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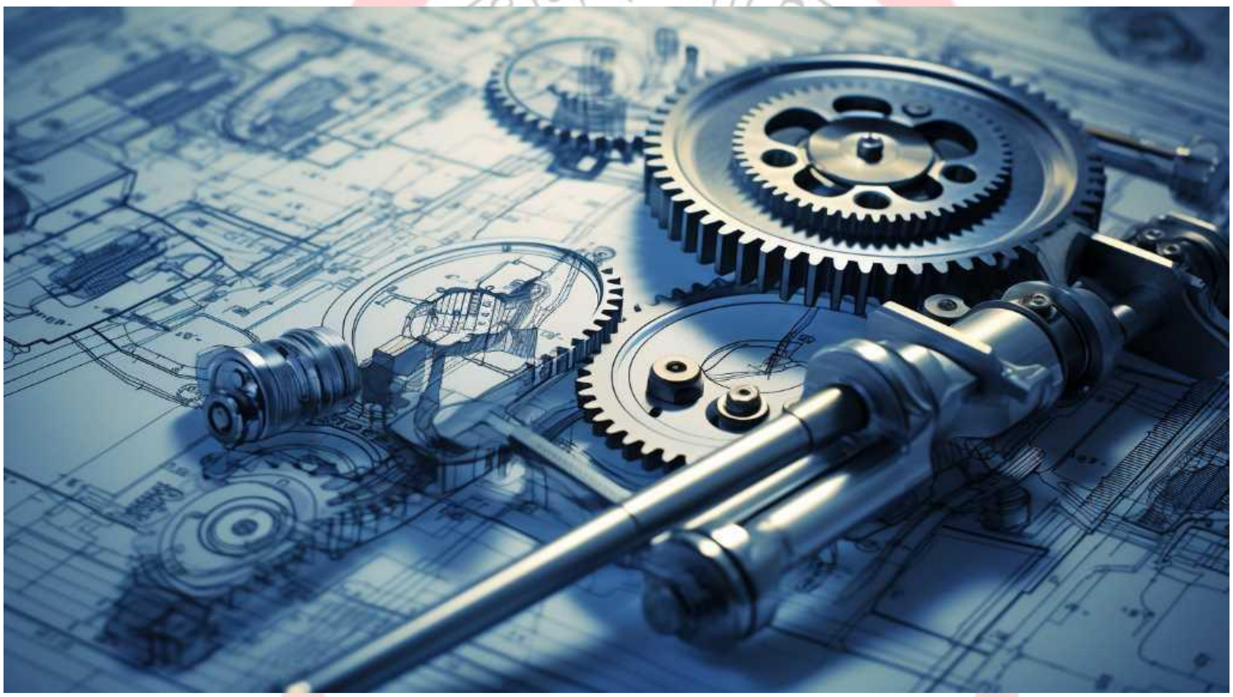
LAB MANUAL

Machine Design-1 Lab

Course Code: ME-342



University of Global Village (UGV), Barishal

Department of Mechanical Engineering

Prepared by
Md. Naeem Hosen Hredoy
Lab Instructor
Department of Mechanical

Course Name: Machine Design-1 Lab

Course Teacher: Md.Naeem Hosen Hredoy	CREDIT:01
Course Code: ME-342	TOTAL MARKS:50
TOTAL Class : 17 Nos	CIE MARKS: 30
Class Hours : 85 Hours	SEE MARKS: 20

Course Learning Outcomes (CLOs): After completing this course successfully, the students will be able to-

CLO 1. **Understand** Comprehend the various components of a machine and their functions.

CLO 2. **Design** Acquire the ability to design a machine or a machine component for a specific task.

CLO 3. **Apply** theoretical knowledge to solve real-world problems.

CLO 4. **Calculate** Develop the ability to solve mathematical problems required for machine design.

CLO 5. **Analyze** the performance and efficiency of a machine.

CLO 6. **Generate** to create new ideas and designs.

Sr No	Content of Course	Hours	CLOs
1	Design of shaft	05	CLO3
2	Design of rigid coupling	10	CLO2,CLO3
3	Design of flexible coupling	10	CLO1,CLO6
4	Design of helical spring	10	CLO2,CLO3
5	Design of leaf spring	10	CLO5,CLO7
6	Design of brake	10	CLO4,CLO5
7	Design of clutches	10	CLO3
8	Design of Journal Bearing	10	CLO1,CLO3
9	Selection of rolling element bearing	05	CLO4,CLO6
10	Design of Spur Gear	05	CLO1,CLO5

Reference Book :

1. "Design of Machine Elements" - V.B. Bhandari

2. "Machine Design" - R.S. Khurmi & J.K. Gupta

3. "Mechanical Engineering Design" - J.E. Shigley & Charles R. Mischke

4. "Fundamentals of Machine Component Design" - Robert C. Juvinall & Kurt M. Marshek

5. "A Textbook of Machine Design" - P.C. Sharma & D.K. Aggarwal

6. "Machine Design Data Handbook" - K. Lingaiah

ASSESSMENT PATTERN

CIE- Continuous Internal

Evaluation

(20 Marks) SEE- Semester End
Examination (30 Marks)

SEE- Semester End Examination (50 Marks) (should be converted in actual marks (30)

Bloom's Category Cognitive	**Tests (20)**
Remember	**05**
Understand	**07**
Apply	**08**
Analyze	**07**
Evaluate	**08**
Create	**05**

Bloom's Category Psychomotor	**Practical Test (30)**
Imitation	10
Manipulation	5
Precision	5
Articulation	5
Naturalization	5

CIE- Continuous Internal Evaluation (40 Marks) (should be converted in actual marks (20)

Bloom's Category Marks (out of 60)	Lab Report (10)	Continuous lab performance (10)	Presentation & Viva (10)	External Participation in Curricular/Co-Curricular Activities (10)
Remember			02	Attendance 10
Understand	05	04	03	
Apply		02		
Analyze		02		
Evaluate	05	02		
Create			05	

Course plan specifying content, CLOs, teaching learning and assessment strategy mapped with CLOs

Week	Topic	Teaching-Learning Strategy	Assessment Strategy	Corresponding CLOs
1	Design of shaft	Lectures and Tutorials	Quiz, Written Exam	**CLO3**
2,3	Design of rigid coupling	Practical Demonstrations	Practical Assessments, Quiz	**CLO2,CLO3**
4,5	Design of flexible coupling	Assignments and Projects	Perrormance-Based Assessments	**CLO1,CLO6**
6,7	Design of helical spring	Showcase different welding techniques, safety procedures, and equipment usage.	Design Project	**CLO2,CLO3**
8,9	Design of leaf spring	Hands on Practice, discussion	Project, Hands on Practice	**CLO5,CLO7**
10,11	Design of brake	Experiment, Demonstration	Proper equipment setup and use	**CLO4,CLO5**
12,13	Design of clutches	Performance-Based Assessments	Assignment, Written, Quiz	**CLO3**
14,15	Design of Journal Bearing	Group Discussions,Problem-Based Learning	Class Participation	**CLO1,CLO3**
16	Selection of rolling element bearing	Design Simulations	Viva, Quiz	**CLO4,CLO6**
17	Design of Spur Gear	Problem-Solving Sessions	Laboratory Reports	**CLO1,CLO5**

List of Experiments

Note:

1. At least eight design problems are to be completed in the semester from the above list.

2. Two more designs of machine components are to be carried out as designed and set by the instructor as per the scope of the syllabus.

Experiment 1: Design of Shaft

1. AIM OF THE EXPERIMENT: Calculate the value of diameters of shaft for given loads and layout.
2. OVERVIEW OF SHAFTS

- A shaft is a rotating member usually of circular cross-section (solid or hollow), which transmits power and rotational motion.
- Machine elements such as gears, pulleys (sheaves), flywheels, clutches, and sprockets are mounted on the shaft and are used to transmit power from the driving device (motor or engine) through a machine.
- Press fit, keys, dowel, pins and splines are used to attach these machine elements on the shaft.
- The shaft rotates on rolling contact bearings or bush bearings.
- Various types of retaining rings, thrust bearings, grooves and steps in the shaft are used to take up axial loads and locate the rotating elements.
- Couplings are used to transmit power from drive shaft (e.g., motor) to the driven shaft (e.g., gearbox, wheels).

3. DESIGNING OF SHAFT

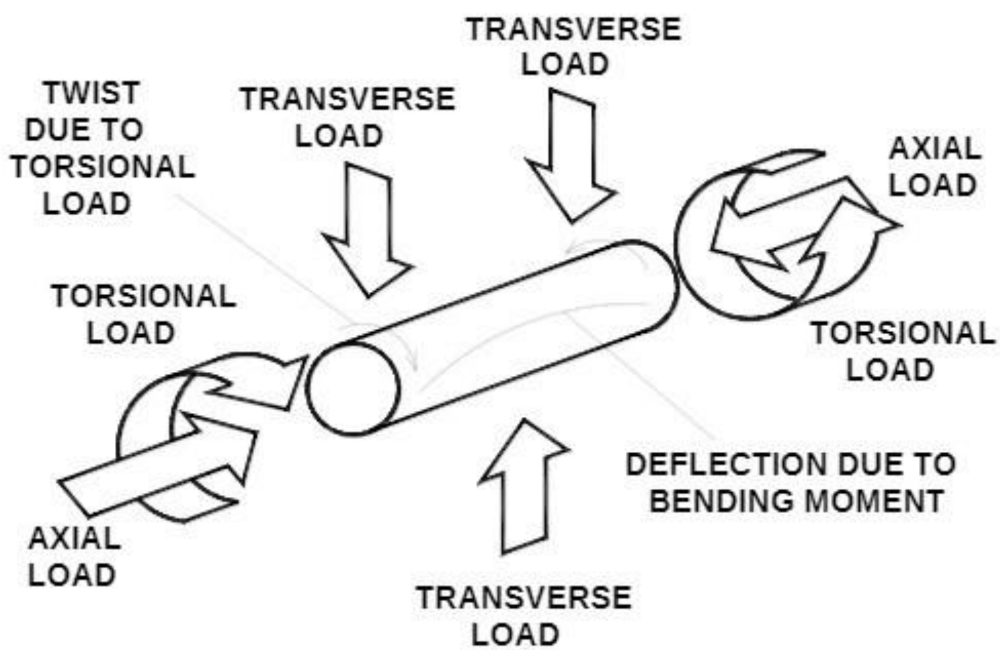
Designing shaft is to calculated value its diameters at critical section. The critical section is a section on the shaft which is highly stressed. The value stresses are calculated with the help of layout and free body diagram.

- The shafts may be designed on the basis of:
- 1. Strength, and 2. Rigidity and stiffness.
- In designing shafts on the basis of strength, the following cases may be considered :
- Shafts subjected to twisting moment or torque only,
- Shafts subjected to bending moment only,
- Shafts subjected to combined twisting and bending moments, and
- Shafts subjected to axial loads in addition to combined torsional and bending loads.

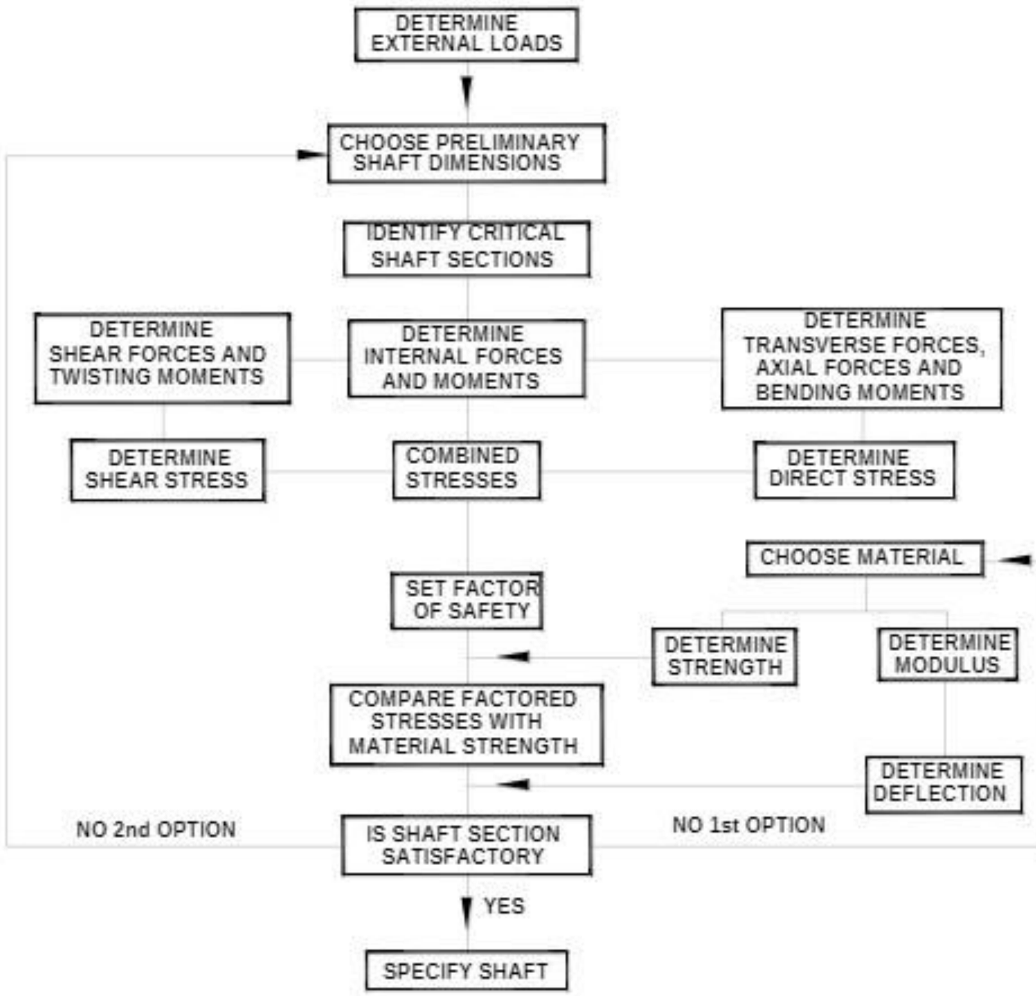
3.1. Considerations in shaft design:

- o Size and spacing of components o Material selection, material treatments o Deflection and rigidity o Stress and strength o Frequency response
- o Assembly, manufacturing & servicing constraints

4. LOADS ON SHAFT:



5. FLOW CHART FOR SHAFT DESIGN:



6. COMMON FORMULA TO DESIGN SHAFT IN COMBINED LOADING:

$$T_e = \sqrt{\left[K_m \times M + \frac{\alpha F\, d_o\, (1+k^2)}{8}\right]^2 + (K_t \times T)^2}$$

$$= \frac{\pi}{16} \times \tau\, (d_o)^3\, (1-k^4)$$

$$M_e = \frac{1}{2}\left[K_m \times M + \frac{\alpha F\, d_o\, (1+k^2)}{8} + \sqrt{\left\{K_m \times M + \frac{\alpha F\, d_o\, (1+k^2)}{8}\right\}^2 + (K_t \times T)^2}\right]$$

$$= \frac{\pi}{32} \times \sigma_b\, (d_o)^3\, (1-k^4)$$

$$\alpha = \frac{1}{1 - 0.0044\,(L/K)}$$ L/k > 115

$$\alpha = \frac{\sigma_y\,(L/K)^2}{C\pi^2 E}$$ L/k < 115

Where alpha is column factor

$$k = \frac{d_i}{d_o}$$

K_m = Combined shock and fatigue factor for bending, and
K_t = Combined shock and fatigue factor for torsion.

Value of Km and Kt by using following table:

Nature of load	K_m	K_t
1. Stationary shafts		
(*a*) Gradually applied load	1.0	1.0
(*b*) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2. Rotating shafts		
(*a*) Gradually applied or steady load	1.5	1.0
(*b*) Suddenly applied load with minor shocks only	1.5 to 2.0	1.5 to 2.0
(*c*) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

7. VALUE BENDING STRESS AND SHEAR STRESS IS OBTAINED FROM DESIGN DATA BOOK FOR THE SELECTED MATERIAL:

Prob.
A hollow shaft of 0.5 m outside diameter and 0.3 m inside diameter is used to drive a propeller of a marine vessel. The shaft is mounted on bearings 6 metre apart and it transmits 5600 kW at 150 r.p.m. The maximum axial propeller thrust is 500 kN and the shaft weighs 70 kN.
Determine :
1. The maximum shear stress developed in the shaft, and
2. The angular twist between the bearings.

Experiment 2: Design of a rigid coupling

1. AIM OF THE EXPERIMENT: Find the value of dimensions of rigid coupling.

2. THEORY: Shafts are usually available up to 7 metres length due to inconvenience in transport. In order to have a greater length, it becomes necessary to join two or more pieces of the shaft by means of a coupling.

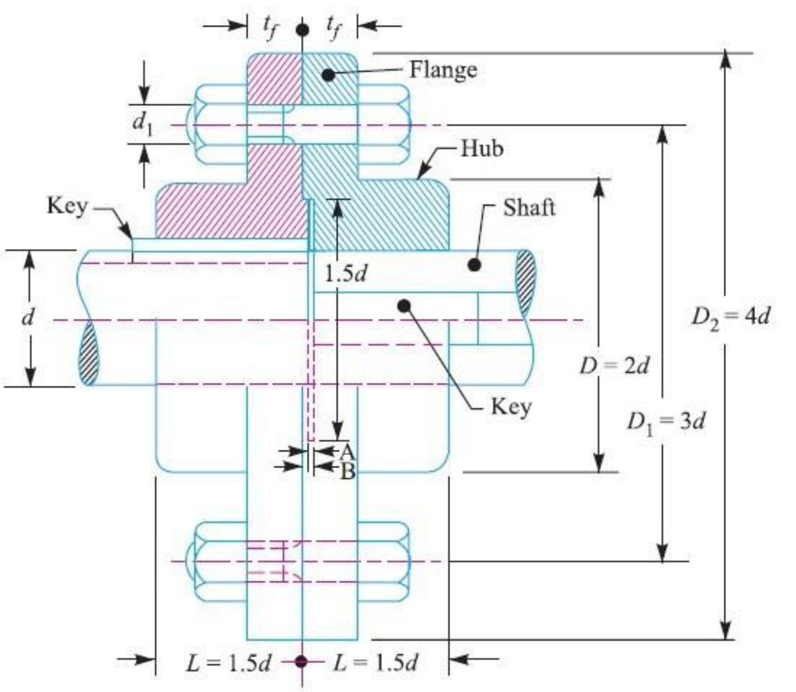
Purposes of couplings:

I. To provide for the connection of shafts of units that are manufactured separately such as a motor and generator and to provide for disconnection for repairs or alternations.

II. To provide for misalignment of the shafts or to introduce mechanical flexibility.

III. To reduce the transmission of shock loads from one shaft to another.

IV. To introduce protection against overloads. V. It should have no projecting parts.



The usual proportions for an unprotected type cast iron flange coupling, as shown in figure above, are as follows :
If d is the diameter of the shaft or inner diameter of the hub, then
Outside diameter of hub,

D = 2 (d)
Length of hub, L = 1.5 (d)

Pitch circle diameter of bolts,

$D_1 = 3(d)$

Outside diameter of flange,

$D_2 = D_1 + (D_1 - D) = 2 D_1 - D = 4 d$

Thickness of flange, $t_f = 0.5$ d

Number of bolts = 3, for d up to 40 mm

= 4, for d up to 100 mm =

6, for d up to 180 mm

3. DESIGN FOR HUB

The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft.

$$T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right)$$

The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore, from the above relation, the induced shearing stress in the hub may be checked.

The length of hub (L) is taken as 1.5 d.

The material of key is usually the same as that of shaft. The length of key is taken equal to the length of hub.

4. DESIGN FOR FLANGE

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the torque transmitted,

T = Circumference of hub × Thickness of flange × Shear stress of flange × Radius of hub

$$= \pi D \times t_f \times \tau_c \times \frac{D}{2} = \frac{\pi D^2}{2} \times \tau_c \times t_f$$

5. DESIGN FOR BOLTS

The bolts are subjected to shear stress due to the torque transmitted. The number of bolts (n) depends upon the diameter of shaft and the pitch circle diameter of bolts (D1) is taken as 3 d. We know that

$$\text{Load on each bolt} = \frac{\pi}{4}(d_1)^2\ \tau_b$$

$\therefore$ Total load on all the bolts

$$= \frac{\pi}{4}(d_1)^2\ \tau_b \times n$$

and torque transmitted,
$$T = \frac{\pi}{4}(d_1)^2\ \tau_b \times n \times \frac{D_1}{2}$$

$$= n \times d_1 \times t_f$$

$$= (n \times d_1 \times t_f)\ \sigma_{cb}$$

$$T = (n \times d_1 \times t_f \times \sigma_{cb})\ \frac{D_1}{2}$$

Prob:

Design a cast iron protective type flange coupling to transmit 15 kW at 900 r.p.m. from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used:

Shear stress for shaft, bolt and key material = 40 MPa

Crushing stress for bolt and key = 80 MPa

Shear stress for cast iron = 8 MPa

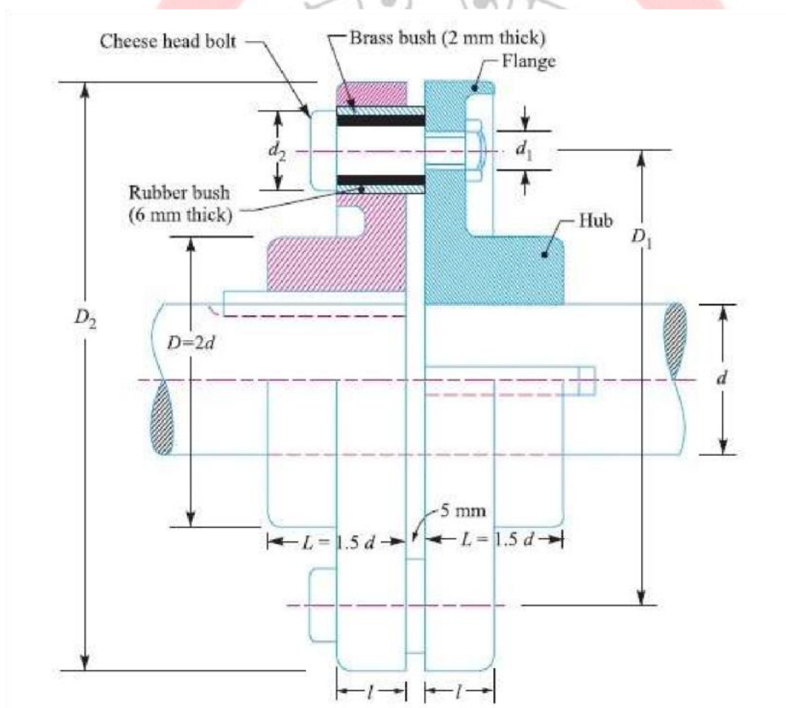
Draw a neat sketch of the coupling.

Experiment 3: Design of Flexible coupling

1. AIM OF THE EXPERIMENT: Find the value of dimensions of rigid coupling.

2. THEORY: Flexible coupling is used to join the abutting ends of shafts when they are not in exact alignment. In a direct electric drive from an electric motor to a machine tool, a flexiblecoupling is used so as to permit an axial misalignemnt of the shaft without undue absorption of the power which the shaft are transmitting. Following are the different types of flexible couplings :

1. Bushed pin flexible coupling, 2. Oldham's coupling, and 3. Universal coupling.



Bushed pin flexible coupling

In designing the bushed-pin flexible coupling, the proportions of the rigid type flange coupling are modified. The main modification is to reduce the bearing pressure on the rubber or leather bushes and it should not exceed 0.5 N/mm^2. In order to keep the low bearing pressure, the pitch circle diameter and the pin size is increased.

Let l = Length of bush in the flange,

d_2 = Diameter of bush,

p_b = Bearing pressure on the bush or pin,

n = Number of pins, and

D_1 = Diameter of pitch circle of the pins.

We know that bearing load acting on each pin,

$$W = p_b \times d_2 \times l$$

$\therefore$ Total bearing load on the bush or pins

$$= W \times n = p_b \times d_2 \times l \times n$$

and the torque transmitted by the coupling,

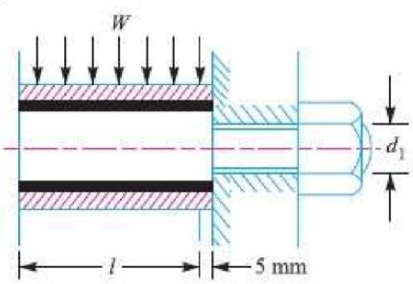
$$T = W \times n \left(\frac{D_1}{2}\right) = p_b \times d_2 \times l \times n \left(\frac{D_1}{2}\right)$$

The threaded portion of the pin in the right hand flange should be a tapping fit in the coupling hole to avoid bending stresses.

The threaded length of the pin should be as small as possible so that the direct shear stress can be taken by the unthreaded neck.

Direct shear stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2}$$



Since the pin and the rubber or leather bush is not rigidly held in the left hand flange, therefore the tangential load (W) at the enlarged portion will exert a bending action on the pin as shown in Fig. 13.16. The bush portion of the pin acts as a cantilever beam of length l. Assuming a uniform distribution of the load W along the bush, the maximum bending moment on the pin,

$$M = W\left(\frac{l}{2} + 5 \text{ mm}\right)$$

We know that bending stress,

$$\sigma = \frac{M}{Z} = \frac{W\left(\frac{l}{2} + 5 \text{ mm}\right)}{\frac{\pi}{32}(d_1)^3}$$

Since the pin is subjected to bending and shear stresses, therefore the design must be checked either for the maximum principal stress or maximum shear stress by the following relations, Maximum principal stress

$$= \frac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right]$$

and the maximum shear stress on the pin

$$= \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2}$$

The value of maximum principal stress varies from 28 to 42 MPa.

Prob.

Design a bushed-pin type of flexible coupling to connect a pump shaft to a motor shaft transmitting 32 kW at 960 r.p.m. The overall torque is 20 percent more than mean torque.
The material properties are as follows :

(a) The allowable shear and crushing stress for shaft and key material is 40 MPa and 80 MPa respectively.

(b) The allowable shear stress for cast iron is 15 MPa.

(c) The allowable bearing pressure for rubber bush is 0.8 N/mm2.

(d) The material of the pin is same as that of shaft and key. Draw neat sketch of the coupling.

Experiment 4: Design of Helical Spring

1. AIM OF THE EXPERIMENT: Design a helical spring based on given maximum force and deflection to consider. Analyze the steps involved in designing the helical spring.

2. OBJECTIVE FOR THE DESIGN:
    - The spring should possess sufficient strength to withstand the external load.
    - It should have different load-deflection characteristic.
    - It should not buckle under the external load.

3. TYPES OF SPRINGS:

Helical spring

Conical or volute spring

Torsional

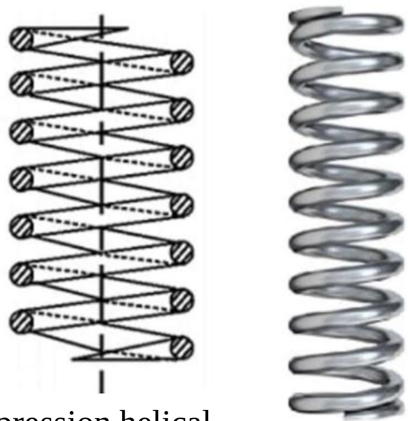
Disc spring

Laminated or leaf spring

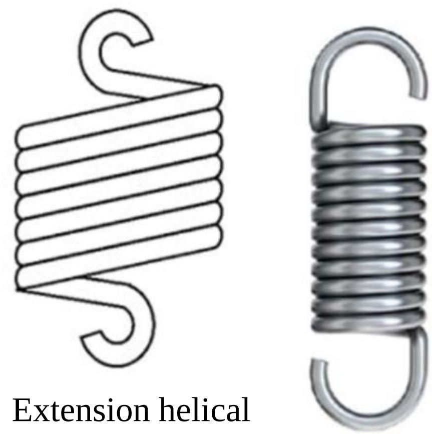
3.1. Two basic types of helical spring:

(i) Compression spring: shorten under the action of external load

(ii) Extension spring: elongate under the action of external load



Compression helical



Extension helical

4. HELICAL COMPRESSION SPRING DESIGN:

The main dimensions of a helical spring subjected to compressive force are shown in the figure. They are:

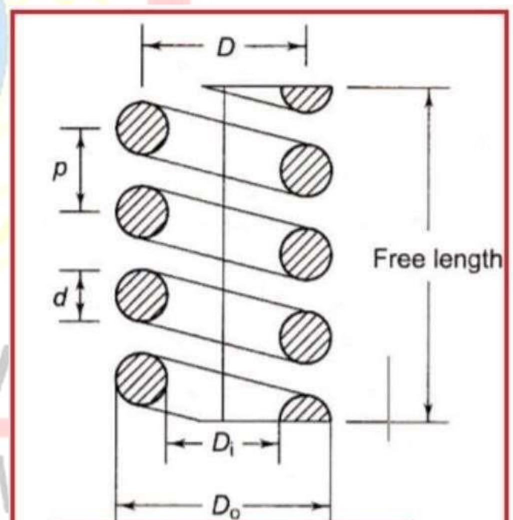
- d = wire diameter of the spring (mm)
- D_i = inner diameter of the spring (mm)
- D_o = inner diameter of the spring (mm)

- D = mean diameter of the spring (mm)

$$D = (D_i + D_o) / 2$$

- p = pitch of the spring (mm)

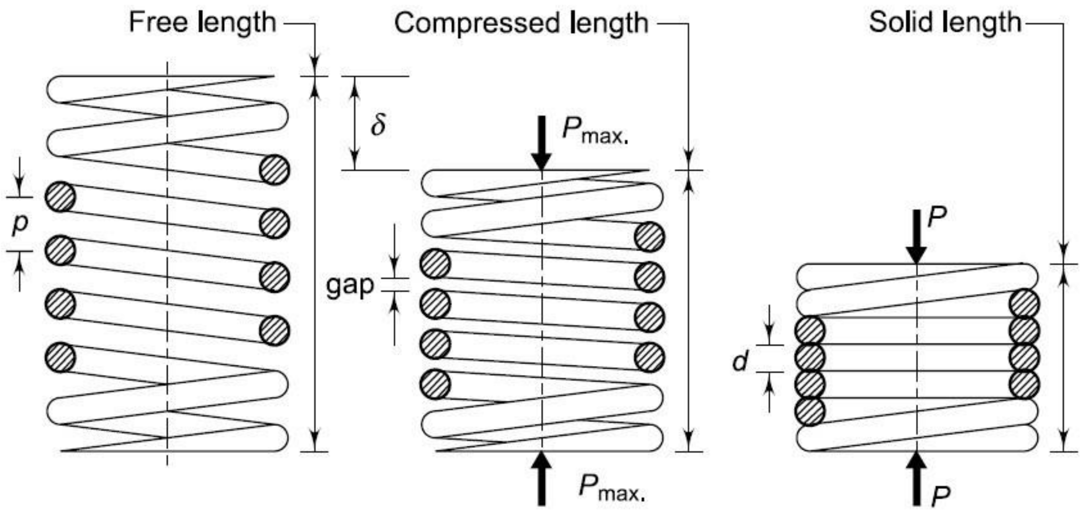
- C = Spring index = D / d



Terminology of Helical Springs

- For lower C values (C < 3): The actual stresses in the wire are excessive due to curvature effect.
- For higher C values (C > 15): Spring is prone to buckling and tangle easily during handling.
- A spring index from 4 to 12 is considered best from manufacturing conditions.

5. TERMINOLOGIES USED IN HELICAL SPRINGS:



- Solid Length: It is defined as the axial length of the spring, which when compressed, the adjacent coils touch each other. No compression is possible beyond this point. Solid length = $N_t d$, N_t = Total no. of coils.
- Compressed Length: It is defined as the axial length of the spring, which is subjected to maximum compressive force. In this case, the spring is subjected to a maximum deflection of δ. Some gap is provided between the adjacent coils to prevent their clashing.

Clashing allowance = 15% of maximum deflection. Also, 1 - 2 mm gap between the adjacent coils is provided as a thumb rule.

Total gap = (N_t - 1)xGap between adjacent coils.

- Free Length: It is defined as the axial length of the unloaded spring. In this case, no external force acts on the spring.
  Free length = compressed length + δ
  = solid length + total axial gap + δ

- Pitch: It is defined as the axial distance between the adjacent coils in the uncompressed state.
  p = free length/(N_t - 1) Stiffness: k = P/ δ P = Axial

Spring Force δ = Axial deflection of spring
corresponding to force P.

6. PROCEDURES FOR DESIGNING A HELICAL SPRING:

The basic procedure for the design of helical spring consists of following steps :

(i) For the given application, maximum spring force (P) and corresponding deflection (δ) are to be estimated.

In some cases, maximum spring force (P) and the stiffness (k) of the spring is specified.

(ii) A suitable spring material is to be selected and its ultimate tensile strength has to be ascertained.

Thepermissible shear stress for the spring wire is estimated by :

$$\tau_P = (0.3 - 0.5)\,\sigma_{UT}$$

(iii) Depending on the application, appropriate spring index (C) has to be choosen. In general, a 'C' value of 8 is considered as a good value. However, 'C' value should not be less than 3.

(iv) The Wahl's factor is to be calculated using the following relationship :

$$K_W = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

(v) The wire diameter(d) is evaluated using the following eqn,

$$\tau = K_W \left(\frac{8PD}{\pi d^3}\right) = K_W \left(\frac{8PC}{\pi d^2}\right)$$

(vi) The mean diameter(D)of the coil is obatined using,

$$D = Cd$$

(vii) The number of active coils is determined using the following eqn,

$$\delta = \frac{8PD^3N}{Gd^4}$$

(viii) Depending upon the type of the ends for a spring, the no. of inactive coils are determined. Total no. of coils (N_t) can be found out by adding no. of active and inactive coils.

(ix) The solid length of the spring = $N_t d$

$$8PD^3N$$

(x) Actual deflection $\delta = \frac{\quad}{Gd^4}$

(xi) Total axial gap between the coils = $(N_t - 1)$ x gap between adjacent coils. In some cases, total gap is taken as 15% of the maximum displacement.

(xii) Free length of the spring is to be determined using,

Free length = Solid length + total gap + δ

(xiii) Pitch(p)of the spring is found out as,

$$p = \frac{\text{Free length}}{(N_t - 1)}$$

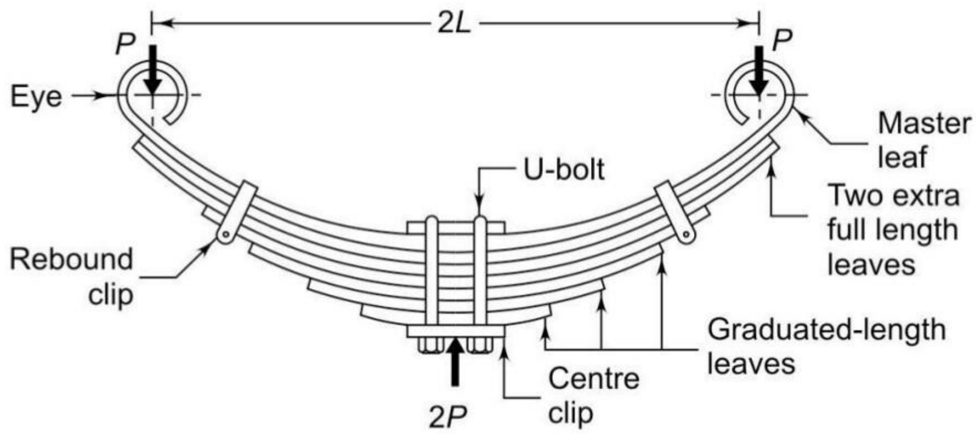
(xiv) The stiffness (k) is calculated using,

$$= \frac{Gd^4 k}{8D^3 N}$$

Experiment 5: Design of Leaf Spring

1. AIM OF THE EXPERIMENT: Find dimension of leave spring and assure that the value of stresses are within the limit.
2. THEORY:

Leaf springs (also known as flat springs) are made out of flat plates. The advantage of leaf spring over helical spring is that the ends of the spring may be guided along a definite path as it deflects to act as a structural member in addition to energy absorbing device. Thus the leaf springs may carry lateral loads, brake torque, driving torque etc., in addition to shocks.



L = length of the cantilever or half the length of semi-elliptic spring (mm)

P = force applied at the end of the spring (N)

P_f = portion of P taken by the extra full-length leaves (N)

P_g = portion of P taken by the graduated-length leaves (N)

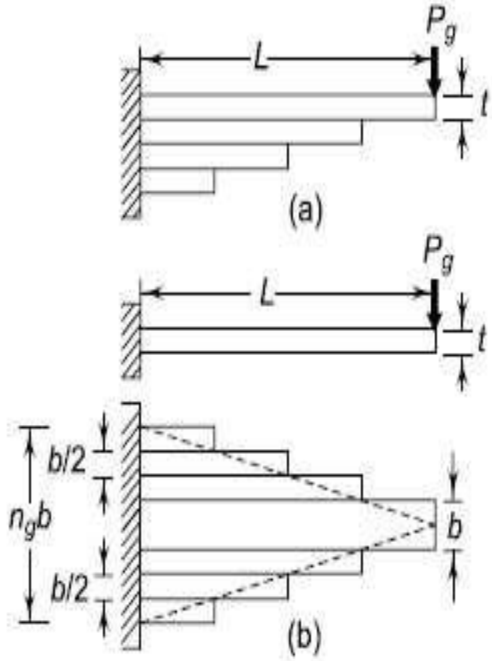
The group of graduated-length leaves along with the master leaf can be treated as a triangular plate, as shown in Fig. give in next slide. In this case, it is assumed

$$\delta_g = \frac{P_g L^3}{2EI_{\text{max.}}} = \frac{P_g L^3}{2E\left[\frac{1}{12}(n_g b)(t^3)\right]}$$

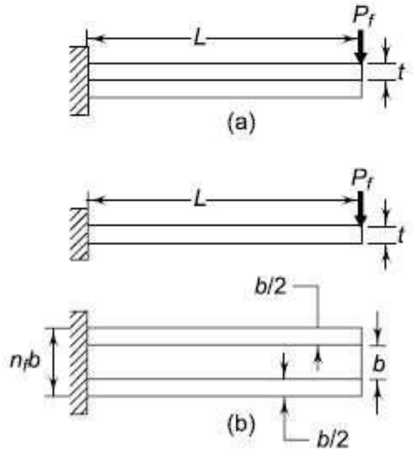
$$\delta_g = \frac{6P_g L^3}{E n_g b t^3}$$

$$(\sigma_b)_g = \frac{M_b y}{I} = \frac{(P_g L)(t/2)}{\left[\frac{1}{12}(n_g b)(t^3)\right]}$$

$$(\sigma_b)_g = \frac{6P_g L}{n_g b t^2} \qquad \text{(a)}$$



$$(\sigma_b)_f = \frac{6P_f L}{n_f b t^2} \qquad \text{(c)}$$



Extra full-length leaves as rectangular plate

□ The deflection at the load point is given by,

$$\frac{P_f L^3}{3EI} = \frac{P_f L^3}{3E\left[\frac{1}{12}(n_f b)(t^3)\right]}$$

$$\delta_f = \frac{4P_f L^3}{E n_f b t^3}$$

Since the deflection of full-length leaves is equal to the deflection of graduated-length leaves,

$$\delta_g = \delta_f$$

$$\frac{6P_g L^3}{E n_g b t^3} = \frac{4P_f L^3}{E n_f b t^3}$$

$$\frac{P_g}{P_f} = \frac{2n_g}{3n_f}$$

$$P_g + P_f = P$$

$$P_f = \frac{3n_f P}{(3n_f + 2n_g)}$$

$$P_g = \frac{2n_g P}{(3\,n_f + 2n_g)}$$

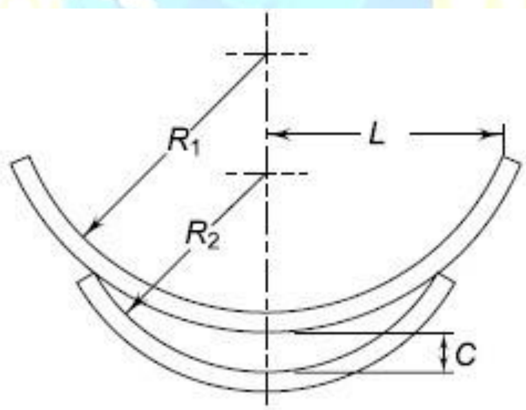
$$(\sigma_b)_g = \frac{12PL}{(3n_f + 2n_g)\,bt^2}$$

$$(\sigma_b)_f = \frac{18PL}{(3n_f + 2n_g)\,bt^2}$$

Bending stresses in full-length leaves are 50% more than those in graduated-length leaves.

$$\delta = \frac{12PL^3}{Ebt^3(3n_f + 2n_g)}$$

3. NIPPING OF LEAF SPRINGS



One of the methods of equalising the stresses in different leaves is to pre-stress the spring. The pre-stressing is achieved by bending the leaves to different radii of curvature, before they are assembled with the centre clip. The initial gap C between the extra full-length leaf and the graduated-length leaf before the assembly, is called a ‘nip’.

The deflection at the end of the spring is determined as given below:

$$(\sigma_b)_f = \frac{6L(P_f - 0.5P_i)}{n_f bt^2}$$

$$(\sigma_b)_f = \frac{6PL}{nbt^2}$$

$$\sigma_b = \frac{6PL}{nbt^2}$$

The deflection of the multi-leaf spring due to the external force P is the same

Prob:

A semi-elliptic leaf spring used for automobile suspension consists of three extra full-length leaves and 15 graduated-length leaves, including the master leaf. The centre-to-centre distance between two eyes of the spring is 1 m. The maximum force that can act on the spring is 75 kN. For each leaf, the ratio of width to thickness is 9:1. The modulus of elasticity of the leaf material is 207 000 N/mm2. The leaves are pre-stressed in such a way that when the force is maximum, the stresses induced in all leaves are same and equal to 450
N/mm2. Determine

(i) the width and thickness of the leaves;

(ii) the initial nip; and

(iii) the initial pre-load required to close the gap C between extra full-length leaves and graduatedlength leaves.

Experiment 6: Design of Brake

1. AIM OF THE EXPERIMENT: To design a block brake with short shoe.
2. THEORY: A brake is defined as a mechanical device that is used to absorb the energy possessed by a moving system or mechanism by means of friction. The primary purpose of brake is to slowdown or completely stops the motion of a moving system, such as rotating drum, machine or vehicle. It is also used to hold the parts of the system in position at rest.
3. TYPES OF BRAKES:

   I. Mechanical Brakes: These are operated by mechanical means such as levers, spring &pedals. Type of block brake is Block Brake, Internal Brake or External Shoe Brake, disc brake, band brake.

   II. Hydraulic & pneumatic brakes: These are operated by fluid pressure such as oil or air pressure.

   III. Electrical brakes:
   These are operated by magnetic forces.
4. ENERGY EQUATIONS:

The braking torque depend upon the amount of energy absorb by the brake. For a translating body, the kinetic energy (K.E) absorbed by brake during

- Braking Period: K.E. = $1/2\ m(v_1^2 - v_2^2)$
- For a rotating body: K.E=$1/2\ I(w_1^2 - w_2^2)$
- In hoist application

  The potential energy(P.E) stored by the brake during braking period = mgh Where h=distance by which mass m falls during braking period.

$$E = M_T \Theta$$

Where E= total energy absorbed by the brake.

M_T= braking torque

Θ= angle through which brake drum rotate during the braking period.

5. DESIGN OF A BLOCK BRAKE WITH SHORT SHOE:

A block brake consists of a simple block, which is pressed against the rotating drum by mean of lever. The friction between the block & brake drum causes the retardation of drum.

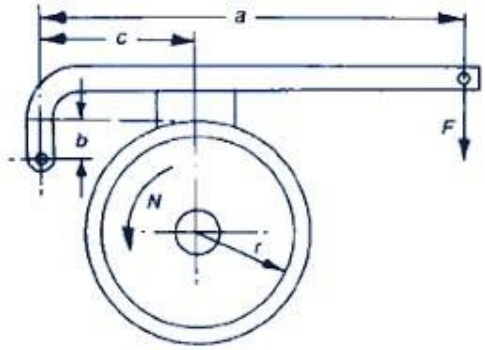


Fig. 1.1 Single Block Brake

The analysis is based on following assumption:

1. The block is rigidly attached to the lever.
2. The angle of contact between the block and brake drum is small resulting in a uniform pressure distribution. Considering the forces acting on the brake drum, $M_T = \mu NR$, where R=radius of brake drum.

The dimensions of block are determined by the following expression: N=plω Where p = permissible pressure between block & brake drum. l &ω= length and width of the block respectively.

 Generally, drum dia./4< ω <drum dia./2

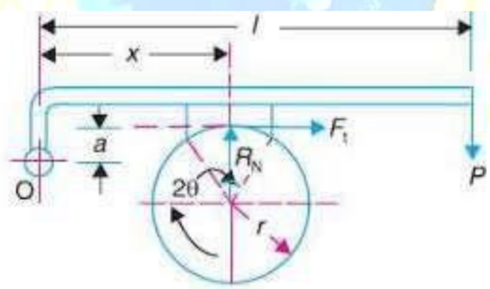


Fig. 1.2. Free Body Diagram (Clockwise Rotation)

- Considering the equilibrium of forces in vertical and horizontal direction: .
- Taking moment of forces acting on the lever about hinge point 0.

  Using Equations:

  P×b-N×a+μN C=0

  P= a C ×-μ   __N

   b

Case 1: a>μC: partially self energising brake.

Case 2: a=µC: self locking brake

Case 3: a<µC: uncontrolled braking and grabbing condition.

6. TO DESIGN A DIFFERENTIAL BAND BRAKE:

The band brake in which one of the band passes through the fulcrum is called simple band brake, while the band brakes in which neither of the band end passes through the fulcrum is called differential band brake.

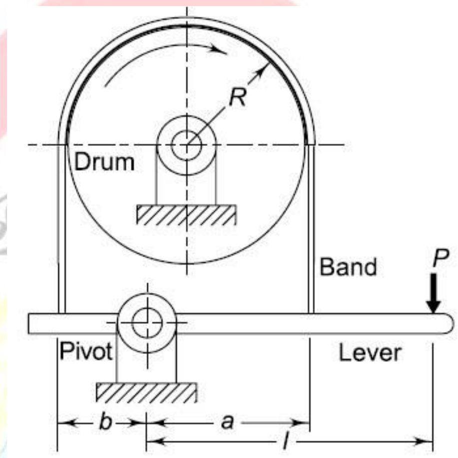


Fig. 1.3. Differential Band Brake

The design steps are as follows:

- Step 1: Tension in Bands: $\frac{P_1}{P_2} = e^{\mu\theta}$

- Step 2: Actuating Force: $P = \frac{P_2\left(a - be^{\mu\theta}\right)}{l}$

- Step 4: Torque Capacity of brake: $M_t = (P_1 - P_2)R$

Step 5: For self-locking condition: $P \le 0$ or negative or $a \le b e^{\mu\theta}$

Hence, the condition for self-locking is given by $\frac{a}{b} \le e^{\mu\theta}$

Experiment 7: Design of clutch

1. AIM OF THE EXPERIMENT: To understand the functioning of a friction clutch and obtain the dimensions of a friction clutch for a given application.

2. THEORY:

- Clutch: Clutch is a mechanical device used to engage and disengage power transmission from a driving shaft to a driven shaft. Different types of clutches:

Clutch Terminology

1. Crank Shaft
2. Flywheel
3. Friction Disk or clutch disk
4. Pressure Plate
5. Friction Lining
6. Diaphragm spring, kept between pressure plate and outer cover
7. Splines
8. Cover or cover of pressure plate
9. Friction Material: Asbestos fibre

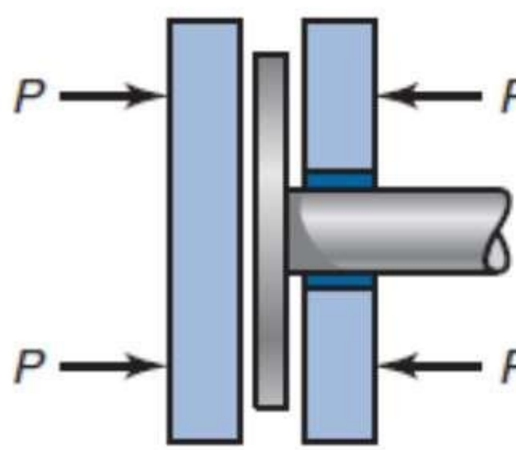


Figure 1: Clutches force balance

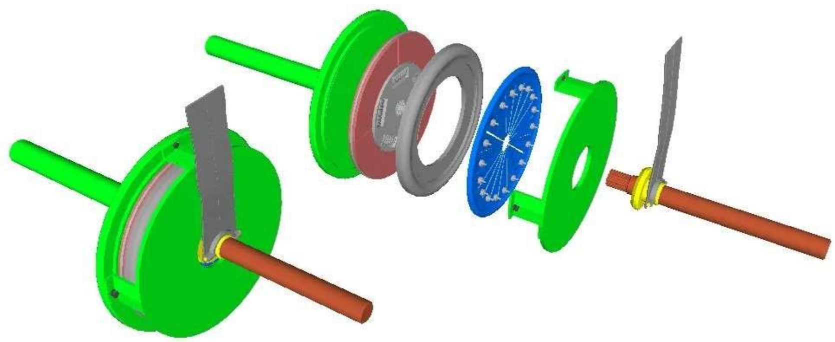


Figure 2: Different parts of a friction clutch

Figure 3: Actual pictures of a clutches

- Positive clutches: Positive clutches have no slip. Simultaneously, they can not be engaged and disengaged in rotating conditions cone

Cones: green and blue

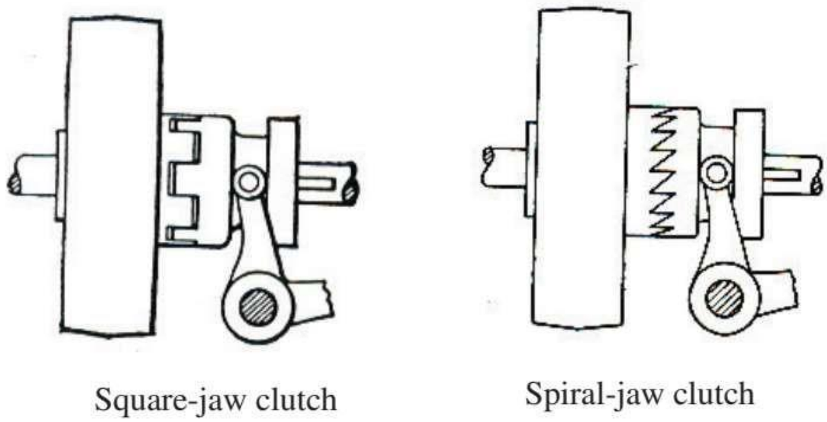
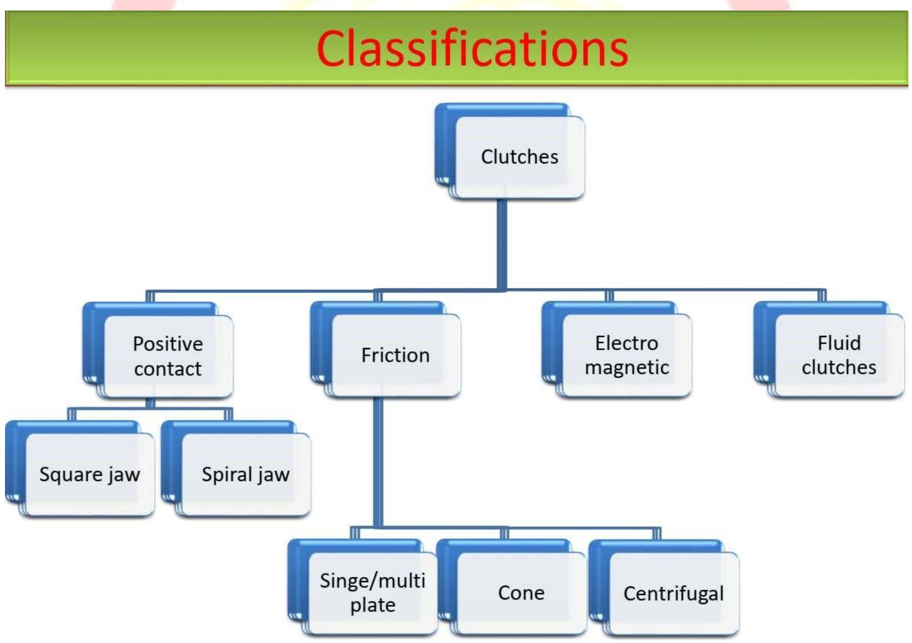


Figure 4: Positive clutches



cone 2. Shaft: the blue is sliding on splines 3. Friction material: usually on the green cone 4. Spring: brings the blue cone back after using clutch control 5. Clutch control: separating both cones by pressing 6. Rotating direction: both directions of the axis are possible

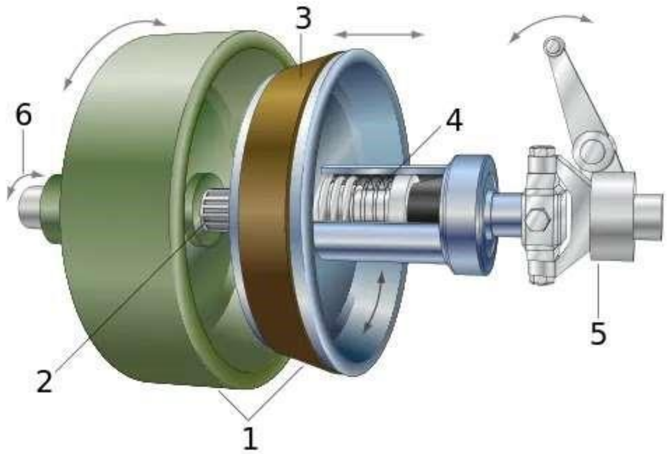


Figure 5: Cone clutch

- Derivations for frictional torque:

The pressure between the contacting surfaces should have a uniform distribution over the surface. In such cases, wear will be greater at a larger diameter. The loss of material due to higher wear results in a change in pressure profile. The disk will now follow the uniform wear condition. A new clutch is close to uniform pressure condition and a worn-out clutch follows a uniform wear condition.

1. Uniform pressure condition: We need an expression for torque in axial force, friction coefficient, and disc size.

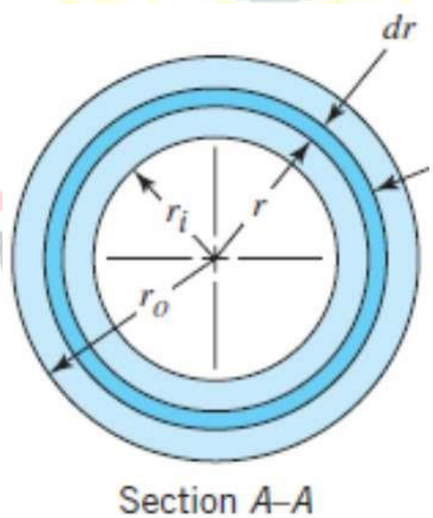


Figure 6: Differential ring of thickness dr.

The normal force acting on a differential ring element is given by

$$dF = (2\pi r dr)p$$

Here, p is the (uniform) pressure. The total normal force acting on the plate is

$$F = \int_{r_i}^{r_o} 2\pi p r dr = \pi p({r_o}^2 - {r_i}^2)$$

The torque that is produced due to frictional forces is given by,

$$dT = (2\pi r dr)pfr$$

Here the coefficient of friction is f.
The total torque that can be developed is given by,

$$T = \int_{r_i}^{r_o} 2\pi p f r^2 dr = \frac{2}{3}\pi p f({r_o}^3 - {r_i}^3)$$

Replacing the pressure p in terms of axial force P, and for N number of plates,

$$T = \frac{2Ff({r_o}^3 - {r_i}^3)}{3({r_o}^2 - {r_i}^2)}N$$

2. Uniform wear theory: Wear rate is proportional to friction work, i.e., friction force times the rubbing velocity.
   Hence, for uniform wear, we have

$$\mathrm{pr} = \mathrm{C}$$

Expression of torque is given by

$$T = Ff\left(\frac{r_o + r_i}{2}\right)N$$

Exercise: A plate clutch consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 120 and 250 mm respectively. The coefficient of friction is 0.12 and the permissible intensity of pressure is 2 N/mm 2. Assuming uniform wear theory, calculate the power transmitting capacity of the clutch at 750 rpm. Calculate the power transmitting capacity of the clutch using the uniform pressure theory.

Experiment 8: Design of Journal Bearing

1. AIM OF THE EXPERIMENT: Design of journal bearing

2. THEORY: Journal bearings are mechanical components used to support shafts of a machine. Journal bearings are therefore designed to carry radial loads. The load carrying capacity is developed due to the generation of pressure by the fluid film formed in the clearance space between the bearing and journal. The shape of the clearance space is shown in Figure 1. The expression for film thickness can be easily determined from the geometry of the film shape.

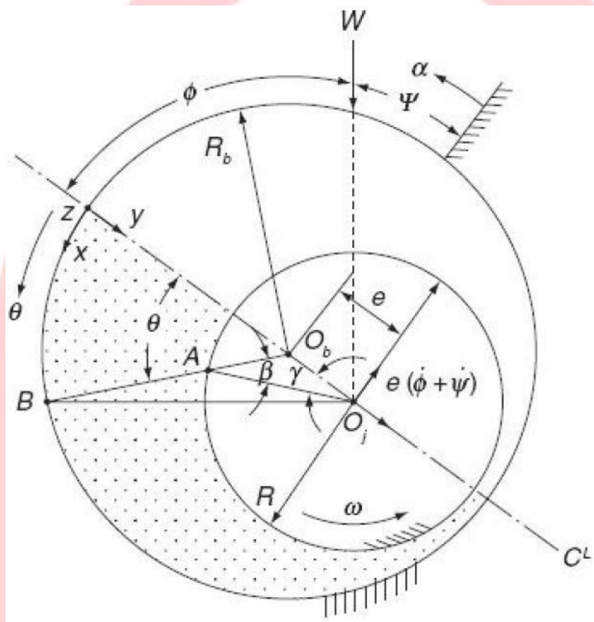


Figure 1. Journal bearing geometry

- Material: Cast iron, alloy steel, bronze, aluminium etc.
- Design equations:
  - o Eccentricity (e) = distance O_bO_j, where O_b= axis of bearing, O_j = axis of journal
  - o Radial clearance (c) = R_b - R, where R_b = radius of bearing, R= radius of journal o Eccentricity ratio (ε) =

-) — μ o

Somerfield number (S) = , where μ = viscosity of lubricant, □ = shaft speed

- Sliding velocity (U) = ωR , ω =2πN

- Film thickness (h) = AB which is the film thickness at B at angle θ from line of action

h = R_b - sin □ − □□□ □□□ □ −	For simplicity assumed << 1, so then h = c(1+ ε cos θ) L= length of bearing

- Infinitely long bearing approximation (L/2R) > 2,

$\left(c^2 \right) (2+\varepsilon^2)(1+\varepsilon\cos\theta)^2$ μ ε (ε) $W_r = 12\mu UL\left(\frac{R}{c}\right)^2 \frac{\varepsilon^2}{(2+\varepsilon^2)(1-\varepsilon^2)}$ $W_t = 6\mu UL\left(\frac{R}{c}\right)^2 \frac{\pi\varepsilon}{(2+\varepsilon^2)(1-\varepsilon^2)^{1/2}}$ p = $W= 6\mu UL\left(\frac{R}{c}\right)^2 \frac{\varepsilon[\pi^2-\varepsilon^2(\pi^2-4)]^{1/2}}{(2+\varepsilon^2)(1-\varepsilon^2)}$ $\Phi= tan^{-1}\left(\frac{W_t}{W_r}\right) = \left(\frac{\pi\sqrt{1-\varepsilon^2}}{2\varepsilon}\right)$ $S= \frac{(2+\varepsilon^2)(1-\varepsilon^2)}{6\pi\varepsilon}\left(\frac{1}{(\pi^2(1-\varepsilon^2))+4\varepsilon^2}\right)$ $f= \frac{c}{R}\left(\frac{\varepsilon \sin\Phi}{2} + \frac{2\pi^2 S}{(1-\varepsilon^2)^{1/2}}\right)$	where, P = Pressure W = Radial load W = Normal load W = Total load carrying capacity Φ = Attitude angle □ = Coefficient of friction

- Infinitely short bearing approximation (L/2 R) < 1

$\frac{3\ U}{Rc^2}\left(\frac{L^2}{4} - y^2\right)\frac{\varepsilon \sin\theta}{(1+\varepsilon\cos\ \theta)^2}$ μ $\frac{\mu UL^3}{c^2}\ \frac{\varepsilon^2}{1-\ 2\ 2}$ p = W = − (ε) W $= \frac{UL}{4c^2}\ \frac{\pi}{(1-\varepsilon^2)^{3/2}}$ μ ε $W= \frac{UL^3}{4c^2}\ \frac{}{(1-\varepsilon^2)^2}[(\pi^2(1-\varepsilon^2)+16\varepsilon^2]^{1/2}$ $\Phi= tan^{-1}\left(\frac{W_t}{W_r}\right) = \left(\frac{\pi\sqrt{1-\varepsilon^2}}{4\varepsilon}\right)$ $S= \frac{(1-\varepsilon^2)^2}{\pi\varepsilon[(\pi^2(1-\varepsilon^2)+1\ \ \varepsilon^2]^{1/2}}\left(\frac{2R}{L}\right)^2$ $f= \frac{c}{R}\left(\frac{2\pi^2 S}{(1-\varepsilon^2)^{1/2}}\right)$ Q_L = εULc μ ε μ F = ____ ________ (ε)	where, P = Pressure W = Radial load W = Normal load W = Total load carrying capacity Φ = Attitude angle □ = Coefficient of friction □ = Leakage flow rate F = Friction force

- Finite journal bearing design: For finite journal bearing, solution of Raimondi and Boyd method can be used. Dimensionless performance parameters are available in the form of Charts and tables. It can be used for solving problems.

Power loss (□) = $2\pi RNF$ Temperature rise (ΔT) = □ ___	where J = mechanical equivalent of heat □ = specific heat

Refer machine design data book for required data if needed.

<u>Problems:</u>

Problem 1: Write a computer program for design of journal bearing.

Problem 2: A full journal bearing of width 20 cm with a journal of diameter 10 cm has diametric clearance of 100 micro meters. The journal rotates at 1200 rpm. The absolute viscosity of lubricant at 20° C is 0.04 Pas. For an eccentricity ratio of 0.6, determine the minimum film thickness, load carrying capacity, attitude angle, Sommerfeld number, friction factor. Mass density, and specific heat of the oil at constant pressure may be taken as 900 kg/m3 and 2.0 J/g/0K, respectively.

Problem 3: A journal bearing is operating under following operating conditions: Journal diameter = 20 cm, bearing length = 10 cm and journal speed = 600 r.p.m. Clearance ratio may be chosen between 0.5, 1, 1.5, and 2.0 mm/m. Select a clearance ratio and determine load carrying capacity, oil flow rate, power loss, and temperature rise of lubricant while the viscosity of the oil at 38°C is 100cS and at 100°C is 12cS. Specific gravity of the oil is 0.9. The bearing is designed to run at an eccentricity ratio 0.6.

Experiment 9: Selection of Rolling Element Bearing

1. AIM OF THE EXPERIMENT: Selection of Rolling Element Bearing from

Manufacturer's Catalogue

2. THEORY: Bearing is mechanical device that permits relative motion between two parts, such as the shaft and the housing, with minimum friction. Bearings are classified in different ways the most important criterion to classify the bearings is the type of friction between the shaft and the bearing surface. Depending on type of friction, bearings are classified into two main group Sliding contact bearing and Rolling contact bearing.

Rolling contact bearing: Rolling contact bearing are also called antifriction bearings or simply ball bearing. Rolling element such as balls or rollers, are introduced between the surfaces that are in in relative motion. Figure 1 shows rolling contact bearing. Rolling contact bearing are used in following applications: Machine tool spindle, Automobile front and rear axle, Gear boxes, Small size electric motors and Rope etc.

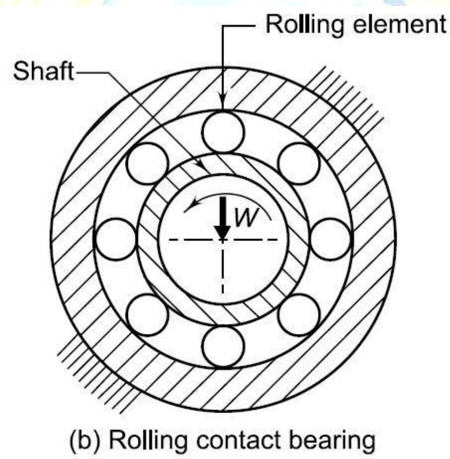


Figure 1. Rolling Contact Bearing

The types of rolling contact bearing, which are frequently used are:

(i) Deep Groove Ball bearing (ii) Cylindrical roller bearing (iii) Angular contact bearing (iv) Self-aligning bearing (v)Taper roller bearing (vi) Thrust ball bearing. Different types of rolling contact bearing are shown in Figure 2.

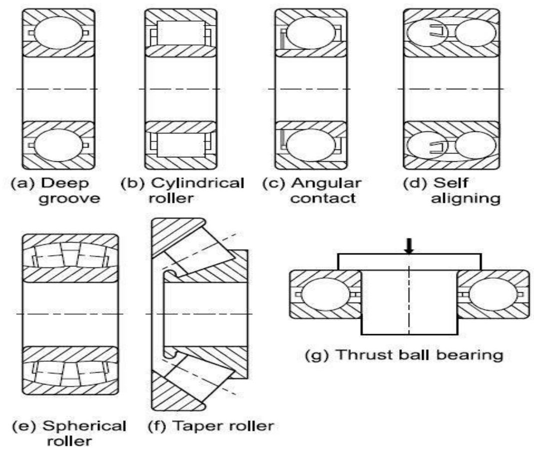


Figure 2. Types of Rolling Contact Bearing

Materials: Chrome Steel - SAE 52100, Stainless Steels, Stainless Steel Bearings– ACD34 /KS440 / X65Cr13 etc.

Selection of Rolling Element Bearing:

The information given here should serve to indicate which are the most important of the following points to be considered when selecting bearing type and thus facilitate an appropriate choice.

• Cylindrical & Needle roller – pure radial load.

• Thrust (cylindrical roller, ball), four point angular contact ball bearings – pure axial load.

• Taper roller, spherical roller, angular contact ball bearings – combined Load.

• Cylindrical roller, angular contact ball bearing– high speed.

• Deep groove, angular contact, and cylindrical roller bearing – high running accuracy.

Design equations/data:

- Equivalent dynamic load:

□ = □□□ + □□	□ = equivalent dynamic load (N) □ = radial load acting on bearing (N), □ = axial or thrust load acting on bearing □ = race- rotation factor □ = radial factor, □= thrust factor

- Load life equation:

□ □ = _ □	where, □ = rating bearing life (in million revolutions) C = dynamic load capacity (N) p = 3 (for ball bearings) p = 10/3 (for roller bearing)

- Life in hours:

$□ = \frac{60□□}{10}$	where, □ = rated bearing life (hours) □ = speed of rotation (rpm)

- Cyclic loads and speed:

$□ = \sqrt{[\frac{N_1P_1 + N_2P_2 + \cdots}{N_1 + N_2 + \cdots}]}$ $= \sqrt{[\frac{\sum □□}{\sum N}]}$ □	where N = □ + □ + ⋯ □ □ = equivalent dynamic load for complete work cycle (N) , □ … □ = dynamic load during first, second, ... nth element of work cycle □ , □ … □ = number of revolutions completed by first, second.... nth element of work cycle N = life of complete work cycle (rev)

- Cyclic loads and speeds (continuous variation of load):

□ = [_1 □ □□]⁄
□

- Bearing with probability of survival other than 90%:

□ = □ (⁄), where □ = reliability (in fraction), L = corresponding life (in million of revolution), □ and □ = constants (□ = 6.84 and □ = 1.17)

$\frac{L}{L_{10}}) = [\frac{\log_e(\frac{1}{R})}{\log_e(\frac{1}{\quad})}]^{⁄}$, where □ = life corresponding to a reliability of 90% or

□ , □ = 0.9

- System reliability: □ = (□) , where □ = number of bearings in the system (each having the same Reliability □), □ = reliability of the complete system Refer machine design data book for required data if needed.

Problems:

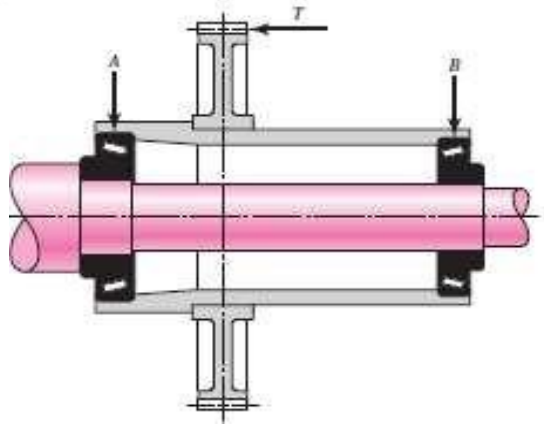
Problem 1: Write a computer program for selection of ball bearing from SKF/FAG manufacturer catalogue.

Problem 2: A ball bearing operates on the following work cycle:

Element No	Radial load (N)	Speed (rpm)	Element time (%)
1	3000	720	30
2	7000	1440	50
3	5000	900	20

The dynamic load capacity of the bearing is16.6 kN. Calculate (i) the average speed of rotation; (ii) the equivalent radial load; and (iii) the bearing life.

Problem 3: A ball bearing is subjected to a radial force of 2500 N and an axial force of 1000 N. The dynamic load carrying capacity of the bearing is 7350 N. The values of X and Y factors are 0.56 and 1.6 respectively. The shaft is rotating at 720 rpm. Calculate the life of the bearing. Problem 3: The gear-reduction unit shown has a gear that is press fit onto a cylindrical sleeve that rotates around a stationary shaft. The helical gear transmits an axial thrust load T of 1kN as shown in the figure. Tangential and radial loads (not shown) are also transmitted through the gear, introducing radial ground reaction forces at the bearings of 3.5kN for bearing A and 2.5 kN for bearing B. The desired life for each bearing is 90 kh at a speed of 150 rev/min with a 90 percent reliability. The first iteration of the shaft design indicates approximate diameters of 28mm at A and 25mm at B. Select suitable tapered roller bearings from TIMKEN catalogue.



Experiment 10: Design of Spur Gear

1. AIM OF THE EXPERIMENT: Design of spur gear

2. THEORY: In spur gears, the teeth are cut parallel to the axis of the shaft. As the teeth are parallel to the axis of the shaft, spur gears are used only when the shafts are parallel. The profile of the gear tooth is in the shape of an involute curve and it remains identical along the entire width of the gear wheel. Spur gears impose radial loads on the shafts. Spur gear nomenclature and a pair of spur gear is shown in Figure 1 and Figure 2 respectively.

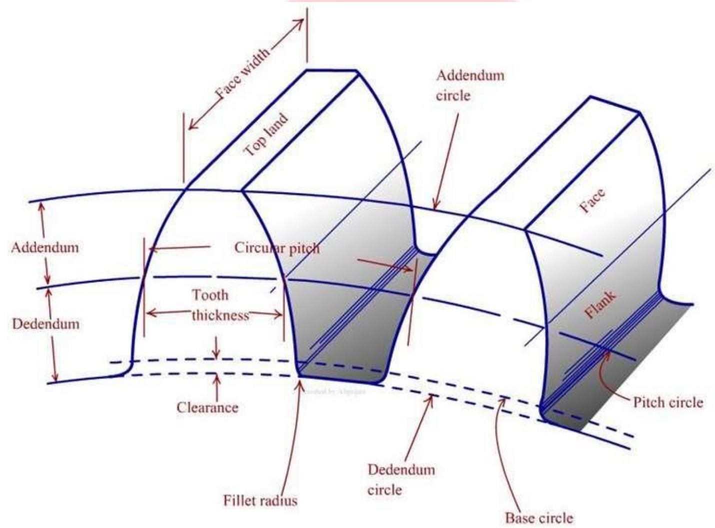


Figure 1. Gear nomenclature

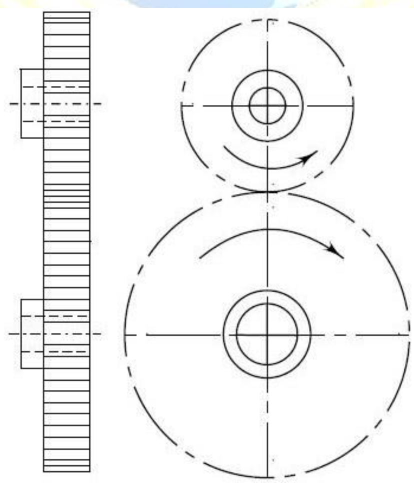


Figure 2. Spur gear

o Material: Steel, hardened teeth, cast iron, bronze, stainless steel, aluminium etc. o Design equations/ data:

- Module of gear

$P = \frac{z}{d}$ $m = \frac{d}{z}$ $d = mz$	where ,d = pitch circle diameter (mm), z = number of teeth on the gear P = Diametral Pitch m = Module (mm)

- **Recommended series of module (mm)**

Choice 1	1	1.25	1.5	2	2.5	3	4
	5	6	8	10	12	16	20
Choice 2	1.125	1.375	1.75	2.25	2.75	3.5	4.5
	`5.5	7	9	11	14	18	-
Choice 3	3.25	3.75	6.5	-	-	-	-

- Gear Ratio and transmission ratio

= i =	where, n_p = speed of pinion, n_g = speed of wheel, z_p = number of teeth on the pinion, z_g = number of teeth on the wheel

- Basic Relationship

P = p = P×p = π ( ) a =	where p = Circular Pitch (mm), d = pitch circle diameter (mm), z = number of teeth on the gear P = Diametral Pitch m = Module (mm) a = centre-to-centre distance (mm) z_p = number of teeth on the pinion, z_g = number of teeth on the wheel

- Standard proportions of gear tooth for 20° full depth involute system

Dimension	Notation	Proportion
Addendum	h_a	$h_a = m$
Dedendum	h_f	$h_f = 1.25m$
Clearance	c	$c = 0.25m$
Working depth	h_k	$h_k = 2$ m
Whole depth	h	H = 2.25 m
Tooth thickness	s	s = 1.5708m
Tooth space	….	1.5708m
Fillet radius	….	0.4m

- Spur gears - Component of tooth force

$M_t = \frac{\times \quad (\quad)}{}$ $P_t =$ $P_r = P_t \tan\alpha$ d_p $= mz_p$	where M_t = Torque transmitted by gear (N-mm) n_p = speed of pinion (rpm) P_t = Tangential component of resultant tooth force (N) P_r = Radial component of resultant tooth force (N) α = Pressure angle (°) d_p = pitch circle diameter of pinion (mm)

- Minimum number of teeth

Pressure angle (α)	14.5°	20°	25°
z_{min} (theoritical)	32	17	11
z_{min} (practical)	27	14	9

Note: The minimum number of teeth to avoid interference is given by, $z_{min} =$ ____

- Face width of tooth

Optimum range of face width : (8m) < b < (12m) or (b=10m)

- Beam Strength of gear tooth (Lewis' equation)

Beam strength (S_b) indicates the maximum value of tangential force that the tooth can transmit without failure:

□ = □ □□ □ □ = □ = $\frac{1}{3}$ □	S_b= beam strength of gear tooth (N) σ_b= permissible bending stress (MPa) Y= Lewis form factor based on virtual number of teeth S_e= endurance limit (MPa) S_{ut}=ultimate tensile strength (MPa)

- Wear strength of gear tooth (Buckingham's Equation)

$\frac{z_g+z_p}{2z_g}$ Q = (for external gear) Q = (for internal gear) S_w = b.Q.d_p.K K = $\frac{\text{□ □□□□□□□□□} \frac{1}{\text{□□}} + 1/\text{□□}}{1.4}$	Q = Ratio factor □ = number of teeth on pinion □ = number of teeth on wheel S_w = wear strength of the gear tooth (N) d_p = pitch circle diameter of pinion (mm) K = load-stress factor (MPa) σ_c = Surface endurance strength of the material (MPa) α = pressure angle E_p = modulus of elasticity of pinion materials (MPa) E_g = modulus of elasticity of wheel materials (MPa)

- For steel gears with 20° pressure angle

K = 0.16 ___	where BHN = Surface hardness of gears (Brinell hardness number)
According to G. Niemann: σ_c = 0.27(BHN) kgf/mm^2 = 0.27 (9.81) (BHN) N/mm^2	

- Values of Modulus of elasticity and poisson's ratio for gear materials

Material	Modulus of elasticity E (MPa)	Poisson's ratio
Steel	206000	0.3
Cast Steel	202000	0.3
Spheroidal cast iron	173000	0.3
Cast tin bronze	103000	0.3
Tin bronze	113000	0.3
Grey cast iron	118000	0.3

- Effective load on gear tooth

Tangential force due to rated torque or rated power (P_t)

× ( ) M_t = P_t =	P_t = Tangential force due to rated torque (N) M_t = rated torque (N-mm) Kw = power transmitted by gears (kW) n_p = speed of pinion (rpm) d_p = pitch circle diameter of pinion (mm)
Effective load on gear tooth (P_{eff}) – Preliminary gear design	
P_{eff} = ___	P_{eff} = effective load on gear tooth (N) C_S = Service factor C_v = Velocity factor
Effective load on gear tooth (P_{eff}) – Final gear design	
$P_{eff} = (C_S P_t + P_d)$	P_d = incremental dynamic load (N) (Buckingham's equation)

- Service factor for speed reduction gearboxes (C_s)

Working characteristic of driving machine	Working characteristic of driven machine		
	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.5	1.75	2.25
Note : For Electric motors, C_s =			

- Velocity factor (C_v)

For ordinary and commercially cut gears made with form cutters and with (v<10 m/s) C_v =
For accurately hobbed and generated gears with (v<20 m/s) C_v =
For precision gears with shaving, grinding, and lapping operations and with (v>20 m/s) . C_v = . √
□ = ______ where, □ = pitch line velocity (m/s) ×

$= \quad \frac{(\quad) P_d}{(\quad)}$ $\sqrt{\quad}$	P_d = dynamic load or incremental dynamic load (N) □ = pitch line velocity (m/s) C = deformation factor (MPa or N/mm^2) e = sum of errors between two meshing teeth (mm) b = face width of tooth (mm) P_t = tangential force due to rated torque (N)

- Deformation factor (C)

Deformation factor C depends upon moduli of elasticity of materials for pinion and gear and the form of tooth or pressure angle	
C = $\overline{[\frac{1}{E_p}+\frac{1}{E_g}]}$	k = constant depending upon the form of tooth E_p = Modulus of elasticity of pinion material (MPa or N/mm^2) E_g = Modulus of elasticity of wheel material (MPa or N/mm^2)
The values of k for various tooth forms are as follows: k = 0.107 (for 14.5° full depth teeth) k = 0.111 (for 20° full depth teeth) k = 0.107 (for 20° stub teeth)	

- Values of deformation factor C

Materials		14.5° full depth teeth	20° full depth teeth	20° stub teeth
Pinion Material	Gear Material			
Grey C.I.	Grey C.I.	5500	5700	5900
Steel	Grey C.I.	7600	7900	8100
Steel	Steel	11000	11400	11900

Refer machine design data book for required data if needed.

Problems:

Problem 1: Write a computer program for design of spur gear.

Problem 2: It is required to design a pair of spur gears with 20° full-depth involute teeth based on the Lewis equation. The velocity factor is to be used to account for dynamic load. The pinion shaft is connected to a 10kW, 1440 rpm motor. The starting torque of the motor is 150% of the rated torque. The speed reduction is 4:1. The pinion as well as the gear is made plain carbon steel 40C8 (S_{ut} = 600 N/mm^2). The factor of safety can be taken as 1.5. Design the gears, specify their dimensions and suggest suitable surface hardness for the gears.

Problem 3: A pair of spur gears with 20° full-depth involute teeth consists of a 20 teeth pinion meshing with a 41 teeth gear. The module is 3 mm while the face width is 40 mm. The material for pinion as well as gear is steel with an ultimate tensile strength of 600 N/mm^2. The gears are heat treated to a surface hardness of 400 BHN. The pinion rotates at 1450 rpm and the service factor for the application is 1.75. Assume that velocity factor accounts for the dynamic load and the factor of safety is 1.5. Determine the rated power that the gears can transit.

Problem 4: A pair of spur gears consists of a 24 teeth pinion, rotating at 1000 rpm and transmitting power to a 48 teeth gear. The module is 6 mm, while the face width is 60 mm. Both gears are made of steel with an ultimate tensile strength of 450 N/mm2. They are heat treated to a surface hardness of 250 BHN. Assume that velocity factor accounts for the dynamic load. Calculate (i) beam strength; (ii) wear strength; and (iii) the rated power that the gears can transmit, if service factor and the factor of safety are 1.5 and 2, respectively.

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