_m}{2}\tan(\phi - \alpha)$$

$$\text{Efficiency, } \eta = \frac{\tan(\phi-\alpha)}{\tan\alpha} \times 100\%$$

ii) if $\phi < \alpha$, then T < 0

- Torque required to lower the load will be **negative**.
- In other words, the load will start moving downward without the application of any torque (**load will lower itself of its own**).
- Such a condition is known as **over hauling of screws**.



$$P_{low} = W \tan(\alpha - \phi)$$

$$T_{low} = \frac{W d_m}{2} \tan(\alpha - \phi)$$

$$\text{Efficiency, } \eta_{low} = \frac{\tan(\alpha - \phi)}{\tan \alpha} \times 100\%$$

8.8 **ADVANTAGES AND DISADVANTAGES**

Power screws offers following advantages:

- Large load carrying capacity
- Simple to design
- Compact construction
- Easy to manufacture
- A load of 15 KN can be raised by applying small effort as 400 N
- It gives smooth and noiseless service without any maintenance
- Self-locking property

The disadvantages of power screw are as follows:

- Very poor efficiency as low as 40%
- It can be used for intermittent motion
- High friction in threads causes rapid wear

Application of Power Screw

- Jack screws
- Lead screws of a lathe
- Screw for vices
- Presses
- C-clamps
- Instron material testing machines
- Machine tools (for positioning of table)

Example 1:

A power screw having square threads of 5 cm means diameter and pitches 1.25mm is operated by a 50 cm long hand lever. Coefficient of friction at the threads is 0.1. Determine the effort needed to be applied at the end of the lever to lift a load of 20 kN. 6 M

Solution:

Given,
d_m = 5cm=50mm
p = 1.25mm
l_H = 50mm=500mm
μ = 0.1
E = ?
W = 20kN

lifting a load @ tighten a screw

$$E \times l_h = T = \frac{1}{2} P d_m$$

$$P_{raise} = W \tan(\phi + \alpha)$$

$$\tan\alpha = \frac{l_S}{\pi d_m}$$

l = p=1.25mm (assuming start=1)

$$\tan\alpha = \frac{1.25}{\pi(50)}, \ \alpha = 0.456^o$$

$$\phi = \tan^{-1}\mu = \tan^{-1}0.1 = 5.7106^o$$

Thus, $$P_{raise} = 20k \tan(5.7106 + 0.456) = 2160.9N$$

$$\therefore \ E \times 500 = \frac{1}{2}(2160.9)(50)$$

$\therefore$ E = 108.045N#

Example 2:

A double-start thread, diameter and pitch of 30mm and 8mm is used with an axial load of 2 kN. The coefficient friction is 0.125, determine;

a) Torque required to continuously raise and its efficiency.
b) Load required to be lifted.
c) Torque required to continuously lower and its efficiency.

20M

Solution:

Given,
start = 2
d_m = 30mm
p = 8mm
P = 2kN
μ = 0.125

a) T_{raise} = ?, η = ?

$\mathbf{T_{raise}} = \frac{1}{2}Pd_m = \frac{1}{2}(2k)(30)$ = 30kNmm = **30Nm#**

$\eta_{raise} = \frac{\tan\alpha}{\tan(\alpha+\phi)}$ x100%

$\tan\alpha = \frac{l_s}{\pi d_m}$, $\quad l_s$ = start x p = 2 x 8 = 16mm

$= \frac{16}{\pi(30)}$, $\alpha = 9.635^o$

$\phi = \tan^{-1}\mu = \tan^{-1}0.125 = 7.125^o$

$\therefore$ $\boldsymbol{\eta_{raise}} = \frac{\tan\alpha}{\tan(\alpha+\phi)}$ x100% = $\frac{\tan 9.635}{\tan(9.635+7.125)}$ x100% = **56.37%#**

b) W,

$P_{raise} = W\tan(\phi+\alpha)$

$2k = W\tan(7.125+9.635)$

$\therefore$W = 6641.072N = **6.641kN#**

c) T_{low} = ?, η = ?

$T_{low} = \frac{1}{2}P_{low}d_m$

when $\alpha > \phi$

$P_{low} = W\tan(\alpha-\phi) = 6641.072\tan(9.635-7.125)$

= 291.117N

$\therefore$ $\mathbf{T_{low}} = \frac{1}{2}(291.117)(30)$ = 4366.752Nmm = **4.367kNm#**

$\boldsymbol{\eta_{low}} = \frac{\tan(\alpha-\phi)}{\tan\alpha}$ x100% = $\frac{\tan(9.635-7.125)}{\tan 9.635}$ x100% = **25.82%#**

1. List the function of power screw. 2M
2. Name THREE (3) forms of power screw thread. 6M
3. Define "efficiency" of power screw. 3M
4. The mean diameter of a power screw square thread is 45 mm and 12 mm pitch is manage by a hand lever 600 mm length. The coefficient of friction for the thread is 0.15. Determine the correspondent force that should be apply at the end of the lever to raise 25 kN load. 6M

5. A simple screw jack has a square threaded screw whose mean diameter is 5 cm and pitch is 12.5 mm. If the coefficient of friction between the screw and nut is 0.13 and the jack must raise a weight 2500 kg, calculate;
   i. Torque for raise the load. 6M
   ii. Efficiency for raise the load 3M
   iii. Torque for lower the load 3M
   iv. Efficiency for lower the load 3M

Previous Years' Question Papers

Dis 16

Q4

a. Describe the definition and function of power Screw. 8M

b. Calculate torque required to overcome thread friction if axial load of 8 kN acts on a power screw having double start square threads of 30 mm mean diameter, Friction angle and lead angle are 7^o and 9^o respectively. 7M

JUN 16

Q4

a. Identify **THREE (3)** types power screw thread. 6M

b. Sketch any **TWO (2)** types of power screw thread diagram based on question 4(c). 7M

Dis 15

Q4

a. Describe the functions of power screw. 3M

Jun 15

Q6

a. Describe the function of power screws. 4M

b. Sketch and label **THREE (3)** different types of screw thread. 12M

Dis 14

Q6

a. State the function of power screws. 4M

b. Sketch and label THREE (3) different types of screw thread. 12M

REFERENCES

Budynas, R. G. (2011). *Shigley's Mechanical Engineering Design 9TH ED.* NEW YORK: McGraw-Hill.

Gupta, B.V.R. (2010). Theory of Machines: Kinematics and Dynamics. I. K. International Pvt Ltd.

http://144.162.92.233/faculty/djones/todays_class/introduction_to_brake_syste.pdf

http://highered.mheducation.com/sites/dl/free/0070591202/303462/89FrictionClutches.pdf

http://mechteacher.com/universal-joint/

http://seabeemagazine.navylive.dodlive.mil/files/2014/05/14264A-Construction-Mechanic-Basic-Chapters-11.pdf

http://techminy.com/oldham-coupling

http://umpir.ump.edu.my/297/1/3407.pdf

http://www.ignou.ac.in/upload/Unit-6-60.pdf

https://books.google.com.my/books?id=t6W0ib33kWMC&printsec=frontcover#v=onepage&q&f=false

https://industrialdesigninfo.wordpress.com/tag/clamp-or-split-muff-type-coupling/

https://www.cs.cmu.edu/~rapidproto/mechanisms/chpt6.html

https://www.motioncontroltips.com/bellows-couplings/

https://www.quora.com/What-are-some-universal-coupling-applications

https://www.reliance.co.uk/catalogue/precision-couplings/bellows-couplings/

https://www.youtube.com/watch?v=qQZgyriXmAk

Khurmi, R. S. & Gupta, J. K. (2005). Machine Design. S. Chand & Company Ltd.

Khurmi, R. S. & Gupta, J. K. (2006). A Text Book of Theory of Machines. S. Chand & Company Ltd.

Khurmi, R. S. & Gupta, J.K. (2006). Theory of Machines. New Delhi: Rajendra Ravindra (Pvt) Ltd.

Khurmi, R.S. & Gupta, J. K. (2016). Theory Of Machines (2005 ed.). Eurasia Publishing House. Retrieved from http://engineeringstudymaterial.net/ebook/theory-of-machines-rs-khurmi-jk-gupta/

Khurmi, R.S. & Gupta, J.K. (2007). Theory of Machines Fourteenth Revised Edition. Eurasia Publishing House.

Khurmi, R.S. (2013). A Textbook of Engineering Mechanics. S. Chand & Company Ltd.

KUNDUR, N. F. (2009). *POWER TRANSMISSION.* Port Dickson: POLITEKNIK PORT DICKSON.

Martin, G.H. (1982). Kinematics and Dynamics of Machines (2nd Edition). Mc Graw Hill Education.

Rothbart, H. (2006). *Mechanical Design Handbook, Measurement, Analysis, and Control of Dynamic System.* NEW YORK: McGRAW-HILL.

www.acorn-ind.co.uk/power-transmission/industrial-chains/silent-chains/

www.slideshare.net/AAhadNoohani/clamp-or-compression-coupling

www.theengineer.co.uk/issues/14-january-2000/spiders-are-key-to-jaw-coupling-performance/

e ISBN 978-967-2742-33-3

9 789672 742333

POLITEKNIK NILAI
KOMPLEKS PENDIDIKAN BANDAR ENSTEK
71760 BANDAR ENSTEK
NEGERI SEMBILAN