KARPAGAM ACADEMY OF HIGHER EDUCATION

(Deemed to be University Established Under Section 3 of UGC Act 1956)
Pollachi Main Road, Eachanari Post, Coimbatore – 641 021. INDIA

FACULTY OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

SYLLABUS

16BEME502 DESIGN OF MACHINE ELEMENTS 3 1 0 4 100

OBJECTIVES

1. To familiarize the various steps involved in the Design Process
2. To understand the principle involved in evaluating the shape and dimensions of a component to satisfy functional and strength requirements.
3. To learn to use standard practices and standard data
4. To learn to use catalogues and standard machine components

UNIT I STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS 12

Introduction to the design process – factors influencing machine design, selection of materials based on mechanical properties – Factor of safety. Direct, Bending and torsional stress equations – Impact and shock loading – calculation of principle stresses for various load combinations, eccentric loading – Design of curved beams – crane hook and 'C' frame – theories of failure – stress concentration – design for variable loading – Soderberg, Goodman and Gerber relations.

UNIT II DESIGN OF SHAFTS AND COUPLINGS 12

Design of solid and hollow shafts based on strength, rigidity and critical speed – Design of keys and key ways – Design of rigid and flexible couplings – Introduction to gear and shock absorbing couplings – design of knuckle joints.

UNIT III DESIGN OF FASTENERS AND WELDED JOINTS 12

Threaded fasteners – Design of bolted joints including eccentric loading – Design of welded joints for pressure vessels and structures – theory of bonded joints.

UNIT IV DESIGN OF SPRINGS AND FLYWHEEL 12

Design of helical, leaf, disc and torsional springs under constant loads and varying loads – Concentric torsion springs – Belleville springs – Design of flywheels involving stresses in rim and arm.

UNIT V DESIGN OF BEARINGS AND LEVERS 12

Selection of bearings – sliding contact and rolling contact types – Cubic mean load – Selection of journal bearings – McKees equation – Lubrication in journal bearings – calculation of bearing dimensions – Design of Levers.

TOTAL 60

(Permitted to use PSG design data book in the examination)

TEXT BOOKS

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
1	Juvinall R.C and Marshek K.M	Fundamentals of Machine Component Design, 5e	John Wiley and Sons, New Delhi	2015
2	Bhandari V. B	Design of Machine Elements, 4e	Tata McGraw–Hill Book Co, New Delhi	2016

REFERENCES

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
1	Norton R. L	Design of Machinery	Tata McGraw–Hill Book Co., New Delhi	2011
2	Orthwein W	Machine Component Design	Jaico Publishing Co., New Delhi	2013
3.	Bhandari V B	Introduction to Machine Design	McGraw–Hill Book Co., New York	2013
4	Spotts M.F, Shoup T. E	Design of Machine Elements	Pearson Education, New Delhi	2008

WEB REFERENCES:

1. www.roymech.co.uk

2. www.ncbi.nlm.nih.gov
3. www.engineersedge.com
4. www.bearings.machinedesign.com
5. www.efunda.com

KARPAGAM ACADEMY OF HIGHER EDUCATION

(Deemed to be University Established Under Section 3 of UGC Act 1956)
Pollachi Main Road, Eachanari Post, Coimbatore – 641 021. INDIA

FACULTY OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

LESSON PLAN

Subject Name : **Design of Machine Elements**

Subject Code : **16BEME502** (Credits - 4)

Name of the Faculty : **S. ARAVIND**

Designation : **Assistant Professor**

Year/Semester : **III / V**

Branch : **Mechanical Engineering**

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
UNIT – I: STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS			
1.	1	Introduction to the design process - factor influencing machine design, selection of materials based on mechanical properties - Factor of safety	T [1] 1-120, R [1] 1-20
2.	1	Direct, Bending and torsional stress equations – Impact and shock loading -eccentric loading, calculation of principle stresses for various load combinations	T [1] 131-150, T [2] 76-83
3.	1	Solving problems from principle stresses for various load combinations	T [2] 138-141, R [3] 177-181
4.	1	Design of curved beams – crane hook and 'C' frame	T [2] 130-134
5.	1	Solving problems from crane hook and 'C' frame	T [2] 138-141
6.	1	Tutorial – 1: Problems from principle stresses for various load combinations, crane hook and 'C' frame	T [2] 138-141
7.	1	theories of failure – stress concentration	T [1] 162-166, T [2] 106-116
8.	1	Solving problems from theories of failure, stress concentration	T [2] 138-141, 181-184
9.	1	design for variable loading – Soderberg, Goodman and Gerber relations	T [1] 312-348
10.	1	Solving problems from design for variable loading	T [2] 181-184
11.	1	Solving problems from design for variable loading	R [1] 402-414
12.	1	Tutorial – 2: Problems from theories of failure, stress concentration, design for variable loading	T [2] 138-141, 181-184
13.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - I			**13**

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
UNIT – II: DESIGN OF SHAFTS AND COUPLINGS			
14.	1	Design of solid and hollow shafts based on strength, rigidity and critical speed	T [1] 716-725, T [2] 330-346
15.	1	Solving problems from Design of solid and hollow shafts based on strength	T [2] 389-393, R [1] 569-576
16.	1	Solving problems from Design of solid and hollow shafts based on strength	T [2] 389-393, R [1] 569-576
17.	1	Solving problems from Design of solid and hollow shafts based on rigidity and critical speed	T [2] 389-393, R [1] 569-576
18.	1	Tutorial – 3: problems from Design of solid and hollow shafts based on strength, rigidity and critical speed	T [2] 389-393, R [1] 569-576
19.	1	Design of keys and key ways, Design of rigid and flexible couplings	T [2] 346-372, R [3] 214-236
20.	1	Solving problems from Design of keys and key ways	T [2] 389-393, R [3] 274-280
21.	1	Solving problems from Design of rigid couplings	T [2] 389-393, R [3] 274-280
22.	1	Solving problems from Design of flexible couplings	T [2] 389-393, R [3] 274-280
23.	1	Tutorial – 4: problems from Design of keys and key ways & Design of rigid couplings	T [2] 389-393, R [3] 274-280
24.	1	Introduction to gear and shock absorbing couplings - design of knuckle joints.	T [2] 94-103, R [1] 558-565
25.	1	Solving problems from design of knuckle joints	T [2] 94-103
26.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - II			**13**

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
UNIT – III: DESIGN OF FASTNERS AND WELDED JOINTS			
27.	1	Threaded fasteners - Design of bolted joints including eccentric loading	T [1] 411-470
28.	1	Solving problems from Design of bolted joints including eccentric loading	T [2] 269-272, R [1] 871-876
29.	1	Solving problems from Design of bolted joints including eccentric loading	T [2] 269-272, R [1] 871-876
30.	1	Solving problems from Design of bolted joints including eccentric loading	T [2] 269-272, R [1] 871-876

31.	1	Tutorial – 5: Problems from Design of bolted joints including eccentric loading	T [2] 269-272, R [3] 401-406
32.	1	Design of welded joints for pressure vessels and structures- Theory of bonded joints.	T [1] 474-486, T [2] 272-298
33.	1	Solving problems from Design of welded joints for pressure vessels and structures	T [2] 325-330, R [3] 494-502
34.	1	Solving problems from Design of welded joints for pressure vessels and structures	T [2] 325-330, R [4] 268-274
35.	1	Solving problems from Design of welded joints for pressure vessels and structures	T [2] 325-330, R [4] 268-274
36.	1	Tutorial – 6: Problems from Design of welded joints for pressure vessels and structures	T [2] 325-330, R [3] 494-502
37.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - III			**11**

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
UNIT – IV: DESIGN OF SPRINGS AND FLYWHEELS			
38.	1	Design of helical, torsional & concentric torsion springs under constant loads and varying loads	T [1] 497-528
39.	1	Solving problems from Design of helical springs under constant loads and varying loads	T [2] 443-447
40.	1	Solving problems from Design of torsional& concentric torsion springs under constant loads and varying loads	R [1] 807-810, R [4] 155-190
41.	1	Tutorial – 7: Problems from & Design of helical springs under constant loads and varying loads	T [2] 443-447
42.	1	Design of leaf springs under constant loads and varying loads	T [1] 522-527, T [2] 437-440
43.	1	Solving problems from Design of leaf springs under constant loads and varying loads	T [2] 443-447
44.	1	Tutorial – 8: Problems from Design of leaf springs under constant loads and varying loads	R [1] 807-810, R [4] 155-190
45.	1	Design of Belleville springs - Design of flywheels involving stresses in rim and arm.	R [1] 792-797, 539-546
46.	1	Solving problems from Design of Belleville springs & Design of flywheels involving stresses in rim and arm.	T [2] 749-767
47.	1	Tutorial – 9: Problems from Design of flywheels involving stresses in rim and arm.	T [2] 749-767
48.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - IV			**11**

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
UNIT – V: DESIGN OF BEARINGS AND LEVERS			
49.	1	Design of bearings – sliding contact and rolling contact types – Cubic mean load	T [1] 587-619, T [2] 564-597
50.	1	Solving problems from Design of rolling contact bearings	T [2] 597-600, R [1] 630-634
51.	1	Solving problems from Design of rolling contact bearings	T [2] 597-600, R [1] 630-634
52.	1	Tutorial – 10: Problems from Design of rolling contact bearings	T [2] 597-600, R [1] 630-634
53.	1	Design of journal bearings – McKee's equation – Lubrication in journal bearings – calculation of bearing dimensions	T [1] 546-586, R [1] 577-607
54.	1	Solving problems from Design of journal bearings	T [1] 546-586, T [2] 643-645
55.	1	Solving problems from Design of journal bearings	T [2] 643-645, R [4] 301-315
56.	1	Tutorial – 11: Problems from Design of journal bearings	T [2] 643-645, R [4] 301-315
57.	1	Design of Levers	T [2] 117-127, R [3] 132-156
58.	1	Solving problems from Design of Levers	T [2] 117-127, R [3] 132-156
59.	1	Tutorial – 12: Problems from Design of Levers.	T [2] 117-127, R [3] 132-156
60.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - V			**12**

TOTAL PERIODS : 60

TEXT BOOKS

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
T [1] - 1	Juvinall R.C and Marshek K.M	Fundamentals of Machine Component Design, 5e	John Wiley and Sons, New Delhi	2015
T [2] - 2	Bhandari V. B	Design of Machine Elements, 4e	Tata McGraw–Hill Book Co, New Delhi	2016

REFERENCES

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
R [1] - 1	Norton R. L	Design of Machinery	Tata McGraw–Hill Book Co., New Delhi	2011
R [2] - 2	Orthwein W	Machine Component Design	Jaico Publishing Co., New Delhi	2013
R [3] - .	Bhandari V B	Introduction to Machine Design	McGraw–Hill Book Co., New York	2013
R [4] - 4	Spotts M.F, Shoup T. E	Design of Machine Elements	Pearson Education, New Delhi	2008

WEBSITES

W [1] - nptel.iitk.ac.in/courses/Webcourse-contents/.../Module-3_lesson-2.pdf

W [2] - web.itu.edu.tr/temizv/VTDN/4_Fatigue.pdf
W [3] - www.ignou.ac.in/upload/Unit-7-60
W [4] - www.staff.zu.edu.eg/matta/userdownloads/My%20Courses/.../ch7.pdf
W [5] - www.engr.sjsu.edu/youssefi/me154/notes/Threaded%20Fasteners.pdf
W [6] - nptel.iitk.ac.in/courses/.../Machine%20design1/pdf/mod10les4.pdf
W [7] - nptel.ac.in/courses/IIT-MADRAS/Machine_Design_II/pdf/4_3.pdf
W [8] - www.cpo.com/ipcres/pdfs/unit2/Ch4Sec2.pdf
W [9] - www.engr.sjsu.edu/youssefi/me157/notes/Bearings.ppt
W [10] - turbolab.tamu.edu/proc/turboproc/T13/T13179-188.pdf

JOURNALS

J [1] - Feras Darwish, Mohammad Gharaibeh, Ghassan Tashtoush, 2012, “A modified equation for the stress concentration factor in countersunk holes”, European Journal of Mechanics - A/Solids, Volume 36, Pages 94–103.

J [2] - G. Urquiza, J.C. García, J.G. González, L. Castro, J.A. Rodríguez, M.A. Basurto-Pensado, O.F. Mendoza June 2014, “Failure analysis of a hydraulic Kaplan turbine shaft”, Engineering Failure Analysis, Volume 41, Pages 108-117.

J [3] - A. Esderts, J. Willen, M. Kassner, January 2012, “Fatigue strength analysis of welded joints in closed steel sections in rail vehicles”, International Journal of Fatigue, Volume 34, Issue 1, Pages 112-121.

J [4] - Mohamed Taktak, Khalifa Omheni, Abdessattar Aloui, Fakhreddine Dammak, Mohamed Haddar, March 2014, “Dynamic optimization design of a cylindrical helical spring”, Applied Acoustics, Volume 77, Pages 178-183.

J [5] - Gengyuan Gao, Zhongwei Yin, Dan Jiang, Xiuli Zhang, July 2014, “Numerical analysis of plain journal bearing under hydrodynamic lubrication by water”, Tribology International, Volume 75, Pages 31-38

UNIT	Total No. of Periods Planned	Lecture Periods	Tutorial Periods
I	12	10	2
II	12	10	2
III	12	10	2
IV	12	10	2
V	12	10	2
TOTAL	60	50	10

I. CONTINUOUS INTERNAL ASSESSMENT : 40 Marks

(Internal Assessment Tests: 25, Attendance: 5, Assignment:5, Seminar: 5)

II. END SEMESTER EXAMINATION : 60 Marks

TOTAL : 100 Marks

UNIT 1

STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS

Introduction to the design process – factor influencing machine design, selection of materials based on mechanical properties – Direct, Bending and torsional stress equations – Impact and shock loading – calculation of principle stresses for various load combinations, eccentric loading – Design of curved beams – crane hook and 'C' frame – Factor of safety – theories of failure – stress concentration – design for variable loading – Soderberg, Goodman and Gerber relations.

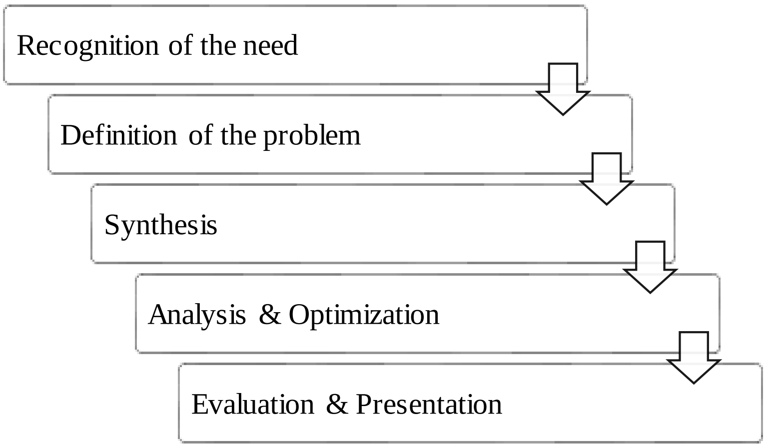
MACHINE DESIGN

Machine design can be defined as applying the basic scientific principles and technical information of machine elements like gears, pulley, fasteners, shaft, bearings, flywheel, keys, welded joints etc to create a machine which complies the basic functions that has to be performed as per the requirement in an efficient and economical way.

Simply design can be defined as the collection of data's or details that are required to convert the designer's idea into reality which in turn satisfies the customer requirements.

DESIGN PROCESS:

In order to create a new design or improvising the existing design of a product, a designer involves into various stages. The following flowchart shows the various stages involved and are called as design process.



The design process starts with the recognition of the customers need. The need analysis is performed with the data collected by the marketing team. This is a crucial stage as it provides the basic knowledge for starting the design.
The next stage of design process involves the collection of information related to mechanisms, physical arrangement, specifications (material, dimensions, bill of materials, drawings, cost, manufacturing details, operating condition, expected life, reliability, performance etc,.), standards, codes (design & Safety) and ends up with feasibility study.
In synthesis stage, the designer combines separate parts to produce an overall new idea or concept by using various new and old ideas & concepts. The philosophy, functionality, and uniqueness of the product are determined during synthesis stage.
During this stage, the designer tries to find out the performance level and efficiency of the design created so far by conducting necessary tests. If the design fails it leads back to the synthesis stage. Hence the analysis & optimization process goes in hand with the synthesis stage. This stage ends up in providing a prototype of the design.
The prototype produced undergoes testing and evaluation process to confirm the customer requirements. Several iterations are performed to make necessary alterations to meet the requirements before presenting the design to others.
The last stage of the design process is presenting the design created to the others in an understandable patterns like visual, written or verbal format. Normally this stage gives the detailed drawings of various parts to be manufactured & assembled.

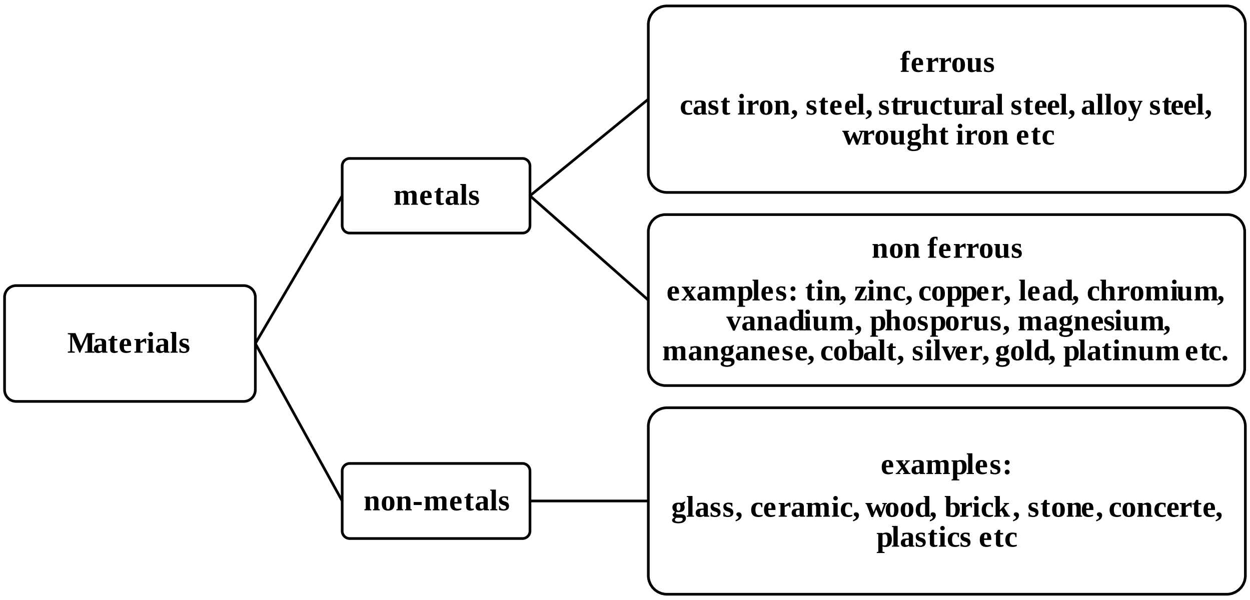
FACTORS INFLUENCING MACHINE DESIGN:

A machine which is to be designed involves various processes and it is influenced by various factors. They are

Strength, Rigidity & Stiffness of the material selected, surface finish, tolerance & fits, Manufacturing conditions, economic conditions, ergonomics, aesthetics, noise, environment, wear rate, reliability etc. while designing a machine it is mandatory to consider these factors for obtaining a better design.

Materials:

Materials are majorly classified into metals & non-metals.



From the broad classification of materials, it is required to select a proper material that will suit the design. The following four factors can be considered for selection of material

a. Availability,
b. Cost,
c. Manufacturability &
d. Mechanical properties.

The following are the various mechanical properties of the material

- Elasticity - ability of the material to get stretched and regains its original shape after removal of the load
- Plasticity – ability of the material to retain its deformed shape after removal of the load
- Ductility – ability of the material to get elongated under the action of tensile load
- Malleability – ability of the material to get compressed under the action of compressive load
- Resilience – ability of the material to absorb energy and release it under elastic limit
- Toughness – ability of the material to absorb maximum energy before the sign of crack or failure
- Hardness – ability of the material to resist penetration or scratches
- Brittleness – ability of the material to break without any plastic deformation

DIRECT, BENDING & TORSIONAL STRESS EQUATIONS (PSGDB pg.no. 7.1)

Stress can be defined as the resisting force offered by the material per unit cross sectional area normal to the direction of the applied external force and it can be expressed as

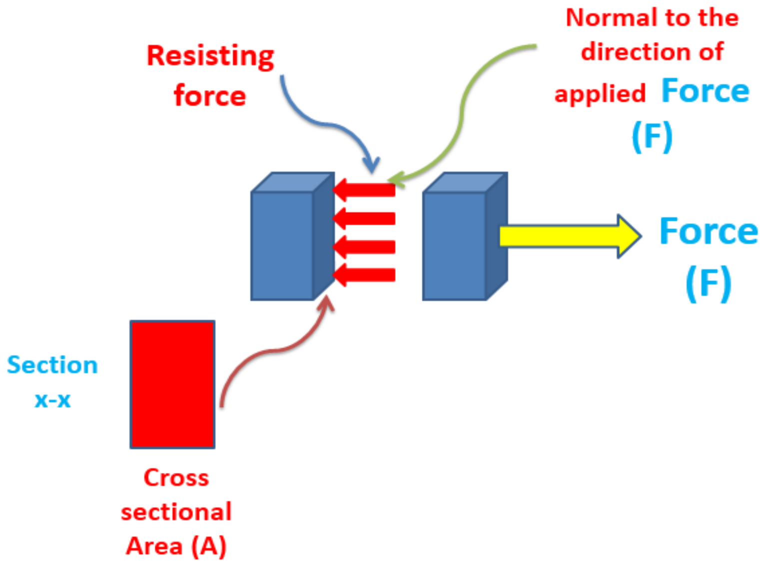
$$\sigma = \frac{external\ load}{cross\ sectional\ area} = \frac{P}{A}\ ,\quad N/mm^2$$

The direct stress can be tensile or compressive in nature based on the nature of load applied. If the force applied is an axial pull force material produces tensile stress. If the force is an axial push force it creates compressive stress.

$$Tensile\ stress, \sigma_t = \frac{Tensile\ force}{cross\ sectional\ area}, \qquad N/mm^2$$

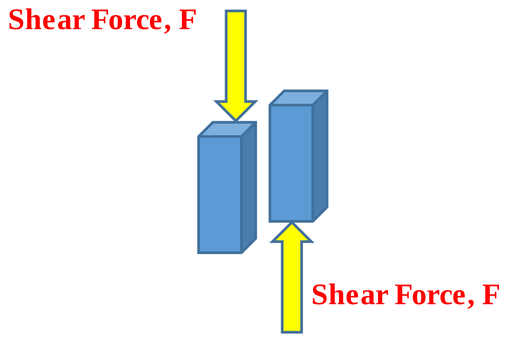
Similarly

$$Compressive\ stress, \sigma_c = \frac{compression\ force}{cross\ sectional\ area}, \qquad N/mm^2$$



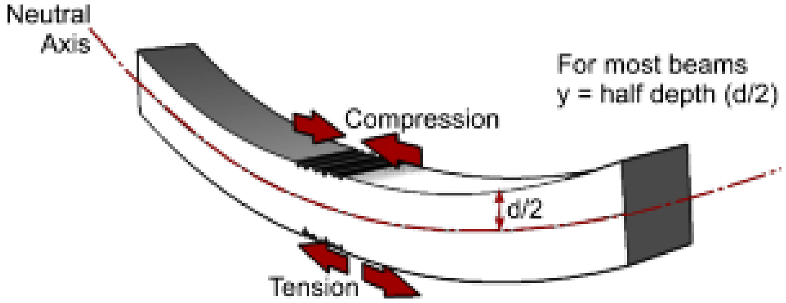
If the direction of the applied force is parallel to the cross section of the part, then it is subjected to transverse shear stress.

$$\tau = \frac{shear\ force}{cross\ sectional\ area}, \qquad N/mm^2$$



When a material is subjected to bending load (bending moment) it produces bending stress and it is given as

$$\sigma_b = \frac{M_b\ y}{I}, \qquad N/mm^2$$



Where

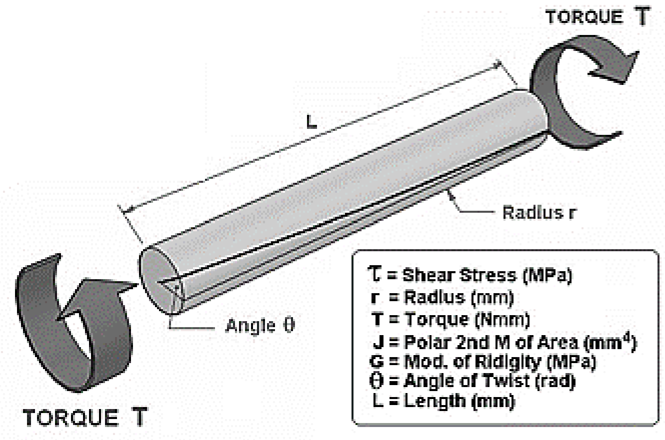
y – Distance of the centroidal axis from the top or bottom layer of the cross section, mm
I – Area moment of inertia of the cross section, mm^4

When a material is subjected to torsional load (Torque or twisting moment) it produces torsional shear stress and it is given as

$$\tau = \frac{T \times r}{J}, \qquad N/mm^2$$

Where

r – Radius of the circular cross section, mm
J – Polar moment of inertia of the cross section, mm^4



ECCENTRIC LOADING: (PSGDB pg.no. 7.1)

When an object is subjected to eccentric loading it produces combination of stresses (Tensile or compressive stress & bending stress)

$$\sigma = \frac{P}{A} \pm \frac{P\,e}{z}$$

Where $\qquad z = \frac{I}{y}$

P – External applied load, N

A – Cross sectional area, mm^2

e – Eccentric distance form central axis to the load applied, mm

z – Section modulus, mm^3

Section modulus is the ratio of area moment of inertia of the cross section to the distance of the top or bottom layer of the cross section from the centroidal axis.

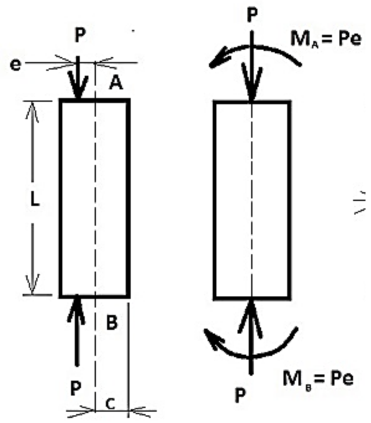
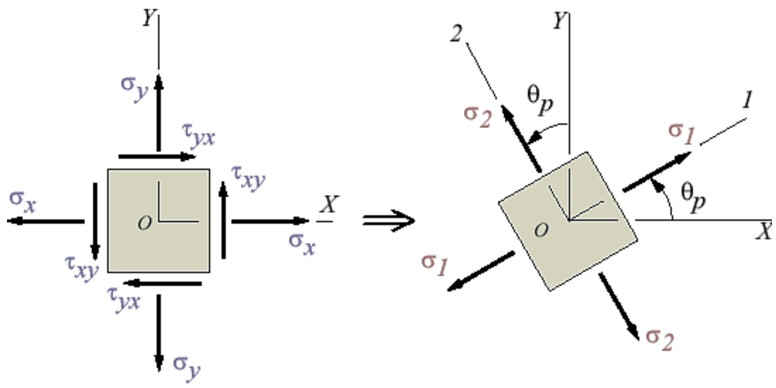


Fig. Eccentric Loading

IMPACT & SHOCK LOADING

PRINCIPAL STRESSES: (PSGDB pg.no. 7.2)

When a part is subjected to biaxial or triaxial stress conditions, it is required to calculate equivalent stress and they are called as the principal stresses. The maximum & minimum principal stresses are given as



Maximum & minimum principal normal stresses

$$\sigma_{1,2} = \frac{1}{2}\left\{\left(\sigma_x + \sigma_y\right) \pm \sqrt{\left(\sigma_x^2 - \sigma_y^2\right) + 4\,\tau_{xy}}\right\}$$

Maximum & minimum principal shear stresses

$$\tau_{1,2} = \pm\frac{1}{2}\sqrt{\left(\sigma_x^2 - \sigma_y^2\right) + 4\,\tau_{xy}}$$

DESIGN OF CURVED BEAMS: (PSGDB pg.no. 6.2 & 6.3)

A curved beam can be defined as a beam in which the neutral axis of the beam is curved even under unloaded condition. The curved beams are subjected to bending stress and it is given as

$$\sigma_b = \frac{M_b\, y}{a\, e\, (r_n - y)}, \qquad N/mm^2$$

The maximum bending stress acts on the inner and outer fibre are given as

$$\sigma_{bi} = \frac{M_b\, h_i}{a\, e\, r_i}, \qquad N/mm^2$$

$$\sigma_{bo} = \frac{M_b\, h_o}{a\, e\, r_o}, \qquad N/mm^2$$

Where

h_i & h_o – Distance of the N.A from the inner & outer fibre, mm

$$h_i = r_n - r_i, \qquad mm$$

$$h_o = r_o - r_n, \qquad mm$$

r_i & r_o – radius of the inner & outer fibre of the curved beam, mm

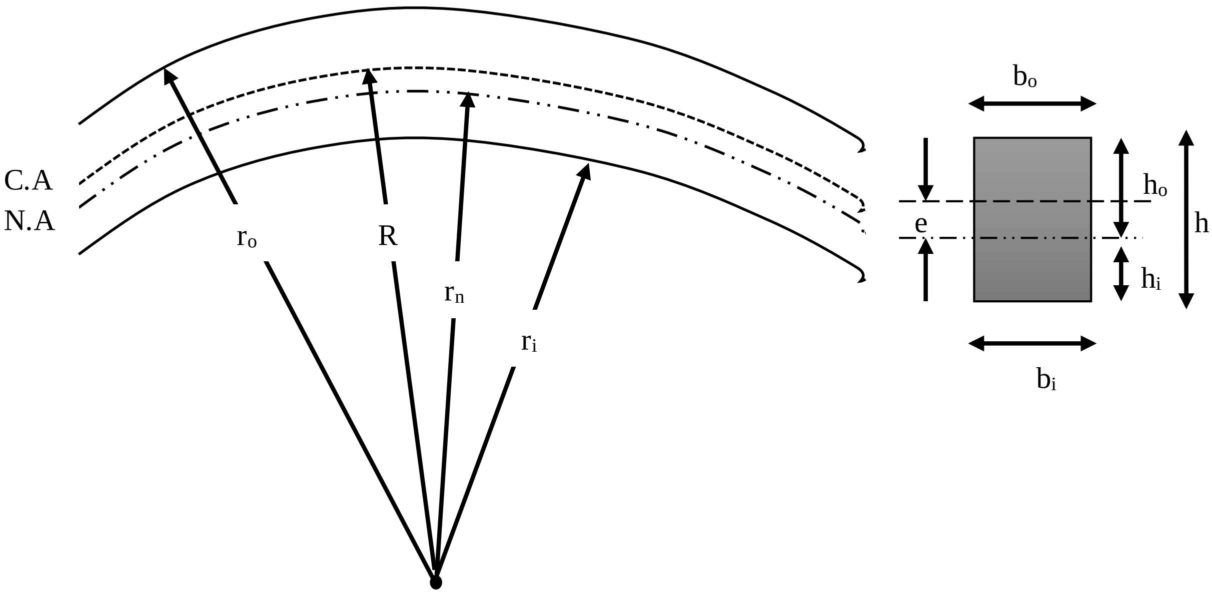
$$r_o = r_i + h, \qquad mm$$

e - Eccentricity in the position of N.A & C.A

$$e = R - r_n, \qquad mm$$

R - Radius of the Centroidal axis (C.A), mm

r_n – Radius of the Neutral axis (N.A), mm



Factor of Safety, n

It is defined as the ratio of maximum stress to the working stress of the material.

$$n = \frac{maximum\ failure\ stress}{working\ or\ permissible\ stress}$$

The failure strength of the ductile material is the yield strength of it and for the brittle material the failure strength is the ultimate strength.

$$for\ ductile\ material, \qquad n = \frac{yield\ strength}{permissible\ stress} = \frac{\sigma_y}{\sigma_p}$$

$$for\ Brittle\ material, \qquad n = \frac{ultimate\ strength}{permissible\ stress} = \frac{\sigma_u}{\sigma_p}$$

Applications	Factor of Safety - FOS -
For use with highly reliable materials where loading and environmental conditions are not severe and where weight is an important consideration	1.3 - 1.5
For use with reliable materials where loading and environmental conditions are not severe	1.5 - 2
For use with ordinary materials where loading and environmental conditions are not severe	2 - 2.5
For use with less tried and for brittle materials where loading and environmental conditions are not severe	2.5 - 3
For use with materials where properties are not reliable and where loading and environmental conditions are not severe, or where reliable materials are used under difficult and environmental conditions	3 - 4

Stress Concentration: PSGDB pg.no. 7.8

It is defined as the localization of high stresses due to abrupt changes in cross section or irregularities present in the component. To consider this effect in design, a factor known as stress concentration factor is used. It is defined as the ratio of maximum stress to the nominal stress at the net section.

$$k_t = \frac{maximum\ stress}{nominal\ stress\ at\ the\ net\ section} = \frac{\sigma_{max}}{\sigma_{nom}}$$

The subscript ‘t’ represents the theoretical stress concentration factor.

Reasons for stress concentration:

1. Variation in the material properties (due to internal cracks, flaws like blow hole, cavities in weld, air holes, foreign material inclusions etc.)
2. Load application (due to load concentration on small area)
3. Abrupt changes in cross section
4. Discontinuities in the component (oil holes, oil grooves, keyways, splines etc.)
5. Machining scratches (stamp marks, inspection marks, surface irregularities)

Reduction of stress concentration:

It is not possible to eliminate the stress concentration, but it can be reduced to certain level. The following methods can be used to reduce the effect of stress concentration.

1. By providing additional notches and holes for members under tension.
2. By providing fillet radius, undercut and notch for members under bending load
3. By drilling additional holes for shaft
4. By either reducing the shank dimeter of the threaded component or by drilling hole in the axial part of shank portion.

The theoretical stress concentration factor, k_t can be found for some standard components subjected to axial, bending and torsion load from graphs provided in PSGDB pg.no. 7.9 to 7.17.

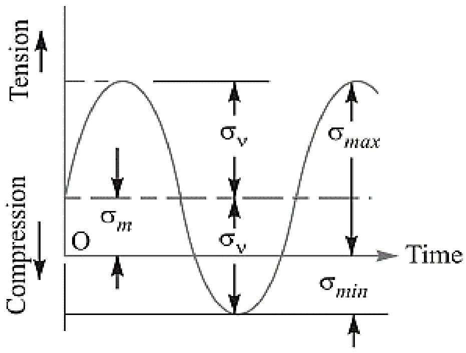
Variable stresses or Fluctuating stresses:

In many engineering applications the load applied is not static. They vary with respect to time. The stresses induced due to such loading is termed as fluctuating stresses or variable stresses. It can be observed that about 80% of failure of the mechanical components are due to fatigue failure resulting from the fluctuating stresses. For the study purpose and design analysis, simple models of stress-time relationship are used. The most popular model among all is the sine curve.

Types of mathematical models for cyclic stresses:

1. **Fluctuating or Alternating stress cycle:**

   In this model the stress varies with time in sinusoidal form and it has its mean value as well as amplitude value as it varies between two limits – maximum and minimum stress value.

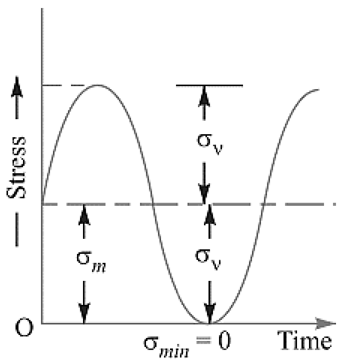


$$Mean\ stress, \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$Amplitude\ stress, \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

2. **Repeated stress cycle:**

   In this case the stress variation is in the form of sinusoidal pattern. But the minimum stress is always zero in this stress cycle.



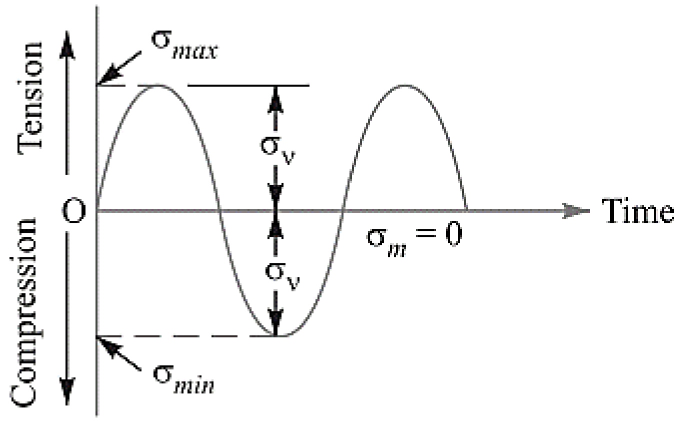
$$Mean\ stress, \sigma_m = \frac{\sigma_{max}}{2}$$

$$Amplitude\ stress, \sigma_a = \frac{\sigma_{max}}{2}$$

Hence, $\sigma_m = \sigma_a$ for repeated stress cycle, since $\sigma_{min} = 0$

3. **Completely reversed stress cycle:**

   In this cyclic stress pattern, the minimum stress value is equal and opposite of the maximum stress value in the cycle.



$$Mean\ stress, \sigma_m = 0$$

$$Amplitude\ stress, \sigma_a = \sigma_{max}$$

For completely reversed stress cycle, $\sigma_{min} = -\sigma_{max}$

Fatigue failure:

It is defined as the time delayed fracture under cyclic loading.

Endurance limit:

The fatigue or endurance limit of material is defined as the maximum stress amplitude of completely reversed stress cycle that the standard specimen can withstand for an unlimited number of cycles without fatigue failure. Since the fatigue test cannot be conducted for unlimited number of cycles, the number of cycles is limited to 106 cycles.

Fatigue life:

It is defined as the number of stress cycles that the standard specimen can complete during the test before the appearance of the first fatigue crack.

S-N curve:

It is the graphical representation of stress amplitude versus the number of stress cycles (N) before fatigue failure on a log-log graph paper.

Fatigue Stress Concentration Factor

When a machine member is subjected to cyclic or fatigue loading, the value of fatigue stress concentration factor shall be applied instead of theoretical stress concentration factor. Since the determination of fatigue stress concentration factor is not an easy task, therefore from experimental tests it is defined as

$$k_f = \frac{\text{Endurance limit without stress concentration}}{\text{Endurance limit with stress concentration}}$$

Notch Sensitivity

In cyclic loading, the effect of the notch or the fillet is usually less than predicted using the theoretical factors as discussed before. The difference depends upon the stress gradient in the region of the stress concentration and on the hardness of the material. The term notch sensitivity is applied to this behaviour.

It is defined as the ability of the material to withstand the damaging effects of stress raising notches in fatigue loading. The notch sensiticity factor is defined as

$$q = \frac{\text{Increase in actual stress over the nominal stress}}{\text{increase in theoretical stress over the nomial stress}}$$

$$q = \frac{k_f \sigma_{nom} - \sigma_{nom}}{k_t \sigma_{nom} - \sigma_{nom}}$$

$$q = \frac{k_f - 1}{k_t - 1}$$

$$k_f = 1 + q\,(k_t - 1)$$

Endurance limit approximation:

The endurance limit of components is different from the endurance limit of the rotating beam specimen, this is corrected using certain factors known as derating factors. The derating factors include size factor, load factor, surface finish factor.

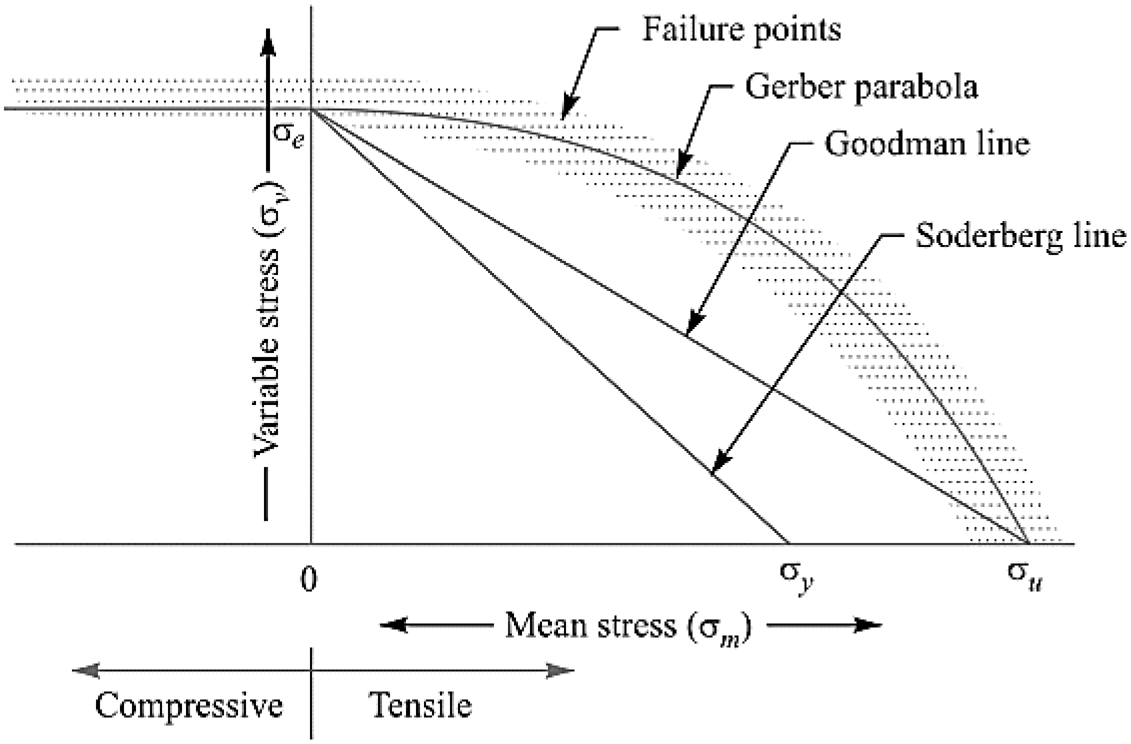
From PSGDB pg.no. 1.42, standard relationships are available to calculate the endurance limit of the material under study. Then in PSGDB pg.no 1.43, the relation for calculating approximate endurance limit is given.

$$\sigma_{-1} = \frac{\sigma_{-1}\, A\, B\, C}{k_f}$$

Where, A is the load factor, B is the size factor, C is the surface finish factor. The factor values are listed for different cases in the same page.

Design for Fatigue loading:

The failure points from fatigue tests made with different steels and combinations of mean and variable stresses are plotted in Fig. as functions of amplitude stress (σ_a) and mean stress (σ_m). The most significant observation is that, in general, the failure point is little related to the mean stress when it is compressive but is very much a function of the mean stress when it is tensile. In practice, this means that fatigue failures are rare when the mean stress is compressive (or negative). Therefore, the greater emphasis must be given to the combination of a variable stress and a steady (or mean) tensile stress.



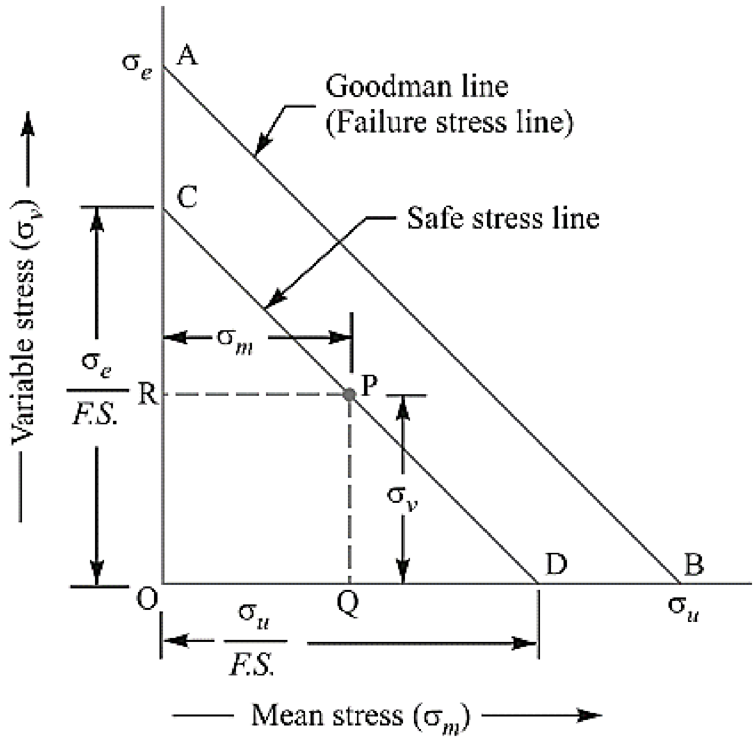
There are several ways in which problems involving this combination of stresses may be solved, but the following are important from the subject point of view :

1. Gerber method, 2. Goodman method, and 3. Soderberg method.

Goodman Method for Combination of Stresses:

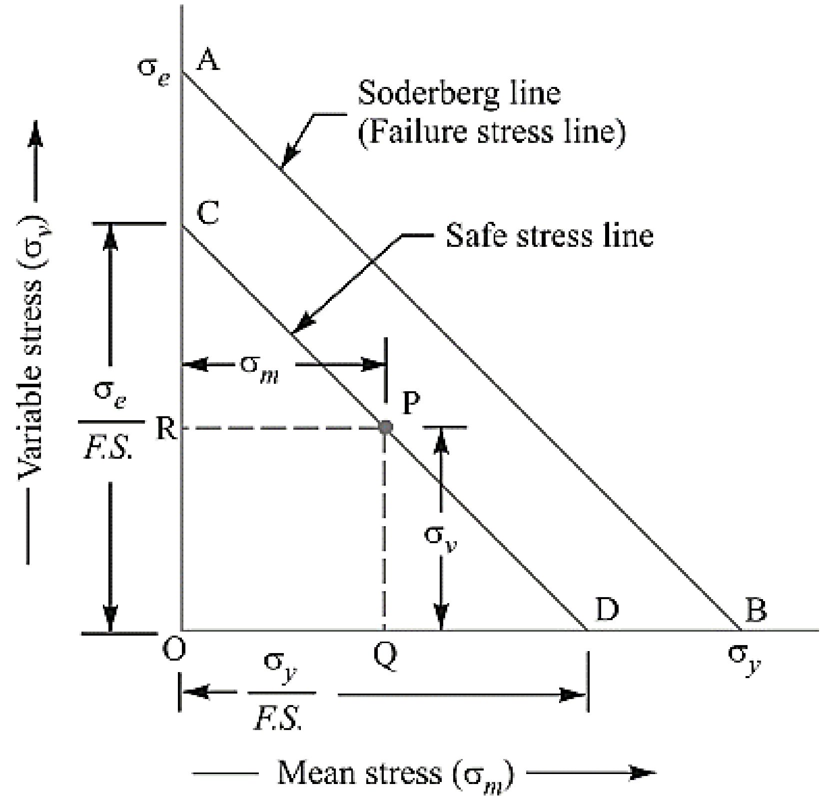
A straight line connecting the endurance limit (σ e) and the ultimate strength (σ u), as shown by line AB in Fig. 6.16, follows the suggestion of Goodman. A Goodman line is used when the design is based on ultimate strength and may be used for ductile or brittle materials.

line AB connecting σ e and σ u is called Goodman's failure stress line. If a suitable factor of safety (F.S.) is applied to endurance limit and ultimate strength, a safe stress line CD may be drawn parallel to the line AB.



Soderberg Method for Combination of Stresses

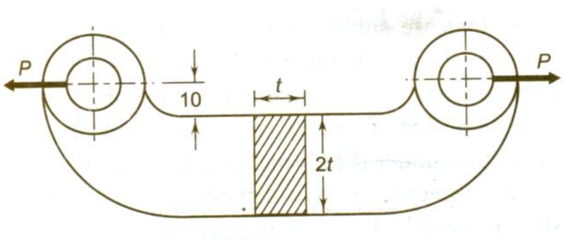
A straight line connecting the endurance limit (σ_{-1}) and the yield strength (σ_Y), as shown by the line AB in Fig. 6.17, follows the suggestion of Soderberg line. This line is used when the design is based on yield strength. The line AB connecting σ_{-1} and σ_Y , as shown in Fig, is called Soderberg's failure stress line. If a suitable factor of safety (F.S.) is applied to the endurance limit and yield strength, a safe stress line CD may be drawn parallel to the line AB.



Problems:

COMBINED LOADING (Principle of superposition concept)

1. An offset link subjected to a force of 25 kN is shown in fig. It is made of gray cast iron FG300 and the factor of safety is 3. Determine the dimensions of the cross section of the link.



Given Data:

Load, P = 25 kN = 25 x 10^3 N

Ultimate strength of the grey cast iron, σ_u = 300 N/mm^2

Factor of safety, n = 3

To find:

Dimensions of the cross section

Solution:

The offset link shown in the fig is subjected to tensile and bending stress, therefore the total stress acting on the link is given as

$$\sigma = \sigma_t + \sigma_b$$

Hence

$$\sigma = \frac{P}{A} + \frac{M_b\, y}{I}$$

Where the bending moment, M_b = Load x perpendicular distance

$$M_b = P \times (10 + t)$$

$$M_b = 25 \times 10^3 \times (10 + t)$$

The area moment of inertia of the given cross section is,

$$I = \frac{b\,d^3}{12} = \frac{t \times (2t)^3}{12} = \frac{2\,t^4}{3}\ mm^4$$

Therefore

$$\sigma = \frac{25 \times 10^3}{t \times 2t} + \frac{25 \times 10^3 \times (10 + t) \times \frac{2t}{2}}{\frac{2\,t^4}{3}}$$

On simplification

$$\sigma = \frac{50 \times 10^3\,t^3 + 37.5 \times 10^4\,t^2}{t^5}$$

The permissible stress is

$$\sigma = \frac{\sigma_u}{n} = \frac{300}{3} = 100\ N/mm^2$$

Therefore

$$100 = \frac{50 \times 10^3\,t^3 + 37.5 \times 10^4\,t^2}{t^5}$$

$$100\,t^5 - 50 \times 10^3\,t^3 - 37.5 \times 10^4\,t^2 = 0$$

Dividing the above equation by t^2

$$100\,t^2 - 50 \times 10^3\,t - 37.5 \times 10^4 = 0$$

On solving the above cubic equation, we get

$$t = 25.5\ mm$$

Result:

The cross sectional dimension of the given section in the offset link is determined as

$$b = t = 25.5\,mm \cong 26\ mm$$

$$d = 2t = 52\ mm$$

2. Determine the diameter of a steel bar, which is of ductile nature subjected to an axial tensile load of 60 kN and torsional moment of 1600 N-m. Use the factor of safety of 2.5, E = 200 Gpa.

Given data:

Axial tensile load, P = 60 kN

Torsional moment, T = 1600 Nm

Factor of safety, n = 2.5

To Find:

The diameter of the steel bar, d =?

Solution:

The given steel bar is subjected to tensile stress due to axial tensile load and torsional shear stress due to the torsional moment, hence the total stress produced in the steel rod is given as

$$\sigma = \sigma_t + \tau$$

And

$$\sigma = \frac{P}{A} + \frac{T\,r}{J}$$

The permissible stress of the steel bar is taken from the PSGDB pg.no 1.9, for C15 steel the yield stress is given as

$$\sigma_y = 240\ N/mm^2$$

Therefore

$$\sigma = \frac{\sigma_y}{n} = \frac{240}{2.5} = 80\ N/mm^2$$

On substituting the values,

$$80 = \frac{60\ \times 10^3}{\frac{\pi}{4}\ d^2} + \frac{1600\ \times 10^3 \times \frac{d}{2}}{\frac{\pi}{32}\ d^4}$$

On simplification

$$80\ d^5 = 76394.37\ d^3 + 8.149\ \times 10^6\ d^2$$

Dividing the above equation by d^2

$$80\ d^3 - 76394.37\,d - 8.149\ \times 10^6 = 0$$

On solving the above cubic equation,

$$d = 53.48 \cong 54\ mm$$

Result:

The diameter of the steel rod is determined as

$$d = 53.48 \cong 54\ mm$$

3. A cantilever of span 800 mm carries uniformly distributed load of 12 kN/m. The yield value of material of cantilever is 400 MPa. Factor of safety is 2.5. Find economical section of cantilever among
   i) Circular cross section of diameter‘d’.
   ii) Rectangular cross section of depth‘d’ and width ‘w’ with d/w = 2.5.
   iii) ‘I’ section of total depth 7t and width 5t where‘t’ is thickness.

Find the dimension and cross sectional area of the economic section.

Given:

Length of the beam, l = 800 mm

Uniformly distributed load = 20 kN/m

Yield stress, σ_y = 400 MPa = 400 N/mm^2

Factor of safety, n = 2.5

To Find:

Economical section for the cantilever beam

Whether circular cross section or rectangular section of depth ‘d’ and width ‘w’ with a d/w = 2.5 or I section of total depth 7t and width 5t where ‘t’ is the thickness

Solution:

The cantilever beam is subjected to bending moment, M_b due to the uniformly distributed load, hence it has bending stress, σ_b

To find the bending moment acting on the cantilever beam, use the formula from PSGDB pg.no 6.4,

For a cantilever beam loaded with uniformly distributed load,

$$M_b = \frac{w\,l^2}{2} = \frac{20 \times 10^3 \times 800^2}{2}$$

$$M_b = 6.4 \times 10^9\ N - mm$$

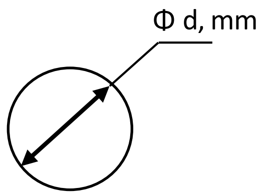
Hence the bending stress,

$$\sigma_b = \frac{M_b\, y}{I}$$

The permissible stress for the given material is given as

$$\sigma = \frac{\sigma_y}{n} = \frac{400}{2.5}$$

$$\sigma = 160\ N/mm^2$$



Case (i) circular cross section

$$160 = \frac{6.4 \times 10^9 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4}$$

On simplifying and solving,

$$d = 741.34\ mm$$

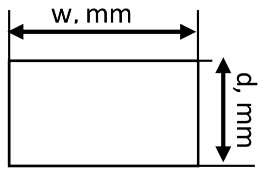
The cross sectional area of the circle is

$$A = \frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times 741.34^2$$

$$A = 431643.04\ mm^2$$

Case (ii) Rectangular cross section

Given, d/w = 2.5 ⇒ d = 2.5 w



$$160 = \frac{6.4 \times 10^9 \times \frac{d}{2}}{\frac{w\,d^3}{12}}$$

$$160 = \frac{6.4 \times 10^9 \times \frac{2.5\,w}{2}}{\frac{w\,(2.5\,w)^3}{12}}$$

On simplifying and solving,

$$w = 337.37\ mm\ and\ d = 843.43\ mm$$

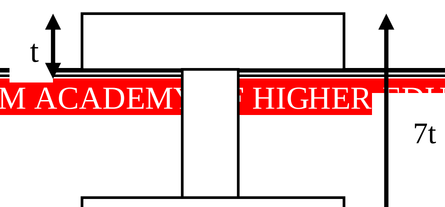
The cross sectional area of the rectangle is

$$A = w \times d = 337.37 \times 843.43$$

$$A = 284547.97\ mm^2$$

Case (iii) I – section

To find the area moment of inertia,



$$I = \frac{B\,D^3}{12} - \frac{b\,d^3}{12}$$

$$I = \frac{5t \times 7t^3}{12} - \frac{4t \times 5t^3}{12}$$

$$I = \frac{15t^4}{12}\,mm^4$$

Then,

$$160 = \frac{6.4 \times 10^9 \times \frac{7t}{2}}{\frac{15t^4}{12}}$$

On simplifying and solving,

$$t = 482.03\,mm$$

The cross sectional area of I – section is

$$A = (7t \times 5t) - (4t \times 5t) = 15\,t^2 = 15 \times 482.03^2$$

$$A = 3485271.44\,mm^2$$

Result:

Out of the above three cross sectional area, the cross sectional of the rectangular section is the lowest, hence the economical section for the given cantilever beam is Rectangular section

The cross sectional area of the rectangle is

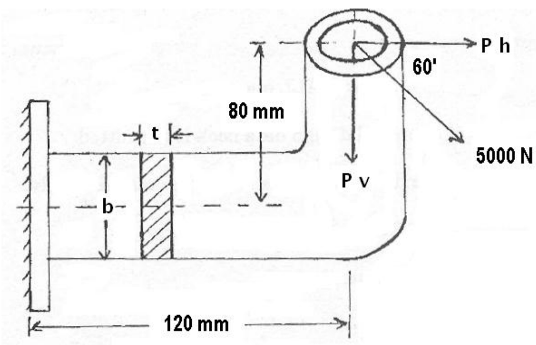
$$A = 284547.97\,mm^2$$

4. For mounting worktop to a cabinet, an angle bracket as shown in figure below is used. Design angle bracket for the following requirements to serve for the above purpose.

Static load - 5000 N acting at 60° to its horizontal axis.

Cross section - rectangular (b = 2t)

Material used - Plain carbon steel (45C8) Factor of safety - 2



Given Data:

Load acting on the angle bracket, P = 5000 N @ ∠ 60°

Rectangular cross section with b = 2t

Material used = 45C8

Factor of safety, n = 2

To Find:

Cross sectional dimensions of the given section

Solution

From the given fig it is clear that, the angle bracket is subjected to tensile stress due to the horizontal load, P_h and bending stress due to the bending moment of the P_h and P_v forces.

Resolving the given load into horizontal and vertical components

$$P_h = P\ \cos\theta = 5000 \times \cos 60° = 2500\ N$$

$$P_v = P\ \cos\theta = 5000 \times \sin 60° = 4330.13\ N$$

The tensile stress acting on the bracket,

$$\sigma_t = \frac{P_h}{A} = \frac{2500}{b \times t} = \frac{2500}{2t \times t}$$

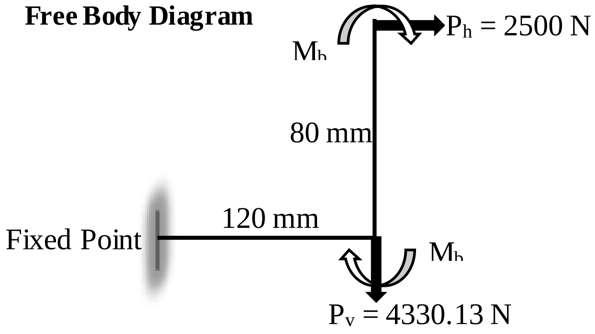
$$\sigma_t = \frac{1250}{t^2}\ N/mm^2$$

The bending stress acting on the bracket,

$$\sigma_b = \frac{M_b\, y}{I}$$

The bending moment M_b is

Free Body Diagram



$$M_b = (P_h \times 80) + (P_v \times 120)$$

$$M_b = (2500 \times 80) + (4330.13 \times 120)$$

$$M_b = 719615.6\ N.mm$$

Therefore the bending stress

$$\sigma_b = \frac{M_b\, y}{I}$$

$$\sigma_b = \frac{M_b \times \frac{b}{2}}{\frac{bt^3}{12}} = \frac{719615.6 \times \frac{2t}{2}}{\frac{2t \times t^3}{12}}$$

On simplifying and solving,

$$\sigma_b = \frac{4.318 \times 10^6}{t^3}\ N/mm^2$$

The permissible stress for the given material 45C8 is

$$\sigma = \frac{\sigma_y}{n}$$

Where σ_y is the yield stress of the material taken from PSGDB pg.no 1.9 as

$$\sigma_y = 360 N/mm^2$$

Therefore

$$\sigma = \frac{\sigma_y}{n} = \frac{360}{2}$$

$$\sigma = 180\ N/mm^2$$

This permissible stress must be equal to the sum of the stresses acting on the bracket, hence

$$\sigma = \sigma_t + \sigma_b$$

$$\sigma = \frac{1250}{t^2} + \frac{4.318 \times 10^6}{t^3}$$

On simplifying the above equation a cubic equation shown below is obtained

$$180\,t^3 - 1250\,t - 4.318 \times 10^6 = 0$$

On solving the above cubic equation,

$$t = 28.92 \cong 29\,mm$$

Result:

The cross sectional dimensions are

$$b = 58\,mm\,\&\,t = 29\,mm$$

THEORIES OF FAILURE

1. A bolt is subjected to an axial force of 10000 N with a transverse shear force of 5000 N. Find the diameter of the bolt required using
   i) Maximum principal stress theory
   ii) Maximum shear stress theory
   iii) Maximum principal strain theory
   iv) Maximum strain energy theory

   Take the factor of safety as 2 and Poisson's ratio as 0.3.

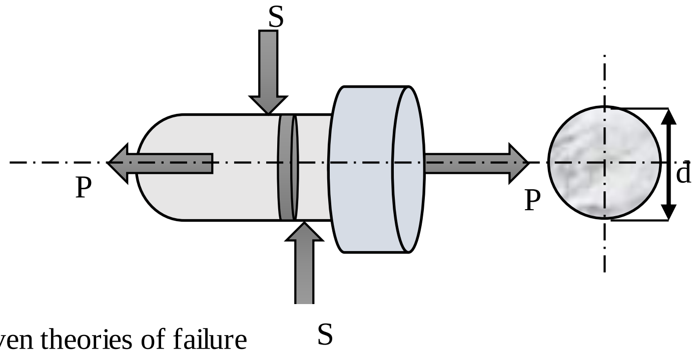
Given Data:

Axial force, P = 10000 N

Transverse shear force, S = 5000 N

Factor of safety, n = 2

Poisson's ratio, ν = 0.3



To Find:

Diameter of the bolt section using given theories of failure

Solution:

The given bolt is subjected to axial and transverse shear forces, hence this bolt is subjected to axial tensile stress and shear stress

Axial tensile stress,

$$\sigma_t = \frac{P}{A} = \frac{10000}{\frac{\pi}{4} \times d^2}$$

$$\sigma_t = \frac{12732.39}{d^2} N/mm^2$$

The transverse shear stress,

$$\tau = \frac{P}{A} = \frac{5000}{\frac{\pi}{4} \times d^2}$$

$$\tau = \frac{6366.19}{d^2} N/mm^2$$

To find the principal stresses use the formulas from PSGDB pg.no 7.2

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\,\tau_{xy}^2}\right]$$

Here

$$\sigma_x = \sigma_t = \frac{12732.39}{d^2} N/mm^2 \; and \; \tau_{xy} = \frac{6366.19}{d^2} N/mm^2 \; also \; \sigma_y = 0$$

Therefore on substituting the above values in principal stress equations we get

$$\sigma_{1,2} = \frac{1}{2}\left[\left(\frac{12732.39}{d^2}\right) \pm \sqrt{\left(\frac{12732.39}{d^2}\right)^2 + 4\left(\frac{6366.19}{d^2}\right)^2}\right]$$

On simplification

$$\sigma_1 = \frac{15369.35}{d^2} N/mm^2$$

$$\sigma_2 = \frac{1318.48}{d^2} N/mm^2$$

The permissible stress for the given material 45C8 is

$$\sigma = \frac{\sigma_y}{n}$$

Where σ_y is the yield stress of the material taken from PSGDB pg.no 1.9 as

$$\sigma_y = 360 N/mm^2$$

Therefore

$$\sigma = \frac{\sigma_y}{n} = \frac{360}{2}$$

$$\sigma = 180 \, N/mm^2$$

From PSGDB pg.no 7.3

(i) According to maximum stress theory

$$\sigma_1 \; or \; \sigma_2 = \sigma = \frac{\sigma_y}{n}$$

$$\frac{15369.35}{d^2} = 180$$

On solving

$$d = 9.24 \, mm$$

(ii) According to shear stress theory

$$\sigma_1 - \sigma_2 = \sigma = \frac{\sigma_y}{n}$$

$$\frac{15369.35}{d^2} - \frac{1318.48}{d^2} = 180$$

On solving

$$d = 8.84 \, mm$$

(iii) According to maximum strain theory

$$\sigma_1 - \nu \, \sigma_2 = \sigma = \frac{\sigma_y}{n}$$

$$\frac{15369.35}{d^2} - 0.3 \times \frac{1318.48}{d^2} = 180$$

On solving

$$d = 9.12 \, mm$$

(iv) According to maximum strain energy theory

$$\sigma_1{}^2 + \sigma_2{}^2 - 2\,\nu\,\sigma_1\,\sigma_2 = \sigma^2 = \left(\frac{\sigma_y}{n}\right)^2$$

$$\left(\frac{15369.35}{d^2}\right)^2 + \left(\frac{1318.48}{d^2}\right)^2 - \left(2\times 0.3 \times \frac{15369.35}{d^2} \times \frac{1318.48}{d^2}\right) = 180^2$$

On solving

$$d = 9.14\ mm$$

(v) According to distortion energy theory

$$\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1\,\sigma_2 = \sigma^2 = \left(\frac{\sigma_y}{n}\right)^2$$

$$\left(\frac{15369.35}{d^2}\right)^2 + \left(\frac{1318.48}{d^2}\right)^2 - \left(\frac{15369.35}{d^2} \times \frac{1318.48}{d^2}\right) = 180^2$$

On solving

$$d = 9.05\ mm$$

2. A Machine part is statically loaded and has an yield point strength of 350 N/mm^2. If the principal stresses are 70 N/mm^2 and 35 N/mm^2, both are tensile. Find the factor of safety for the following cases,

   i) Maximum normal stress theory
   ii) Maximum shear stress theory
   iii) Distortion energy theory

Given Data:

Yield strength, σ_y = 350 N/mm^2

Maximum principal stress, σ_1 = 70 N/mm^2

Minimum principal stress, σ_2 = 35 N/mm^2

To Find:

Factor of safety, n =?

Solution:

From PSGDB pg.no 7.3

(i) According to maximum stress theory

$$\sigma_1\ or\ \sigma_2 = \sigma = \frac{\sigma_y}{n}$$

$$70 = \frac{350}{n}$$

On solving

$$n = 5$$

(ii) According to shear stress theory

$$\sigma_1 - \sigma_2 = \sigma = \frac{\sigma_y}{n}$$

$$70 - 35 = \frac{350}{n}$$

On solving

$$n = 10$$

(iii) According to distortion energy theory

$$\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1\,\sigma_2 = \sigma^2 = \left(\frac{\sigma_y}{n}\right)^2$$

$$(70)^2 + (35)^2 - (70 \times 35) = \left(\frac{350}{n}\right)^2$$

On solving

$$n = 5.17$$

STRESS CONCENTRATION:

1. Taking stress concentration into account determine the maximum stress when it is subjected to axial tensile load of 20 kN for the following
   i) A rectangular plate of width 80 mm and thickness 20 mm with a central transverse hole of diameter 16 mm
   ii) A stepped shaft of diameters 80 mm and 60 mm with a fillet radius of 6 mm.

Given:

Tensile load, P = 20 kN = 20 x 10^3 N

Width of plate, w = 80 mm

Thickness of the plate, h = 20 mm

Diameter of the transverse hole, a = 16 mm

Diameters of the stepped shaft, D = 80 mm & d = 60 mm

Fillet radius, r = 6 mm

To Find:

Maximum stress, σ_{max} =?

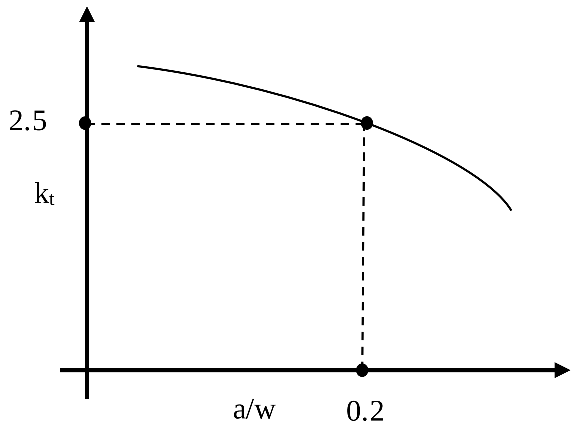
Solution:

From PSGDB pg.no 7.8, the stress concentration factor is given as

$$k_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

Case (i)

From PSGDB pg.no 7.10 from the graph,



$$\frac{a}{w} = \frac{16}{80} = 0.2$$

For the above value, from the graph the stress concentration factor value is selected or plotted as

$$k_t = 2.5$$

w.k.t

$$\sigma_{max} = k_t\,\sigma_{nom}$$

The nominal stress acting on the rectangular plate can be found using the relation

$$\sigma_{nom} = \frac{P}{(w-a)h} = \frac{20 \times 10^3}{(80-16) \times 20}$$

$$\sigma_{nom} = 15.625 \text{ N/mm}^2$$

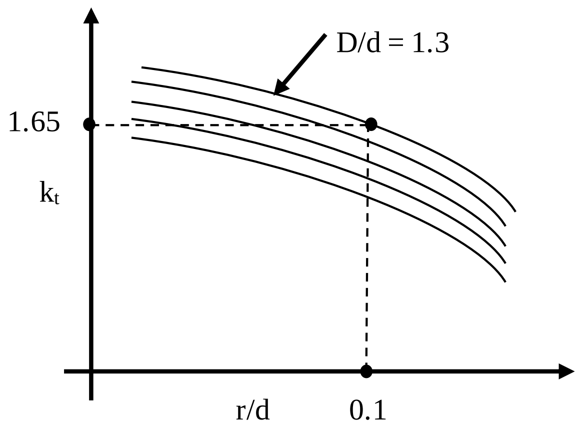
Therefore the maximum stress,

$$\sigma_{max} = 2.5 \times 15.625$$

$$\sigma_{max} = 39.06 \text{ N/mm}^2$$

Case (ii)

From PSGDB pg.no 7.11 from the graph



$$\frac{r}{d} = \frac{6}{60} = 0.1$$

$$\frac{D}{d} = \frac{80}{60} = 1.33$$

For the above value, from the graph the stress concentration factor value is selected or plotted as

$$k_t = 1.65$$

w.k.t

$$\sigma_{max} = k_t \, \sigma_{nom}$$

The nominal stress acting on the rectangular plate can be found using the relation

$$\sigma_{nom} = \frac{P}{\frac{\pi}{4} d^2} = \frac{20 \times 10^3}{\frac{\pi}{4} \times 60^2}$$

$$\sigma_{nom} = 7.074 \text{ N/mm}^2$$

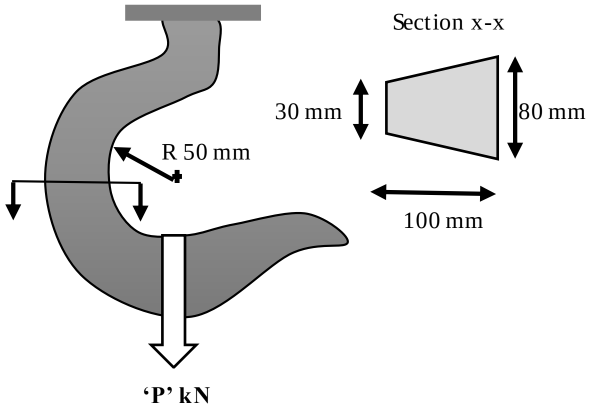
Therefore the maximum stress,

$$\sigma_{max} = 1.625 \times 7.074$$

$$\sigma_{max} = 11.5 \text{ N/mm}^2$$

DESIGN OF CURVED BEAMS

1. A crane hook having an approximate trapezoidal cross section is shown in the figure. It is made of 45C8 steel and the factor of safety is 3. Determine the load carrying capacity of the crane hook.



Given:

Material – 45C8 Steel

Factor of safety, n = 3

From Figure

Radius of inner fibre, r_i = 50 mm

Radius of outer fibre, $r_o = r_i + h$ = 150 mm

Cross section dimension, b_i = 80 mm, b_o = 30 mm, h = 100 mm

To Find:

Load carrying capacity, 'P' =?

Solution:

The permissible stress for the given material 45C8 is

$$\sigma = \frac{\sigma_y}{n}$$

Where σ_y is the yield stress of the material taken from PSGDB pg.no 1.9 as

$$\sigma_y = 360 N/mm^2$$

Therefore

$$\sigma = \frac{\sigma_y}{n} = \frac{360}{3}$$

$$\sigma = 120\ N/mm^2$$

The crane hook is subjected to tensile stress and bending stress due to the applied external load, therefore

(i) Tensile stress, σ_t

$$\sigma_t = \frac{P}{a}$$

Cross sectional area of the trapezoid is given as

$$a = \frac{1}{2}(b_i + b_o)h = \frac{1}{2} \times (80 + 30) \times 100$$

$$a = 5500\ mm^2$$

Hence

$$\sigma_t = \frac{P}{5500}$$

$$\sigma_t = 1.82 \times 10^{-4} \times P \quad N/mm^2$$

(ii) Bending stress, σ_b

The bending stress will be maximum in the inner fibre so let's find the bending stress of the inner fibre, from PSGDB pg.no 6.2

$$\sigma_{bi} = \frac{M_b h_i}{a\, e\, r_i}$$

h_i – distance of the neutral axis from the inner fibre and it is given as

$$h_i = r_n - r_i$$

Where r_n is the radius of the neutral axis

Bending moment is given as

$$M_b = P \times R$$

Where R – radius of the centroidal axis

The r_n and R can be determined using the formulas from PSGDB pg.no 6.3

$$R = r_i + \frac{h\,(b_i + 2\,b_o)}{3\,(b_i + b_o)}$$

$$R = 50 + \frac{100 \times (80 + 2 \times 30)}{3\,(80 + 30)}$$

$$\boldsymbol{R = 92.42\ mm}$$

$$r_n = \frac{\frac{1}{2}\,(b_i + b_o)h}{\left(\frac{b_i r_o - b_o r_i}{h}\right)\ln\left(\frac{r_o}{r_i}\right) - (b_i - b_o)}$$

$$r_n = \frac{5500}{\left(\frac{80 \times 150 - 30 \times 50}{100}\right)\ln\left(\frac{150}{50}\right) - (80 - 30)}$$

$$\boldsymbol{r_n = 84.12\ mm}$$

Therefore

$$h_i = r_n - r_i = 84.12 - 50 = 34.12\ mm$$

$$M_b = P \times R = P \times 92.42 \quad Nmm$$

Eccentricity, e

$$e = R - r_n = 92.42 - 84.12$$

$$e = 8.3\ mm$$

Bending stress,

$$\sigma_{bi} = \frac{P \times 92.42 \times 34.12}{5500 \times 8.3 \times 50}$$

$$\sigma_{bi} = 1.382 \times 10^{-3} \times P \quad N/mm^2$$

This permissible stress must be equal to the sum of the stresses acting on the bracket, hence

$$\sigma = \sigma_t + \sigma_{bi}$$

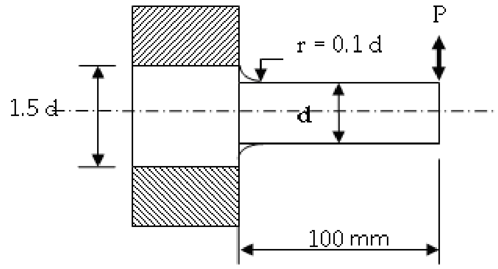
$$120 = (1.82 \times 10^{-4} \times P) + (1.382 \times 10^{-3} \times P)$$

On solving and simplifying the above equation,

$$P = 76.7\ kN$$

VARIABLE STRESSES

1. A cantilever beam made of steel Fe 540 ($\sigma_u = 540\ N/mm^2$ and $\sigma_y = 320\ N/mm^2$) is subjected to a completely reversed load (P) of 5 kN is shown in Fig. The shaft is machined (K_a = 0.8) and the reliability is 50% (K_c = 1). The factor of safety is 2 and the notch sensitivity factor is 0.9. Calculate (i) Endurance strength at the fillet section and (ii) Diameter (d) for infinite life.



Given Data:

Ultimate strength, $\sigma_u = 540$ N/mm^2

Yield strength, $\sigma_y = 320$ N/mm^2

Reversed load, $P_{min} = -5$ kN & $P_{max} = 5$ kN

Surface finish factor $K_a = 0.8$

Reliability factor, $K_c = 1$

Factor of safety n = 2

Notch sensitivity factor q = 0.9

To Find:

(i) Endurance limit, σ_{-1} =? (ii) Diameter of the shaft, d =?

Solution:

The applied external reversed load produces bending stress, therefore the maximum and minimum bending stresses are determined as follows

Maximum bending stress,

$$\sigma_{b,max} = \frac{M_{b,max}\ y}{I} = \frac{P_{max} \times 100 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4} = \frac{5 \times 10^3 \times 100 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4}$$

$$\boldsymbol{\sigma_{b,max} = \frac{5.09 \times 10^6}{d^3}\ N/mm^2}$$

Similarly the minimum bending stress,

$$\sigma_{b,min} = \frac{M_{b,min}\ y}{I} = \frac{P_{min} \times 100 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4} = \frac{-5 \times 10^3 \times 100 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4}$$

$$\boldsymbol{\sigma_{b,min} = -\frac{5.09 \times 10^6}{d^3}\ N/mm^2}$$

To find the mean stress and amplitude stress,

$$mean\ stress, \sigma_m = \frac{\sigma_{b,max} + \sigma_{b,min}}{2}$$

$$\boldsymbol{\sigma_m = 0}$$

$$amplitude\ stress, \sigma_a = \frac{\sigma_{b,max} - \sigma_{b,min}}{2}$$

$$\boldsymbol{\sigma_a = \frac{5.09 \times 10^6}{d^3}\ N/mm^2}$$

To find the endurance limit, use the empirical relation provided in PSGDB pg.no 1.42

$$\sigma_{-1} = 0.46\ \sigma_u = 0.46 \times 540$$

$$\boldsymbol{\sigma_{-1} = 248.4\ N/mm^2}$$

By considering the endurance limit correction factors, the modified endurance limit is given as

$$\sigma'_{-1} = K_a \times K_c \times \sigma_{-1} = 0.8 \times 1 \times 248.4$$

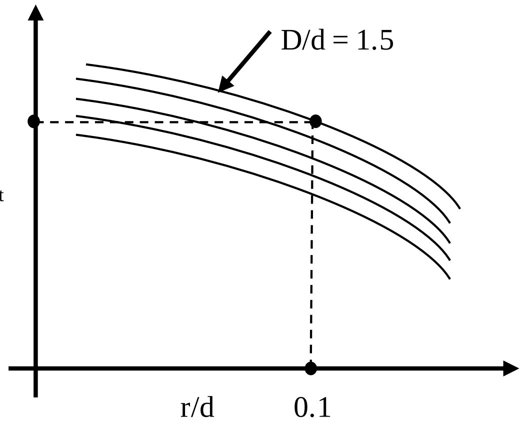
$$\boldsymbol{\sigma'_{-1} = 198.72\ N/mm^2}$$

To find the fatigue stress concentration factor, k_f

From PSGDB pg.no 7.6

$$k_f = 1 + q\ (k_t - 1)$$

To find the theoretical stress concentration
From PSGDB pg.no 7.14 from the graph



$$\frac{r}{d} = \frac{0.1\,d}{d} = 0.1$$
$$\frac{D}{d} = \frac{1.5\,d}{d} = 1.5$$

For the above value, from the graph the stress concentration factor value is selected or plotted as

$$\boldsymbol{k_t = 1.5}$$

Therefore

$$k_f = 1 + 0.9\,(1.5 - 1)$$
$$\boldsymbol{k_f = 1.45}$$

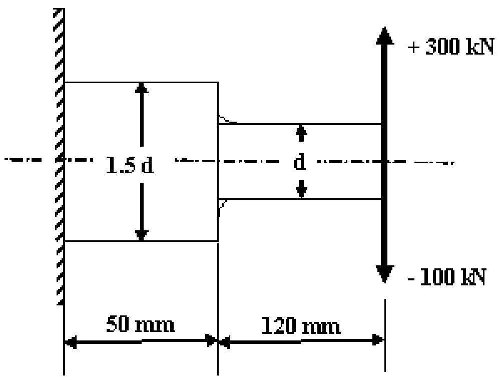
From PSGDB pg.no 7.6 according to the Soderberg equation

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + k_f \frac{\sigma_a}{\sigma'_{-1}}$$
$$\frac{1}{2} = \frac{0}{320} + 1.45 \times \frac{\frac{5.09 \times 10^6}{d^3}}{198.72}$$

On solving the above equation

$$\boldsymbol{d = 42.03\ mm}$$

2. A cantilever rod of circular section is subjected to a cyclic transverse load; varying from -100 kN to + 300 kN as shown in Fig. Determine the diameter d of the rod by (i) Goodman method and (ii) Soderberg method using the following data. Factor of safety = 2.,Theoretical stress concentration factor, K_t = 1.4, Notch sensitivity factor, q = 0.9, Ultimate strength, σ_u = 540 MPa, Yield strength, σ_y = 320 MPa, Endurance limit, σ_{-1} = 275 MPa, Size correction factor, K_b = 0.85



Given Data:
Reversed load, P_{min} = - 100 kN & P_{max} = + 300 kN
Factor of safety, n = 2
Theoretical stress concentration factor, K_t = 1.4
Notch sensitivity factor, q = 0.9
Ultimate strength, σ_u = 540 MPa = 540 N/mm^2

Yield strength, σ_y = 320 MPa = 320 N/mm^2
Endurance limit, σ_{-1} = 275 MPa = 275 N/mm^2
Size correction factor, K_b = 0.85
To Find:
Diameter of the shaft, d =?
Solution:
The applied external reversed load produces bending stress, therefore the maximum and minimum bending stresses are determined as follows
Maximum bending stress,

$$\sigma_{b,max} = \frac{M_{b,max}\,y}{I} = \frac{P_{max} \times 100 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4} = \frac{300 \times 10^3 \times 120 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4}$$

$$\boldsymbol{\sigma_{b,max} = \frac{366.69 \times 10^6}{d^3}\ N/mm^2}$$

Similarly the minimum bending stress,

$$\sigma_{b,min} = \frac{M_{b,min}\,y}{I} = \frac{P_{min} \times 100 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4} = \frac{-100 \times 10^3 \times 120 \times \frac{d}{2}}{\frac{\pi}{64} \times d^4}$$

$$\boldsymbol{\sigma_{b,min} = -\frac{122.23 \times 10^6}{d^3}\ N/mm^2}$$

To find the mean stress and amplitude stress,

$$mean\ stress, \sigma_m = \frac{\sigma_{b,max} + \sigma_{b,min}}{2}$$

$$\sigma_m = \frac{\frac{366.69 \times 10^6}{d^3} - \frac{122.23 \times 10^6}{d^3}}{2}$$

$$\boldsymbol{\sigma_m = \frac{122.23 \times 10^6}{d^3}\ N/mm^2}$$

$$amplitude\ stress, \sigma_a = \frac{\sigma_{b,max} - \sigma_{b,min}}{2}$$

$$\sigma_a = \frac{\frac{366.69 \times 10^6}{d^3} - \left(-\frac{122.23 \times 10^6}{d^3}\right)}{2}$$

$$\boldsymbol{\sigma_a = \frac{244.46 \times 10^6}{d^3}\ N/mm^2}$$

By considering the endurance limit correction factors, the modified endurance limit is given as

$$\sigma'_{-1} = K_b \times \sigma_{-1} = 0.85 \times 540$$

$$\boldsymbol{\sigma'_{-1} = 459\ N/mm^2}$$

To find the fatigue stress concentration factor, k_f
From PSGDB pg.no 7.6

$$k_f = 1 + q\,(k_t - 1)$$

$$k_f = 1 + 0.9\,(1.4 - 1)$$

$$\boldsymbol{k_f = 1.36}$$

(i) From PSGDB pg.no 7.6 according to the Goodman equation

$$\frac{1}{n} = k_t\left(\frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma'_{-1}}\right)$$

$$\frac{1}{2} = 1.4 \times \left(\frac{\frac{122.23 \times 10^6}{d^3}}{320} + \frac{\frac{244.46 \times 10^6}{d^3}}{459} \right)$$

$$\frac{1}{2} = \frac{1.280 \times 10^6}{d^3}$$

On solving the above equation

$$\boldsymbol{d = 136.8\ mm}$$

(ii) From PSGDB pg.no 7.6 according to the Soderberg equation

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + k_f \frac{\sigma_a}{\sigma'_{-1}}$$

$$\frac{1}{2} = \frac{\frac{122.23 \times 10^6}{d^3}}{320} + 1.36 \times \frac{\frac{244.46 \times 10^6}{d^3}}{459}$$

$$\frac{1}{2} = \frac{1.1063 \times 10^6}{d^3}$$

On solving the above equation

$$\boldsymbol{d = 130.3\ mm}$$

3. A simply supported beam has a concentrated load at the centre which fluctuates from a value of P to 4P. The span of the beam is 500 mm and its cross-section is circular with a diameter of 60 mm. Taking for the beam material an ultimate stress of 700 MPa, a yield stress of 500 MPa, endurance limit of 330 MPa for reversed bending and a factor of safety of 1.3, calculate the maximum value of P. Take a size factor of 0.85 and a surface finish factor of 0.9.

Given Data:

Concentrated load, P_{min} = P & P_{max} = 4 P

Span of the SSB, L = 500 mm

Cross section –circle, d = 60 mm

Ultimate stress, σ_u = 700 MPa = 700 N/mm^2

Yield stress σ_y = 500 MPa = 500 N/mm^2

Endurance limit σ_{-1} = 330 MPa = 330 N/mm^2

Factor of safety n = 1.3

Size factor K_b = 0.85

Surface finish factor K_a = 0.9

To Find:

Load, P =?

Solution:

The given simply supported beam is subjected to bending stress due to bending moment produced by the applied external concentrated load.

The bending moment can be found using the equation taken from PSGDB pg.no 6.4

$$M_b = \frac{PL}{4}$$

Maximum bending moment,

$$M_{b,max} = \frac{4P \times 500}{4} = 500P\ N.mm$$

Minimum bending moment

$$M_{b,min} = \frac{P \times 500}{4} = 125P\ N.mm$$

The distance of the centroidal axis of the given circular cross section,

$$y = \frac{d}{2} = \frac{60}{2} = 30\ mm$$

Area moment of inertia of the given cross section is

$$I = \frac{\pi}{64} \times d^4 = \frac{\pi}{64} \times 60^4 = 636.17 \times 10^3\ mm^4$$

Maximum bending stress,

$$\sigma_{b,max} = \frac{M_{b,max}\ y}{I} = \frac{500\ P \times 30}{636.17 \times 10^3}$$

$$\boldsymbol{\sigma_{b,max} = 0.024\ P\ N/mm^2}$$

Similarly the minimum bending stress,

$$\sigma_{b,min} = \frac{M_{b,min}\ y}{I} = \frac{125\ P \times 30}{636.17 \times 10^3}$$

$$\boldsymbol{\sigma_{b,min} = 0.006\ P\ N/mm^2}$$

To find the mean stress and amplitude stress,

$$mean\ stress, \sigma_m = \frac{\sigma_{b,max} + \sigma_{b,min}}{2}$$

$$\sigma_m = \frac{0.024P + 0.006P}{2}$$

$$\boldsymbol{\sigma_m = 0.015\ P\ N/mm^2}$$

$$amplitude\ stress, \sigma_a = \frac{\sigma_{b,max} - \sigma_{b,min}}{2}$$

$$\sigma_a = \frac{0.024P - 0.006P}{2}$$

$$\boldsymbol{\sigma_a = 0.009\ P\ N/mm^2}$$

By considering the endurance limit correction factors, the modified endurance limit is given as

$$\sigma'_{-1} = K_a \times K_b \times \sigma_{-1} = 0.9 \times 0.85 \times 700$$

$$\boldsymbol{\sigma'_{-1} = 535.5\ N/mm^2}$$

From PSGDB pg.no 7.4 according to the Soderberg equation

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma'_{-1}}$$

$$\frac{1}{1.3} = \frac{0.015\ P}{500} + \frac{0.009P}{535.5}$$

On solving the above equation

$$\boldsymbol{P = 16.43\ kN}$$

4. A spherical pressure vessel with a 500 mm inner diameter is welded from steel plates. The welded joints are sufficiently strong and do not weaken the vessel. The plates are made from cold drawn steel 20C8 (σ_u = 440 N/mm^2 and σ_y = 242 N/mm^2). The vessel is subjected to internal pressure which varies from 0 to 5 N/mm^2. The expected reliability is 50% and the factor of safety is 3.5. The vessel is expected to withstand infinite number of stress cycles. Calculate the thickness of the plates.

Given Data:

Inner diameter, d = 500 mm

Ultimate stress, σ_u = 440 N/mm^2

Yield stress, σ_y = 242 N/mm^2

Internal pressures, p_{min} = 0 and p_{max} = 5 N/mm^2

Reliability 50%, reliability factor K_c = 1

Fcator of safety, n = 3.5

To find:

Thickness of the steel plate, t =?

Solution:

The spherical pressure vessel is subjected to circumferential stress and it is given as

$$\sigma = \frac{p\,d}{4\,t}, \qquad N/mm^2$$

Therefore, the given spherical pressure vessel is subjected to maximum and minimum stresses due to p_{min} and p_{max} pressures applied and it is given as

$$\sigma_{min} = \frac{p_{min}\,d}{4\,t}$$

$$\sigma_{min} = 0 \ N/mm^2$$

$$\sigma_{max} = \frac{p_{max}\,d}{4\,t} = \frac{(5 \times 500)}{4 \times t}$$

$$\sigma_{max} = \frac{625}{t} \ N/mm^2$$

To find the mean stress and amplitude stress,

$$mean\ stress, \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\sigma_m = \frac{\frac{625}{t} + 0}{2}$$

$$\boldsymbol{\sigma_m = \frac{312.5}{t} \ N/mm^2}$$

$$amplitude\ stress, \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\sigma_a = \frac{\frac{625}{t} - 0}{2}$$

$$\boldsymbol{\sigma_a = \frac{312.5}{t} \ N/mm^2}$$

By considering the endurance limit correction factors, the modified endurance limit is given as

$$\sigma'_{-1} = K_c \times \sigma_{-1} = 1 \times 440$$

$$\boldsymbol{\sigma'_{-1} = 440 \ N/mm^2}$$

From PSGDB pg.no 7.4 according to the Soderberg equation

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma'_{-1}}$$

$$\frac{1}{3.5} = \frac{\frac{312.5}{t}}{242} + \frac{\frac{312.5}{t}}{440}$$

On solving the above equation

$$\boldsymbol{t = 7\ mm}$$

MULTIPLE CHOICE QUESTIONS

S.No	Question	Opt 1	Opt 2	Opt 3	Opt 4	Answers
1	Hooke's law holds good upto	yield point	elastic point	plastic point	breaking point	elastic point
2	The ratio of linear stress to linear strain is called	Modulus of elasticity	Modulus of rigidity	Bulk modulus	Poisson's ratio	Modulus of elasticity
3	The Modulus of elasticity for mild steel is approximately equal to	80KN/mm²	100KN/mm²	110KN/mm²	210KN/mm²	210KN/mm²
4	When the material is loaded within elastic limit ,then the stress is……..to strain	equal	directly proportional	inversly proportional	none of these	directly proportional
5	The ratio of the ultimate stress to the design stress is known as	elastic limit	strain	factor of safety	bulk modulus	factor of safety
6	The factor of safety for steel and for steady load is	2	4	6	8	4
7	Steels are designated by group of letters or numbers indicating--------- properties	Tensile strength	carbon content	composition of alloying elements	any one of these	any one of these
8	55C4 indicates a plain carbon steel with	55 % C & 4% Mn	5.5%C & 4% Mn	0.55%C & 4% Mn	0.55% C & 0.4 % Mn	0.55% C & 0.4 % Mn
9	The IS code used for designation of steels	IS 1762-1974	IS 1570 - 1978	IS 7162-1974	IS 1762-1984	IS 1762-1974
10	Blackheart malleable cast iron is used for making	brake shoes	pipe fittings	motor cycle frames	fittings for bicycle	brake shoes
11	Whiteheart malleable cast iron is used for making	pedal	levers	pipe fittings	wheel hub	pipe fittings
12	__________ is popuarly called as Machinery steel	low carbon steel	high carbon steel	medium carbon steel	alloy steel	medium carbon steel
13	16Mn5Cr4 is used for making	Gears & shaft	axle	cam	camshaft	Gears & shaft
14	An aluminium member is designed based on	yield stress	elastic limit stress	proof stress	ultimate stress	yield stress
15	In a body , a thermal stress is one which arises because of the existence of	latent heat	temperature gradient	total heat	specific heat	temperature gradient
16	A localised compressive stress at the area of contact between two members is known as	tensile stress	bending stress	bearing stress	shear stress	bearing stress
17	The poisson's ratio for steel varies from	0.21 to 0.25	0.25 to 0.33	0.33 to 0.38	0.38 to0.45	0.25 to 0.33
18	For moderate soze steel forgings, the minimum corner radii for a depth of 50 mm is ----------	1.5 mm	2 mm	2.5 mm	3.5 mm	3.5 mm
19	for steel forgings, the recommended value of the minimum section thickness is --------	5 mm	2 mm	3 mm	4 mm	3 mm
20	The stress in the bar when load is applied suddenly is ………..as compared to the stress induced due to graually applied load	same	double	three times	four times	double
21	The energy stored in a body when strained within elastic limit is known as	resilience	proof resilience	strain energy	impact energy	strain energy
22	The maximum energy that can be stored in a body due to external	resilience	proof resilience	strain energy	modulus of resilience	proof resilience

	loading upto the elastic limit is called					
23	The strain energy stored in body, when suddenly loaded,is……the strain energy stored when same load is applied gradually.	equal to	one-half	twice	four times	four times
24	When a machine member is subjected to torsion, the torsional shear stress set up in the member is	zero at both the centroidal axis and outer surface of the member	maximum at both the centroidal axis and outer surface of the member	zero at the centroidal axis and maximum at the outer surface of the member	none of the these	zero at the centroidal axis and maximum at the outer surface of the member
25	The torsional shear stress on any cross -section normal to the axis is …. the distance from the centre of the axis	directly proportional to	inversely proportional to	zero	none of above	directly proportional to
26	The nuetral axis of a beam is subjected to	zero stress	maximum tensile stress	maximum compressive stress	maximum shear stress	zero stress
27	At the neutral axis of a beam	the layers are subjected to maximum bending stress	the layers are subjected to tension	the layers are subjected to compression	the layers do not undergo any strain	the layers do not undergo any strain
28	The bending stress in a curved beam is	zero at the centroidal axis	zero at any point other than centroidal axis	maximum at the neutral axis	none of these	zero at any point other than centroidal axis
29	The maximum bending stress ,in a curved beam having symmetrical section, always occur, at the	centroidal axis	neutral axis	inside fibre	outside fibre	inside fibre
30	Two shafts under pure torsion are of identical length and identical weight and are made of same material. The shafts A is solid and the shafts B is hollow.We can say that	Shaft B is better than shaft A	Shaft A is better than shaft B	both the shafts are equally good	none of above	Shaft B is better than shaft A
31	Rankine's theory is used for	brittle materials	ductile materials	elastic materials	plastic materials	brittle materials
32	Guest's theory is used for	brittle materials	ductile materials	elastic materials	plastic materials	ductile materials
33	At the neutral axis of a beam, the shear stress is	zero	maximum	minimum	none of above	maximum
34	The maximum shear stress developed in a beam of rectangular section is…………... the average shear stress	equal to	4/3 times	1.5 times	none of above	1.5 times
35	The stress which vary from a minimum value to a maximum value of the same nature(i.e. tensile or compressive) is called	repeated stress	yield stress	fluctuating stress	alternating stress	fluctuating stress
36	The endurance or fatique limit is defined as the maximum value of the stress which a polished standard speciman can withstand	Without Failure for infinite no of cycles	without failure for finite no of cycles	without failure for 103 cycles	None of these	Without Failure for infinite no of cycles
37	without failure, for infinite number of cycles ,when subjected to	static load	dynamic load	static as well as dynamic load	completely reversed load	completely reversed load

38	Failure of a material is called fatique when it fails	at the elasticlimit	below the elasticlimit	at the yield point	below the yield point	below the yield point
39	The resistance to fatique of a material is measured by	elastic limit	Young's modulus	ultimate tensile strength	endurance limit	endurance limit
40	The yield point in static loading is……..as compared to fatique loading	higher	lower	same	none of above	higher
41	Factor of safety for fatique loading is the ratio of	elastic limit to the working stress	Young's modulus to the ultimate tensile strength	endurance limit to the working stress	elastic limit to the yield point	endurance limit to the working stress
42	When a material is subjected to fatique loading , the ratio of the endurance limit to the ultimate tensile strength is	0.2	0.35	0.5	0.65	0.5
43	The ratio of endurance limit in shear to the endurance limit in flexure is	0.25	0.4	0.55	0.7	0.55
44	If the size of the standard specimen for a fatique testing machine is increased , the endurance limit for the material will	have same value as that of standard speciman	increase	decrease	none of above	decrease
45	The residential compressive stress by way of surface treatment of a machine member subjected to fatique loading	improves the fatique life	deteriorates the fatique life	does not affect the fatique life	immediately fractures the specimen	improves the fatique life
46	The surface finish factor for a mirror polished material is	0.45	0.65	0.85	1	1
47	Stress concentration factor is defined as the ratio of	maximum stress to the endurance limit	nominal stress to the endurance limit	maximum stress to the nominal limit	nominal stress to the maximum limit	maximum stress to the nominal limit
48	In static loading ,stress concentration is more serious in	brittle materials	ductile materials	brittle as well as ductile materials	elastic materials	brittle materials
49	In cycle loading, stress concentration is more serious in	brittle materials	ductile materials	brittle as well as ductile materials	elastic materials	ductile materials
50	The notch sensitivity q is expressed in terms of fatique stress concentration factor Kf and theoretical stress concentration factor Kt,as	(kf+1)/(kt-1)	(kf-1)/(kt-1)	(kt-1)/(kf-1)	(kf-1)*(kt-1)	(kf-1)/(kt-1)

UNIT II

DESIGN OF SHAFTS AND COUPLINGS

Design of solid and hollow shafts based on strength, rigidity and critical speed – Design of keys and key ways – Design of rigid and flexible couplings – Introduction to gear and shock absorbing couplings – design of knuckle joints.

1. DESIGN OF SHAFTS

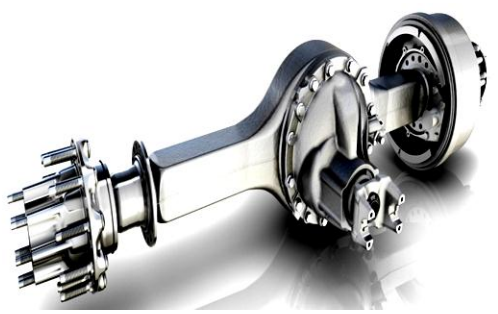
1.1. TRANSMISSION SHAFTS

It is a rotating machine element usually circular in cross section, which is used to support power, torque and motion transmitting elements like gears, pulleys, sprockets etc.

1.2. CATEGORIES OF TRANSMISSION SHAFTS

a) Axle:

It is a type of shaft which is used to support rotating elements like wheels, hoisting drums, rope sheaves, which are fitted to the housing by means of bearings. Such shafts are subjected to bending moment due to transverse reactions of the rotating elements.



Axle

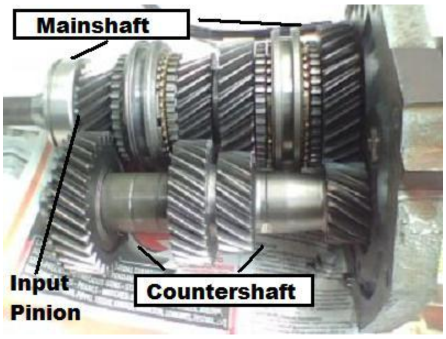
Spindle

b) Spindle

It is a short shaft which is commonly used in machine tools such as small drive shaft of a lathe or a spindle in drilling machines.

c) Counter shaft:

It is a secondary shaft for which the power is transmitted from the main shaft and thereby it transmits power to the machine components. These are used in multistage gear boxes.



d) Jack shaft:

It is an intermediate shaft between two shafts which are used in transmission of power.

e) Line shaft:

A line shaft consists of number of shafts which are connected in axial directions by means of couplings. It is popular in workshops using group drive.

1.3. MATERIALS USED FOR SHAFTS:

- Ordinary transmission shafts – medium carbon steels (30C8 or 40C8) (Machinery steels).
- For greater strength – high carbon steels like 45C8 or 50C8 or alloy steels can be used.
- Alloy steels – (16Mn5Cr4, 40Cr4Mo2, 16Ni3Cr2 etc.,)
- These alloy steels are costlier when compared to plain carbon steels but these are used when higher strength, higher toughness & hardness is required. It also provides higher resistance to corrosion.
- The material selected for shafts should have good strength, good machinability characteristics, low notch sensitivity factor, good heat treatment properties and higher wear resistance.

1.3.1. MANUFACTURING OF SHAFTS:

Commercial shafts are made of low carbon steels, which are manufactured by cold drawing or hot rolling or by turning & grinding operations. Cold drawn shafts are stronger than hot rolled shafts but they do not meet the required tolerance & straightness, it also produces residual stresses on the near surfaces of the shafts. When machining is performed on such shafts, it leads to the release of residual stress causing the shaft to get distorted. And it becomes an expensive process to straighten such distorted shafts. Hence most of the shafts are hot rolled and machined & ground, also hardened by oil quenching process.

1.4. DESIGN OF SHAFTS

The shafts are designed based on the following categories

a) Based on strength
b) Based on torsional rigidity and stiffness
c) Based on critical speed

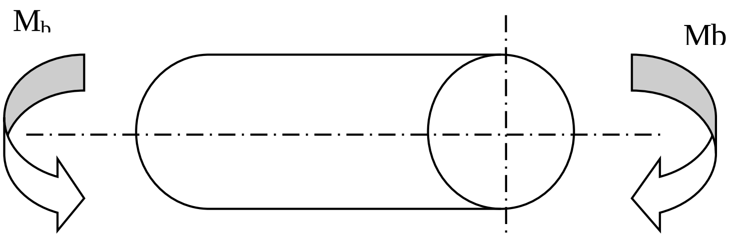
1.4.1. DESIGN OF SHAFT BASED ON STRENGTH

The design of shaft based on strength is classified into different types depending on the type of load the shaft is subjected to

(i) Transverse load (Bending moment)
(ii) Torsional load or Torque (Twisting moment)
(iii) Combined transverse and torsional loads (Bending & twisting moment)
(iv) Combined axial, transverse and torsional loads (axial, Bending & twisting moment)
(v) Variable loading or Fluctuating loads

(i) Shaft subjected to Transverse load (Bending moment)

Consider a shaft of circular cross section of diameter ‘d’ is subjected to bending moment M_b. Due to this loading the shaft material develops internal resistance for the applied bending moment leads to formation of Bending stresses, σ_b



The bending stress, σ_b is given by simple bending equation,

$$\sigma_b = \frac{M_b\, y}{I} \ldots\ldots\ldots\ldots\ldots (From\ PSGDB\ pg.no\ 7.1)$$

M_b – Bending moment, N-mm

y - Distance of the inner or outer fiber from the neutral axis, mm

I – Area moment of inertia, mm^4

For a circular cross section,

$$y = \frac{d}{2}\ , mm$$

$$\&\ \ I = \frac{\pi\, d^4}{64}\ , mm^4 \ldots\ldots\ldots\ldots\ldots (From\ PSGDB\ pg.no\ 6.1)$$

On substituting the above values in the bending stress equation we get,

$$\sigma_b = \frac{M_b\, \frac{d}{2}}{\frac{\pi\, d^4}{64}}$$

$$\boldsymbol{\sigma_b = \frac{32\, M_b}{\pi\, d^3}\ , N/mm^2}$$

Similarly for a hollow circular cross section shaft, having

D – Outer diameter, mm

d – Inner diameter, mm

$$y = \frac{D}{2}\ , mm$$

$$\&\ \ I = \frac{\pi\,(D^4 - d^4)}{64}\ , mm^4 \ldots\ldots\ldots\ldots\ldots (From\ PSGDB\ pg.no\ 6.1)$$

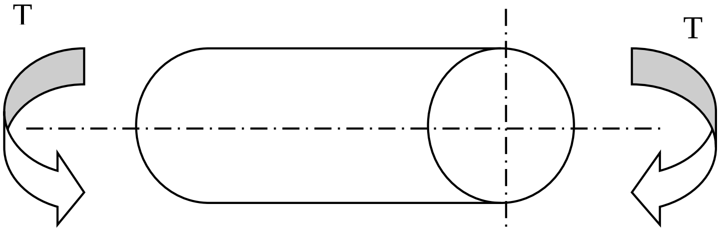
On substituting the above values in the bending stress equation we get,

$$\sigma_b = \frac{M_b\,\frac{D}{2}}{\frac{\pi\,(D^4 - d^4)}{64}}$$

$$\boldsymbol{\sigma_b = \frac{32\, M_b\, D}{\pi\,(D^4 - d^4)}\ , N/mm^2}$$

(ii) Shaft subjected to Torsional load or Torque (Twisting moment)

Consider a shaft of circular cross section of diameter 'd' is subjected to Twisting moment T. Due to this loading the shaft material develops internal resistance for the applied twisting moment that leads to formation of torsional shear stress, τ.



And it is given by the simple torsion equation,

$$\tau = \frac{T\,r}{J},\qquad N/mm^2 \ldots\ldots\ldots\ldots (From\ PSGDB\ pg.no\ 7.1)$$

T – Torsional load (Torque or Twisting moment), N-mm

r - Radius of the shaft, mm

J – Polar moment of inertia, mm^4

For a circular cross section,

$$r = \frac{d}{2}\ , mm$$

$$\&\ \ J = \frac{\pi\, d^4}{32}\ , mm^4$$

On substituting the above values in the bending stress equation we get,

$$\tau = \frac{T\,\frac{d}{2}}{\frac{\pi\, d^4}{32}}$$

$$\boldsymbol{\tau = \frac{16\, T}{\pi\, d^3}\ , N/mm^2}$$

Similarly for a hollow circular cross section shaft, having

D – Outer diameter, mm

d – Inner diameter, mm

$$r = \frac{D}{2}, mm$$

$$\& \; J = \frac{\pi\,(D^4 - d^4)}{32}, mm^4$$

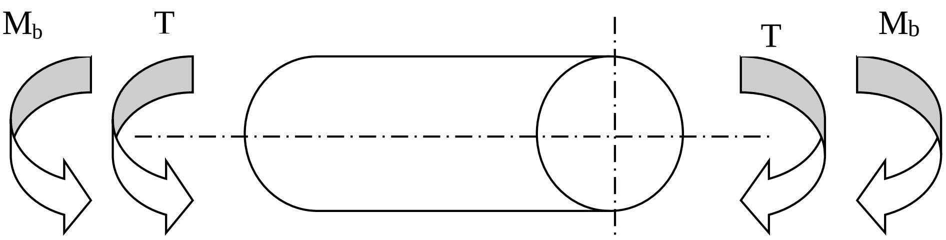
On substituting the above values in the bending stress equation we get,

$$\tau = \frac{T\,\frac{D}{2}}{\frac{\pi\,(D^4 - d^4)}{32}}$$

$$\boldsymbol{\tau = \frac{16\,T\,D}{\pi\,(D^4 - d^4)}, N/mm^2}$$

(iii) Design of Shaft subjected to Combined transverse and torsional loads (Bending & twisting moment)

When a shaft of circular cross section of diameter 'd' is subjected to combined bending and torsional load, it involves the formation of bending stresses, σ_b and torsional stresses, τ.



For shaft subjected to such combined static loads, failure theories are used for designing such shafts. Mostly maximum normal stress or principal stress theory and maximum principal shear stress theory are used out of different theories of failure.

(a) According to maximum normal stress or principal stress theory

The principal stresses for a biaxial stress condition is given as

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x + \sigma_y)^2 + 4\,\tau_{xy}^2}\right] \ldots. PSGDB\; pg.no\; 7.2$$

Here $\sigma_x = \sigma_b, \sigma_y = 0$ and $\tau_{xy} = \tau$

$$\sigma_{1,2} = \frac{1}{2}\left[\sigma_b \pm \sqrt{\sigma_b^2 + 4\,\tau^2}\right]$$

And the theory states that,

$$\sigma_1 \; or \; \sigma_2 = \frac{\sigma_y}{fos}$$

Where,

$$\boldsymbol{\sigma_b = \frac{32\,M_b}{\pi\,d^3}} \; and \; \boldsymbol{\tau = \frac{16\,T}{\pi\,d^3}, N/mm^2}$$

On substitution

$$\sigma_1 = \frac{\sigma_y}{fos} = \frac{1}{2}\left[\frac{32\,M_b}{\pi\,d^3} + \sqrt{\left(\frac{32\,M_b}{\pi\,d^3}\right)^2 + 4\left(\frac{16\,T}{\pi\,d^3}\right)^2}\right]$$

On simplification, we get

$$Equivalent\ bending\ moment, M_e = \left[M_b + \sqrt{{M_b}^2 + T^2}\right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

Similarly for Hollow circular shaft,

$$Equivalent\ bending\ moment, M_e = \left[M_b + \sqrt{{M_b}^2 + T^2}\right] = \frac{\pi}{16} \times \sigma_1 \times \frac{(D^4 - d^4)}{D}$$

Equivalent bending moment is the bending moment which when acting alone will produce the same effect as caused by the combined bending and torsional load.

(b) According to maximum principal shear stress theory

The principal stresses for a biaxial stress condition is given as

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_x + \sigma_y)^2 + 4\,{\tau_{xy}}^2} \ldots\ldots PSGDB\ pg.no\ 7.2$$

Here $\sigma_x = \sigma_b$, $\sigma_y = 0$ and $\tau_{xy} = \tau$

$$\tau_{max} = \frac{1}{2}\sqrt{{\sigma_b}^2 + 4\,\tau^2}$$

And the theory states that,

$$\tau_{max} = \frac{0.5\,\sigma_y}{fos}$$

Where,

$$\boldsymbol{\sigma_b = \frac{32\,M_b}{\pi\,d^3}}\ \ and\ \boldsymbol{\tau = \frac{16\,T}{\pi\,d^3}}\ \boldsymbol{, N/mm^2}$$

On substitution

$$\tau_{max} = \frac{0.5\,\sigma_y}{fos} = \frac{\tau_y}{fos} = \frac{1}{2}\sqrt{\left(\frac{32\,M_b}{\pi\,d^3}\right)^2 + 4\left(\frac{16\,T}{\pi\,d^3}\right)^2}$$

On simplification, we get

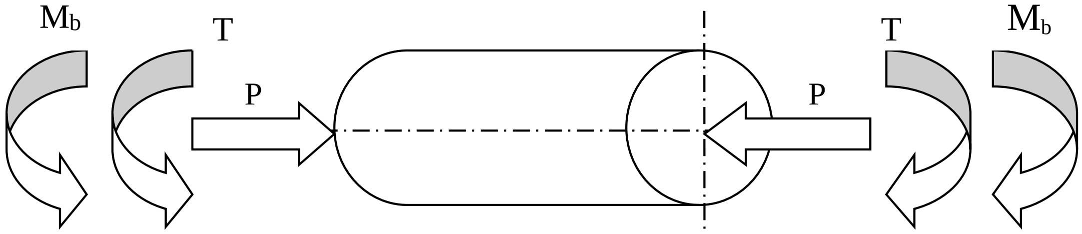
$$Equivalent\ Twisting\ moment, \quad T_e = \sqrt{{M_b}^2 + T^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

Similarly for Hollow circular shaft,

$$Equivalent\ Twisting\ moment, T_e = \sqrt{{M_b}^2 + T^2} = \frac{\pi}{16} \times \tau_{max} \times \frac{(D^4 - d^4)}{D}$$

(iv) Design of Shaft subjected to Combined axial, transverse and torsional loads (axial, Bending & twisting moment)

Consider a circular shaft of diameter (d) is subjected to combined axial, bending and torsional loads. Such a shaft will create axial (compressive or tensile) & bending stresses which are acting normal to the plane of the cross section of the shaft and torsional shear stress.



W.K.T

$$Bending\ stress, \qquad \sigma_b = \frac{32\ M_b}{\pi\ d^3}\ , N/mm^2$$

$$axial\ stress, \qquad \sigma_t = \frac{4\ P}{\pi\ d^2}\ , N/mm^2$$

Since both these stresses are acting normal to the plane, these two can be combined together to get the resultant stress, σ

$$\sigma = \sigma_t + \sigma_b = \frac{32\ M_b}{\pi\ d^3} + \frac{4\ P}{\pi\ d^2}$$

$$\sigma = \frac{32}{\pi\ d^3}\left[M_b + \frac{P\ d}{8}\right]$$

When the shaft is long and slender, it acts like a column and hence it tries to buckle, so in order to consider this effect, column factor (α) is to be included in the above equation,

$$\sigma = \frac{32}{\pi\ d^3}\left[M_b + \frac{\alpha\ P\ d}{8}\right]$$

Similarly for hollow shaft,

$$\sigma = \frac{32}{\pi\ d^3}\left[M_b + \frac{\alpha\ P\ d}{8}\right]$$

From PSGDB pg.no. 7.21

$$Column\ factor, \qquad \alpha = \frac{1}{1\ - 0.0044\left(\frac{l}{r}\right)} \qquad for\ \ \frac{l}{r} < 115$$

$$Column\ factor, \qquad \alpha = \frac{\sigma_y}{\pi^2\ n\ E}\left(\frac{l}{r}\right)^2 \qquad for\ \ \frac{l}{r} > 115$$

Where,

l - Length of the shaft under axial load, N

E - Young's modulus, N/mm^2

r - Radius of gyration, mm (see PSGDB pg.no 6.1)

n - End condition coefficient, (See PSGDB pg.no 6.8)

σ_y - Yield stress, N/mm^2

For such loading condition,

Equivalent Bending moment,

a) For solid shaft of diameter, d

$$M_e = \left[\left[M_b + \frac{\alpha\, P\, d}{8}\right] + \sqrt{\left[M_b + \frac{\alpha\, P\, d}{8}\right]^2 + T^2}\right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

b) For hollow shaft of outer diameter, D and inner diameter, d

$$M_e = \left[\left[M_b + \frac{\alpha\, P\,(D^2 + d^2)}{8\, D}\right] + \sqrt{\left[M_b + \frac{\alpha\, P\,(D^2 + d^2)}{8\, D}\right]^2 + T^2}\right] = \frac{\pi}{16} \times \sigma_1 \times \frac{(D^4 - d^4)}{D}$$

Equivalent twisting moment,

a) For solid shaft of diameter, d

$$T_e = \sqrt{\left[M_b + \frac{\alpha\, P\, d}{8}\right]^2 + T^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

b) For hollow shaft of outer diameter, D and inner diameter, d

$$T_e = \sqrt{\left[M_b + \frac{\alpha\, P\,(D^2 + d^2)}{8\, D}\right]^2 + T^2} = \frac{\pi}{16} \times \tau_{max} \times \frac{(D^4 - d^4)}{D}$$

(v) Shaft subjected to Variable or Fluctuating load

Consider a solid circular shaft subjected to variable loading or shock loads, for such loading conditions two factors, K_b and K_t are used.

K_b – combined shock & fatigue factor for bending load

K_t – combined shock & fatigue factor for torsional loading

K_b and K_t – are taken from PSGDB pg.no. 7.21, for different loading conditions

a) For combined variable bending and torsional loading,

(i) Equivalent Bending moment,

$$M_e = \left[K_b M_b + \sqrt{(K_b M_b)^2 + (K_t T)^2}\right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

Similarly for Hollow circular shaft,

$$M_e = \left[K_b M_b + \sqrt{(K_b M_b)^2 + (K_t T)^2}\right] = \frac{\pi}{16} \times \sigma_1 \times \frac{(D^4 - d^4)}{D}$$

(ii) Equivalent twisting moment, T_e

$$T_e = \sqrt{(K_b M_b)^2 + (K_t T)^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

Similarly for Hollow circular shaft,

$$T_e = \sqrt{(K_b M_b)^2 + (K_t T)^2} = \frac{\pi}{16} \times \tau_{max} \times \frac{(D^4 - d^4)}{D}$$

b) For combined variable axial, bending & torsional loading,

(i) Equivalent Bending moment,

For solid shaft of diameter, d

$$M_e = \left[\left[K_b M_b + \frac{\alpha\, P\, d}{8} \right] + \sqrt{\left[K_b M_b + \frac{\alpha\, P\, d}{8} \right]^2 + (K_t T)^2} \right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

For hollow shaft of outer diameter, D and inner diameter, d

$$M_e = \left[\left[K_b M_b + \frac{\alpha\, P\, (D^2 + d^2)}{8\, D} \right] + \sqrt{\left[K_b M_b + \frac{\alpha\, P\, (D^2 + d^2)}{8\, D} \right]^2 + (K_t T)^2} \right] = \frac{\pi}{16} \times \sigma_1 \times \frac{(D^4 - d^4)}{D}$$

(ii) Equivalent Twisting moment, T_e

For solid shaft of diameter, d

$$T_e = \sqrt{\left[K_b M_b + \frac{\alpha\, P\, d}{8} \right]^2 + (K_t T)^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

For hollow shaft of outer diameter, D and inner diameter, d

$$T_e = \sqrt{\left[K_b M_b + \frac{\alpha\, P\, (D^2 + d^2)}{8\, D} \right]^2 + (K_t T)^2} = \frac{\pi}{16} \times \tau_{max} \times \frac{(D^4 - d^4)}{D}$$

1.4.2. DESIGN OF SHAFTS BASED ON ASME CODE:

According to ASME code, the permissible shear stress for the shaft without keyways is taken as 30% of the yield strength in tension and 18% of the ultimate strength of the material; the minimum of the two is taken for design calculation.

$$\tau = 0.3\, \sigma_y \quad \text{or} \quad \tau = 0.18\, \sigma_u$$

If key way is present, then 75% of the above value is taken.

1.4.3. DESIGN OF SHAFTS BASED ON TORSIONAL RIGIDITY:

A shaft can be treated as a rigid body based on its torsional rigidity, which is if the shaft does not twist too much on the application of external torque. Otherwise if the design is based on lateral rigidity, the shaft should not deflect too much under the application of the external transverse loads (Bending moment).

According to simple torsion equation,

$$\frac{T}{J} = \frac{G\ \theta}{l}$$

From which the angular twist or angle of twist θ is given as,

$$\theta = \frac{T\ l}{G\ J}$$

Where,

θ – Angular twist, degree

T – Torsional moment, N-mm

l – Length of the shaft, mm

G – Rigidity modulus, N/mm^2

J – Polar moment of inertia, mm^4

The permissible angular twist for machine tool application is 0.25° per meter length and for line shafts 3° per meter length of the shaft.

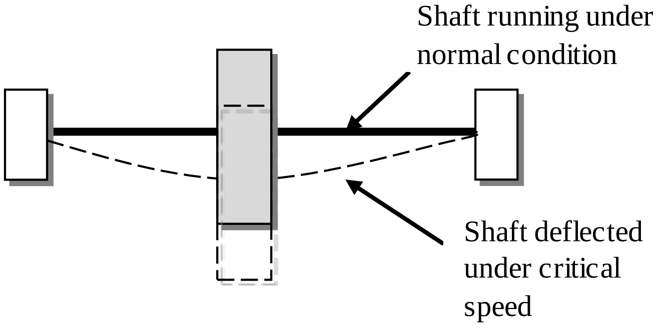
In case of lateral deflection, the permissible deflection is around 0.001 to 0.003l for machine shafts between two bearings, 0.01l m for shaft mounted with spur gears.

There are some possibilities to reduce the lateral deflection by,

i) Reducing the span length
ii) Increasing the number of supports
iii) Selecting the cross section in which the area moment of inertia is large as in case of tubular or hollow shafts.

1.4.4. DESIGN OF SHAFT BASED ON STIFFNESS & CRITICAL SPEED

The critical speed or whirling speed of the shaft is defined as the speed at which the additional deflection from the axis of rotation of the shaft becomes infinite. Simply it is the speed at which the shaft meets its natural frequency resulting in resonance condition.



$$Natural\ frequency, \omega_c = \sqrt{\frac{k}{m}}\ or \sqrt{\frac{g}{\delta}}\quad or\ f = \frac{1}{2\pi}\sqrt{\frac{k}{m}}\ or \sqrt{\frac{g}{\delta}}\ , Hertz$$

Where,

k - Stiffness of the shaft, N/mm

m - Mass of the shaft, kg

g - Acceleration due to gravity (9.81 m/sec)

δ - Deflection of the shaft, mm

Stiffness is defined as the ratio of load per unit deflection, k = W/δ, N/mm

For shaft carrying 'n' no of transmission elements like pulley or gear or flywheel or sprocket, the natural frequency for such systems can be found using Dunkerley's rule

$$\frac{1}{{\omega_c}^2} = \frac{1}{{\omega_1}^2} + \frac{1}{{\omega_2}^2} + \ldots\ldots\ldots + \frac{1}{{\omega_n}^2}$$

PROBLEMS:

i) Design of shaft subjected to torsional load (Twisting moment or torque)

1. A hollow shaft for a rotary compressor is to be designed to transmit a maximum torque of 3500 Nm. The shear stress is the shaft is limited to 50 MPa. Determine inside and outside diameters of the shaft, if the ratio of inside & outside diameters is 0.4.

Given Data:

T = 3500 Nm = 3500 x 10^3 N mm

τ = 50 MPa = 50 N/mm^2

d/D = 0.4

To find:

D= ? & d =?

Solution:

The torsional shear stress acting on this hollow shaft is given as

$$\tau = \frac{16\,T\,D}{\pi\,(D^4 - d^4)}\,,N/mm^2$$

Where d= 0.4 D

On substitution of the values

$$50 = \frac{16 \times 3500 \times 10^3 \times D}{\pi\,(D^4 - (0.4D)^4)}$$

$$50 = \frac{16 \times 3500 \times 10^3 \times D}{\pi\,D^4(1^4 - (0.4)^4)}$$

$$D^3 = \frac{16 \times 3500 \times 10^3}{\pi \times 50 \times (1^4 - (0.4)^4)} = 365873.43$$

$$D = \sqrt[3]{365873.43}$$

$$\boldsymbol{D = 71.5\,mm}$$

ii) Design of Shaft subjected to Combined transverse and torsional loads (Bending & twisting moment)

1. A mild steel shaft transmits 23 kW at 200 rpm. It carries a central load of 900 N and is simply supported between the bearings 2.5 meters apart. Determine the size of the shaft, if the allowable shear stress is 42 MPa and the

maximum tensile or compressive stress is not to exceed 56 MPa. What size of the shaft will be required, if it is subjected to gradually applied loads?

Given Data:

Power, P = 23 kW = 23 x 10^6 N-mm/s; Speed, N = 200 rpm; load, W = 900 N; Length, L = 2500 mm; τ = 42 N/mm^2; σ = 56 N/mm^2

To Find:

Diameter of the shaft, d =?

Solution:

The given mild steel shaft is subjected to torque, T and bending moment, M_b due to the central load.

To find the torque, T

$$P = \frac{2\,\pi\,N\,T}{60}\ watt$$

$$Torque\,,T = \frac{P \times 60}{2\,\pi\,N} = \frac{23 \times 10^6 \times 60}{2\,\pi \times 200}$$

$$T = 1.098 \times 10^6\,, \quad Nmm$$

To find the Bending moment, M_b

From PSGDB pg.no. 6.3, for SSB subjected to central load,

$$M_b = \frac{W \times L}{4} = \frac{900 \times 2500}{4}$$

$$M_b = 0.563 \times 10^6\,, \quad Nmm$$

To find the equivalent bending & torsional moment

Case(i) Steady loading

According to Equivalent bending moment, the diameter of the shaft

$$M_e = \left[M_b + \sqrt{(M_b)^2 + (T)^2}\right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

$$M_e = \left[M_b + \sqrt{(M_b)^2 + (T)^2}\right] = \left[0.563 \times 10^6 + \sqrt{(0.563 \times 10^6)^2 + (1.098 \times 10^6)^2}\right]$$

$$M_e = 1.796 \times 10^6 = \frac{\pi}{16} \times \sigma_1 \times d^3$$

$$\frac{\pi}{16} \times \sigma_1 \times d^3 = 1.796 \times 10^6$$

$$\frac{\pi}{16} \times 56 \times d^3 = 1.796 \times 10^6$$

$$\boldsymbol{d = 54.67\ mm}$$

According to Equivalent twisting moment, the diameter of the shaft

$$T_e = \sqrt{(M_b)^2 + (T)^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

$$T_e = \sqrt{(M_b)^2 + (T)^2} = \sqrt{(0.563 \times 10^6)^2 + (1.098 \times 10^6)^2}$$

$$T_e = 1.234 \times 10^6$$

$$\frac{\pi}{16} \times \tau_{max} \times d^3 = 1.234 \times 10^6$$

$$\frac{\pi}{16} \times 42 \times d^3 = 1.234 \times 10^6$$

$$\boldsymbol{d = 53.08\ mm}$$

Case (ii) fluctuating load

K_b and K_t – are taken from PSGDB pg.no. 7.21

K_b = 1.5 and K_t = 1

According to Equivalent bending moment, the diameter of the shaft

$$M_e = \left[K_b M_b + \sqrt{(K_b M_b)^2 + (K_t T)^2}\right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

$$M_e = \left[K_b M_b + \sqrt{(K_b M_b)^2 + (K_t T)^2}\right]$$

$$M_e = \left[1.5 \times 0.563 \times 10^6 + \sqrt{(1.5 \times 0.563 \times 10^6)^2 + (1 \times 1.098 \times 10^6)^2}\right]$$

$$M_e = 2.23 \times 10^6 = \frac{\pi}{16} \times \sigma_1 \times d^3$$

$$\frac{\pi}{16} \times \sigma_1 \times d^3 = 2.23 \times 10^6$$

$$\frac{\pi}{16} \times 56 \times d^3 = 2.23 \times 10^6$$

$$\boldsymbol{d = 58.75\ mm}$$

According to Equivalent twisting moment, the diameter of the shaft

$$T_e = \sqrt{(K_b M_b)^2 + (K_t T)^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

$$T_e = \sqrt{(K_b M_b)^2 + (K_t T)^2} = \sqrt{(1.5 \times 0.563 \times 10^6)^2 + (1 \times 1.098 \times 10^6)^2}$$

$$T_e = 1.385 \times 10^6\ N - mm$$

$$\frac{\pi}{16} \times \tau_{max} \times d^3 = 1.385 \times 10^6$$

$$\frac{\pi}{16} \times 42 \times d^3 = 1.385 \times 10^6$$

$$\boldsymbol{d = 55.17\ mm}$$

2. Power is transmitted to a shaft supported on bearings, 900mm apart, by a belt drive, running on a 450 mm pulley, which overhangs the right bearing by 200 mm. Power is transmitted from the shaft through a belt drive, running on a 250 mm pulley, located mid-way between the bearings. The belt drives are at right angle to each other and the ratio of belt tensions is 3:1 with the maximum tension in both the belts being limited to 2 kN. Determine the diameter of the shaft, assuming permissible tensile and shear stresses are 100 MPa and 60 MPa respectively.

Given Data:

L = 900 mm; d_a = 450 mm; l_a = 200 mm; d_b = 250 mm; l_b = 450 mm; T_1:T_2 = 3:1; T_{max} = 2x10^3 N; τ = 60 N/mm^2; σ = 100 N/mm^2

To find:

Diameter of the shaft

Solution:

To find the load acting on the shaft, in this case the load is nothing but the tensions prevailing in the belt.

To find the belt tensions of the pulley A

Given, T_1:T_2 = 3:1

$$\frac{T_1}{T_2} = 3 \implies T_2 = \frac{T_1}{3} = \frac{2 \times 10^3}{3} = 666.67\ N$$

To find the torque produced by the system,

$$T = (T_1 - T_2) \times \frac{d_A}{2} = (2000 - 666.67) \times \frac{450}{2}$$

$$T = 3 \times 10^3\ N.mm$$

Let us assume that the troque transmitted by both pulleys A and B are same, therefore

$$\frac{T_3}{T_4} = 3 \implies T_3 = 3T_4$$

$$(T_3 - T_4) \times \frac{d_B}{2} = 3 \times 10^3$$

$$(3\,T_4 - T_4) \times \frac{250}{2} = 3 \times 10^3$$

$$T_4 = 1200\ N$$

In turn,

$$T_3 = 3T_4 = 3 \times 1200 = 3600\ N$$

Since the two belts are running at right angles to each other, the direction of load acting on the shafts are in vertical and horizontal direction, it is calculated as below

Vertical load, P_v

$$P_v = T_1 + T_2 = 2000 + 666.67$$

$$P_v = 2666.67\ N$$

Horizontal load, P_h

$$P_v = T_3 + T_4 = 1200 + 3600$$

$$P_v = 4800\ N$$

The shaft is subjected to bending moment due to these horizontal and vertical loads, and it is found as

Bending moment due to Horizontal load, M_{bh} (From PSGDB pg.no 6.5 for SSB subjected to concentrated load not at the centre is given as)

$$M_{bh} = \frac{P_h\, l_a\, (L - l_a)}{4} = \frac{2666.67 \times 200 \times (900 - 200)}{4}$$

$$M_{bh} = 93.33 \times 10^6\ N.mm$$

Bending moment due to vertical load, M_{bv} (From PSGDB pg.no 6.5 for SSB subjected to concentrated load at the centre is given as)

$$M_{bv} = \frac{P_v\, l_b}{4} = \frac{4800 \times 450}{4}$$

$$M_{bv} = 0.54 \times 10^6\ N.mm$$

To find the resultant bending moment, M_b

$$M_b = \sqrt{(M_{bh})^2 + (M_{bv})^2} = \sqrt{(93.33 \times 10^6)^2 + (0.54 \times 10^6)^2}$$

$$M_b = 93.33 \times 10^6\ N.mm$$

According to Equivalent bending moment, the diameter of the shaft

$$M_e = \left[M_b + \sqrt{(M_b)^2 + (T)^2}\right] = \frac{\pi}{16} \times \sigma_1 \times d^3$$

$$M_e = \left[M_b + \sqrt{(M_b)^2 + (T)^2}\right] = \left[93.33 \times 10^6 + \sqrt{(93.33 \times 10^6)^2 + (3 \times 10^3)^2}\right]$$

$$M_e = 186.66 \times 10^6 = \frac{\pi}{16} \times \sigma_1 \times d^3$$

$$\frac{\pi}{16} \times \sigma_1 \times d^3 = 186.66 \times 10^6$$

$$\frac{\pi}{16} \times 100 \times d^3 = 186.66 \times 10^6$$

$$\boldsymbol{d = 211.84\ mm}$$

According to Equivalent twisting moment, the diameter of the shaft

$$T_e = \sqrt{(M_b)^2 + (T)^2} = \frac{\pi}{16} \times \tau_{max} \times d^3$$

$$T_e = \sqrt{(M_b)^2 + (T)^2} = \sqrt{(93.33 \times 10^6)^2 + (3 \times 10^3)^2}$$

$$T_e = 93.33 \times 10^6$$

$$\frac{\pi}{16} \times \tau_{max} \times d^3 = 93.33 \times 10^6$$

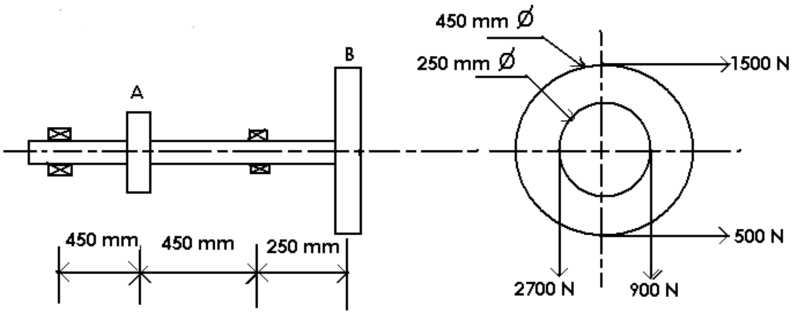
$$\frac{\pi}{16} \times 60 \times d^3 = 93.33 \times 10^6$$

$$\boldsymbol{d = 199.35\ mm}$$

3. A hollow transmission shaft having inner diameter 0.6 times the outer diameter is made of plain carbon steel 40C8 and the factor of safety is 3. A belt & pulley drive of 1000 mm diameter is mounted on the shaft which overhangs the left hand bearing by 250 mm. The belt is vertical and transmits power to the machine shaft below the pulley. The tension on the tight & slack sides of the belts are 3 kN and 1 kN respectively, while the weight of the pulley is 500 N. The angle of wrap of the belt on the pulley is 180°. Calculate the outer & inner diameter of the shaft.

ii) Design of shafts subjected to combined variable fluctuating loads

1. A line shaft supporting two pulleys A and B is shown in Figure. Power is supplied to the shaft by means of vertical belt on pulley A, which is then transmitted to pulley B carrying a horizontal belt. The ratio of belt tensions on tight and loose sides is 3:1 and the maximum tension in either belt is limited to 2.7 kN. The shaft is made of plain carbon steel 40C8 (S_{ut} = 650 N/mm² and S_t = 380 N/mm². The pulleys are keyed to the shaft. Determine the shaft diameter according to the A.S.M.E. code if. k_b = 1.5 and k_t = 1.0



2. A hollow shaft of 1000 mm outside diameter and 500 mm inner diameter is used to drive a propeller shaft of a marine vessel. The shaft is mounted on bearings 5 m apart and it transmits 3200 kW at 200 rpm. The maximum

axial propeller thrust is 500 kN and the shaft weighs 70 kN. Determine the maximum shear stress developed in the shaft and angular twist between the bearings. Take G = 84 GPa and assume the load is applied gradually.

iii) Design of Shaft subjected to Combined axial, transverse and torsional loads (axial, Bending & twisting moment)

1. In an axial flow rotary compressor, the shaft is subjected to a maximum twisting moment of 1500 Nm and a maximum bending moment of 3000 Nm. Neglecting the axial load on the shaft determine the diameter of the shaft, if the allowable shear stress is 50 N/mm^2. If the shaft is to be a hollow one with D/d = 0.4, what will be the material saving in the hollow shaft. It is subjected to the same loading and of the same material as the solid shaft.
2. A hollow steel shaft is to transmit 20 kW at 300 rpm. The loading is such that the maximum bending moment is 1000 N-m, the maximum torsional moment is 500 N-m and axial compressive load is 15 kN. The shaft is supported on rigid bearings 1.5 m apart. The maximum permissible shear stress on the shaft is 40 MPa. The inner diameter is 0.8 times the outer diameter. The load is cyclic in nature and applied with heavy shocks. Determine the size of inner and outer diameters.

iv) Design of shaft based on Torsional rigidity

1. A turbine shaft transmits 500 kW at 900 rpm. The permissible shear stress is 80 N/mm^2 while twist is limited to 0.5 º in a length of 2.5 m. Calculate the diameter of shaft, Take G = 0.8 x 10 5 N/mm^2. If the shaft chosen is hollow with D/d = 0.6. Calculate the percentage of saving in material.

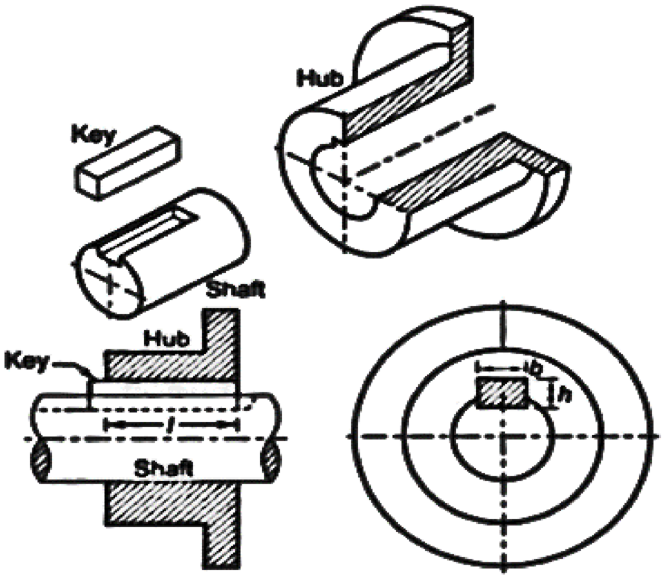
2.5 DESIGN OF KEYS

It is a mechanical element used on shafts to secure rotating of the elements like gears, pulleys or sprockets and prevent relative motion between the two.

Keyway is a slot or recess in shaft and hub of the rotating element to accommodate the key.

2.5.1 CLASSIFICATION OF KEYS

Sunk keys, Saddle keys, Tangent keys, Kennedy keys, Round keys, splines

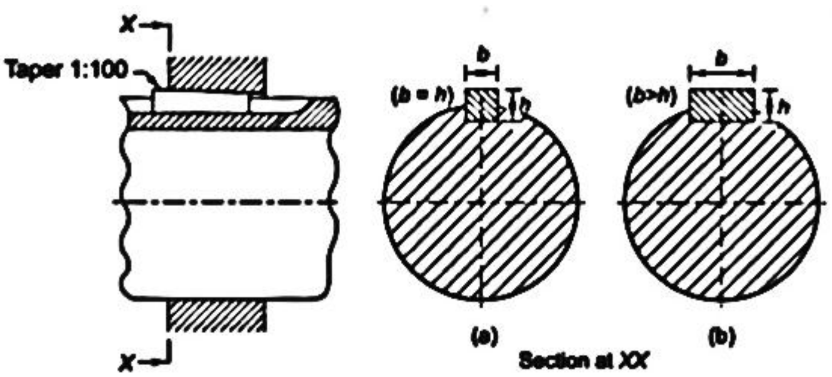


SUNK KEYS:

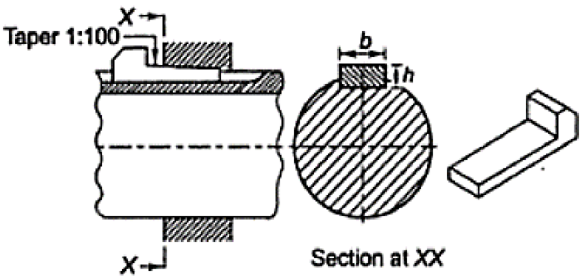
They are provided half in keyway of the shaft and remaining half in the keyway of the hub or boss of the rotating element.

TYPES OF SUNK KEYS

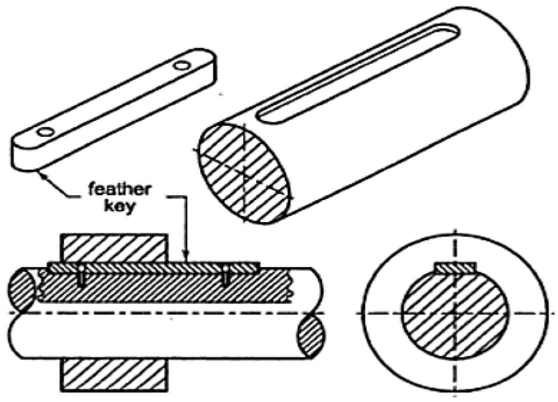
Rectangular key, Square key, Parallel key, Gib head key, Feather key, Woodruff key



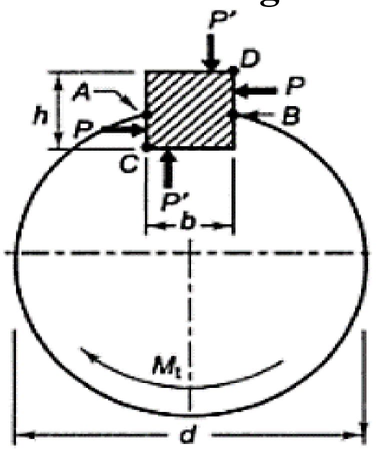
Gib Head Key



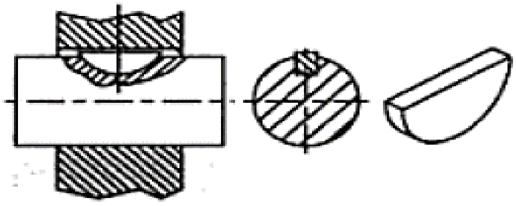
Feather Key



Forces acting on Key

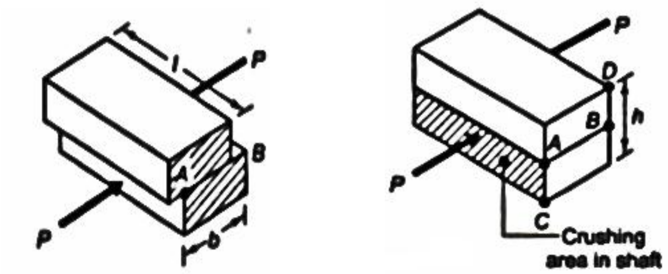


Woodruff Key



2.6 DESIGN OF SQUARE & FLAT KEY

The key subjected to force system as shown in the above figure will have



DIRECT SHEAR STRESS

$$\tau = \frac{P}{area\ of\ the\ plane\ AB} = \frac{p}{bl}$$

Where

b – Width of the key, mm

l – Length of the key, mm

$$Torque, T = p \times \frac{d}{2}$$

Therefore

$$p = \frac{2\,T}{d}$$

On substitution the direct Shear stress becomes

$$\boldsymbol{\tau = \frac{2\,T}{d\,b\,l}, \qquad N/mm^2}$$

Crushing or compressive stress

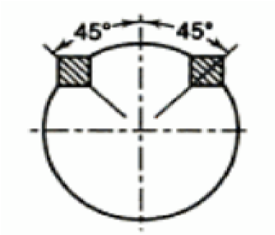
$$\sigma_c = \frac{P}{area\,of\,the\,surface\,AC} = \frac{p}{\frac{h}{2}\,l} = \frac{2\,p}{h\,l}$$

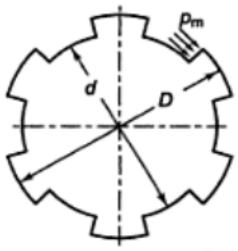
On substituting the load value,

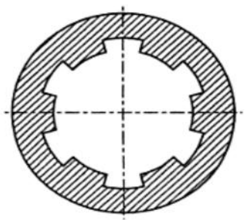
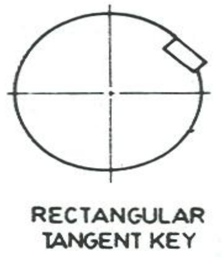
$$\boldsymbol{\sigma_c = \frac{4\,T}{d\,h\,l}, \qquad N/mm^2}$$

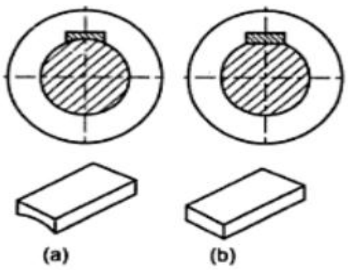
From the above two equations, we see that

$$\boldsymbol{\sigma_c = 2\,\tau}$$







(a) Hollow saddle key, (b) Flat saddle key

Round taper pin

2.7 DESIGN OF TAPER KEYS:

The torque transmitted due to the wedging action of the key

$$T = 0.5\,\mu\,b\,l\,d\,\sigma_c$$

2.8 DESIGN OF KENNEDY KEY

It consists of two square keys on either side of the axis at an angle of 45° to each from its diagonal axis

$$Torque, T = p \times d$$

Therefore

$$p = \frac{T}{d}$$

On substitution the direct Shear stress becomes

$$\boldsymbol{\tau = \frac{T}{\sqrt{2}\,d\,b\,l}, \qquad N/mm^2}$$

Crushing or compressive stress

$$\sigma_c = \frac{\sqrt{2}\,T}{d\,b\,l}, \qquad N/mm^2$$

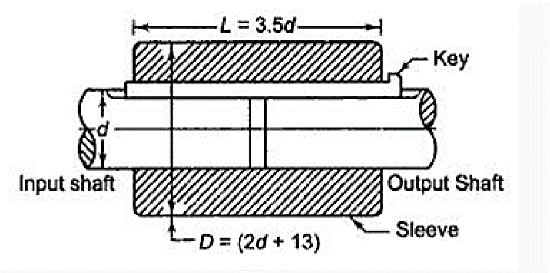
2.9. DESIGN OF COUPLINGS

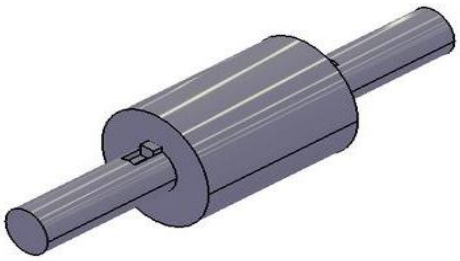
- It is defined as a mechanical device that permanently joins two rotating shafts to each other.
- It is applied over the places where two different shafts of two different machines to form a new machine.
- It introduces mechanical flexibility between two connected units.
- It reduces the transmission of vibrations and shocks between two connected units.
- It makes the provision for disconnection of two units for repair or alterations.

2.9.1 CLASSIFICATION OF COUPLINGS

- Rigid couplings
  - Muff or sleeve coupling
  - Split muff or clamp or compression coupling
  - Flange coupling
- Flexible couplings
  - Bushed pin type coupling
  - Oldham coupling
  - Universal coupling
- The rigid couplings are used to connect shafts which are perfectly aligned.
- The flexible couplings are used to connect two shafts having a small amount of lateral or angular misalignment.

2.10 BOX OR MUFF OR SLEEVE COUPLING





- It consists of a sleeve which is fitted over input and output shafts by means of sunk key.
- It is simple in construction, safer, cheaper.
- It is difficult to assemble and dismantle, needs accurate alignment
- It is used for shafts upto 70 mm in size.
- The standard proportions used in practice

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

  - D – diameter of the sleeve, mm
  - L – axial length of sleeve, mm
  - d – diameter of shaft, mm

2.10.1 DESIGN PROCEDURE

Step: 1 to find the diameter of the shaft, using

Torque or the twisting moment acting on the shaft is given as

$$T = \frac{60 \times P}{2\,\pi\,N}, \qquad N.mm$$

The torsional shear stress due to torque M_t is given as

$$\tau = \frac{16\,M_t}{\pi\,d^3}, \qquad N/mm^2$$

Step: 2 find the dimensions of the sleeve by using

D = (2d + 13) mm

L = 3.5 d mm

Check for torsional shear stress,

$$\tau = \frac{16\,T\,D}{\pi\,(D^4 - d^4)}, \qquad N/mm^2$$

Step: 3 Design the sunk key for its dimensions, take length of the key, l = L/2

The dimensions of the sunk key is to be taken from the PSGDB

Width of the key, b =?

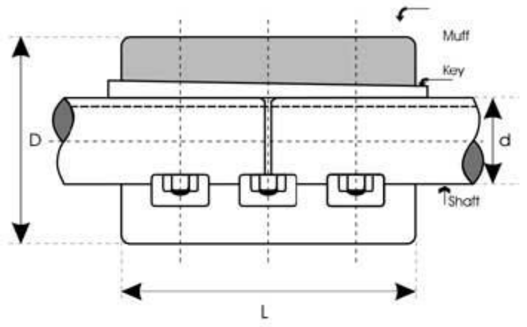
Height of the key, h =?

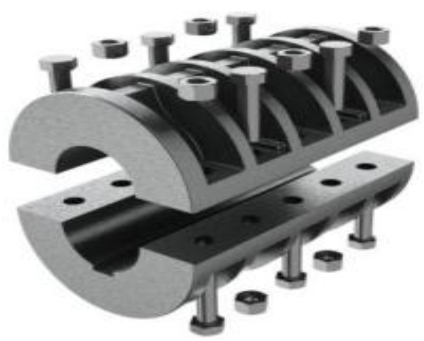
Check for shear and crushing stresses

$$\tau = \frac{2\,T}{d\,b\,l}, \qquad N/mm^2$$

$$\sigma_c = \frac{4\,T}{d\,h\,l}, \qquad N/mm^2$$

2.11 SPLIT MUFF OR CLAMP OR COMPRESSION COUPLING





- In this sleeve is made of two halves split along a plane passing through the axes of the shaft.
- It is connected by means of 4 or 8 bolts.
- Easy to assemble and dismantle
- Difficult to balance dynamically and unsuitable for shock loads

D = 2.5 d and L = 3.5 d are the standard proportions

2.11.1 DESIGN PROCEDURE

Step: 1 to find the diameter of the shaft, using

Torque or the twisting moment acting on the shaft is given as

$$\boldsymbol{T = \frac{60 \times P}{2\,\pi\, N}}, \qquad \boldsymbol{N.mm}$$

The torsional shear stress due to torque M_t is given as

$$\boldsymbol{\tau = \frac{16\,T}{\pi\, d^3}}, \qquad \boldsymbol{N/mm^2}$$

Step: 2 find the dimensions of the sleeve and clamping bolts by using

D = 2.5 d, mm & L = 3.5 d, mm

d_b = 0.15d + 15 mm

Step: 3 calculate the diameter of the clamping bolts

To find the clamping force per bolt

$$P_b = \frac{\pi}{4}\, {d_b}^2\, \sigma_t, \qquad N$$

It is assumed that the half the number of bolts gives clamping pressure over input shaft and remaining to the output shaft, therefore clamping force on each shaft is given as

$$F_c = \frac{P_b . n}{2}, \qquad N$$

The frictional force acting on the bolts is given as

$$F = \mu \times F_c$$

And the torque acting on the bolts due to frictional force, F

$$T = F \times \frac{d}{2}$$

From the above equation the diameter of the bolt can be found

Where

P_b = clamping force per bolt, N

F_c = total clamping force = (frictional force F)

d_b = diameter of the bolt, mm

σ_t = tensile stress of the bolt material, N/mm^2

μ = coefficient of friction, (always 0.3)

n = no of bolts (even number of bolts to be selected)

Step: 4 Design the sunk key for its dimensions, take length of the key, l = L/2

The dimensions of the sunk key are to be taken from the PSGDB. Pg.no.5.16

Width of the key, b =?

Height of the key, h =?

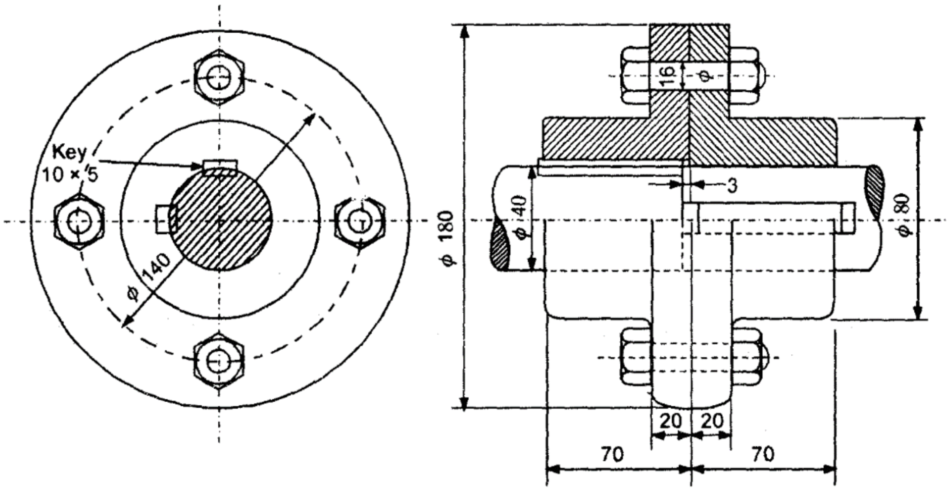
Check for shear and crushing stresses

$$\boldsymbol{\tau = \frac{2\,T}{d\,b\,l}}, \qquad \boldsymbol{N/mm^2}$$

$$\boldsymbol{\sigma_c = \frac{4\,T}{d\,h\,l}}, \qquad \boldsymbol{N/mm^2}$$

2.12 RIGID FLANGE COUPLING

- This coupling has two flanges, one keyed to input shaft and the other to output shaft and connected by means of bolts.
- It has three regions inner hub, central flange with bolts & holes & peripheral outer rim.
- It has high transmitting capacity and easy to assemble & dismantle.
- It cannot tolerate the misalignment and requires more radial space.
- There are two types 1) Protected type & 2) Unprotected type



Rigid unprotected type flanged coupling.

Standard proportions for various dimensions of the flange

- ✓ Outside diameter of the hub, $d_h = 2$ d
- ✓ Length of the hub or effective key length, $l_h = 1.5$ d
- ✓ Pitch circle diameter of bolts, D = 3 d
- ✓ Thickness of flanges, $t_f = 0.5$ d
- ✓ Thickness of the protecting rim, $t_r = 0.25$ d
- ✓ Diameter of spigot & recess, $d_{sr} = 1.5$ d
- ✓ Outside diameter of flange, $D_o = (4\,d + 2\,t_r)$
- ✓ No of bolts
    - i. n = 3, for shafts upto 40 mm dia

ii. n = 4 , for shafts upto 40 to 100 mm dia

iii. n = 6 , for shafts upto 100 to 180 mm diameter

Design procedure

Step: 1 to find the diameter of the shaft, using

Torque or the twisting moment acting on the shaft is given as

$$T = \frac{60 \times P}{2\,\pi\,N}, \qquad N.mm$$

The torsional shear stress due to torque M_t is given as

$$\tau = \frac{16\,T}{\pi\,d^3}, \qquad N/mm^2$$

Step: 2 find the dimensions of the flange

Check for torsional shear stress of the hub,

$$T = \frac{\pi}{16}\,\frac{\left({d_h}^4 - d^4\right)}{d_h}\,\tau$$

Check for torsional shear stress of the hub & flange junction

$$T = \frac{\pi}{2}\,{d_h}^2\,t_f\,\tau$$

Step: 3 calculate the diameter of the bolt

$${d_b}^2 = \frac{8\,T}{\pi\,D\,n\,\tau_b}$$

τ_b – permissible shear stress of the bolt, N/mm^2

Check for crushing stress,

$$\sigma_c = \frac{2\,T}{n\,d_b t_f\,D}$$

Step: 4 calculate the dimensions of the key

Take length of the key as, $l = l_h + t_f$

The dimensions of the sunk key is to be taken from the PSGDB. Pg. no. 5.16

Width of the key, b =?

Height of the key, h =?

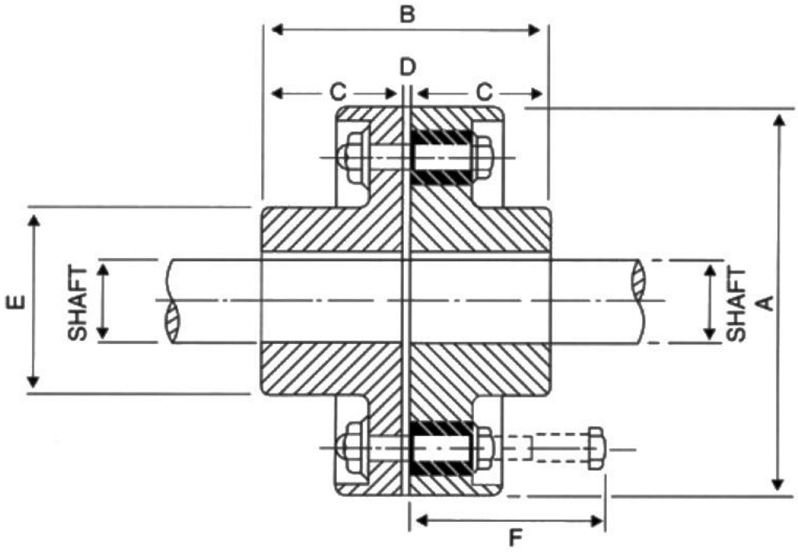
Check for shear and crushing stresses

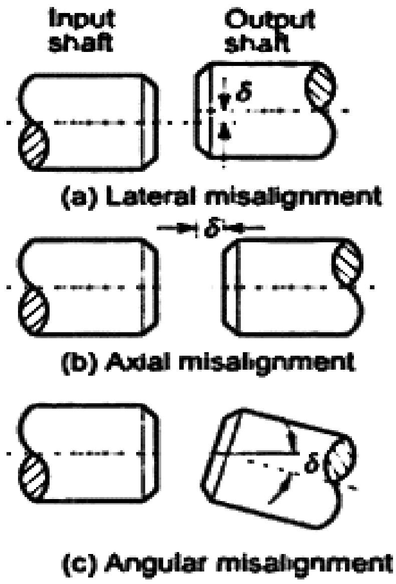
$$\tau = \frac{2\,T}{d\,b\,l}, \qquad N/mm^2$$

$$\sigma_c = \frac{4\,T}{d\,h\,l}, \qquad N/mm^2$$

2.13 BUSHED PIN TYPE FLEXIBLE COUPLING

- It is similar to rigid flange coupling but a rubber bush and pins are used.
- This gives a 0.5° of angular misalignment and 0.5 mm of lateral misalignment





DESIGN PROCEDURE:

Step: 1 to find the diameter of the shaft, using

Torque or the twisting moment acting on the shaft is given as

$$T = \frac{60 \times P}{2\,\pi\,N}, \qquad N.mm$$

The torsional shear stress due to torque M_t is given as

$$\tau = \frac{16\,T}{\pi\,d^3}, \qquad N/mm^2$$

Step: 2 find the dimensions of the flange

From PSGDB pg.no. 7.106-7.108

A - Diameter of the shaft (d)

B - Outer diameter of the flange

C - Diameter of the hub

D - Pitch circle diameter of the pins or bolts or bush

E - Length of the hub

F - Diameter of bolts or pins

G - Length of the bush in the flange

H - Protective layer thickness

n - Number of bolts or pins

d_b - Diameter of the bush

t - Clearance

Check for torsional shear stress of the hub,

$$T = \frac{\pi}{16} \frac{\left(C^4 - A^4\right)}{C}\ \tau$$

Check for torsional shear stress of the hub & flange junction

$$T = \frac{\pi}{2} C^2 \, t_f \, \tau$$

Where t_f – thickness of the flange taken as 0.5 A

Step: 3 check the dimensions of the pin or bolts

The force acting on pin or bush is given as

$$T = P \times n \times \frac{D}{2}$$

The bolts or pins are subjected to shear stress and bending stress

Shear stress acting on the pins or bolts is given as

$$\tau = \frac{P}{\frac{\pi}{4} F^2}$$

Bending stress acting on the bolts or pins is given as

$$\sigma_b = \frac{M_b \, y}{I} = \frac{32 \, P \left(\frac{G}{2} + t\right)}{\pi \, F^3}$$

The resultant shear stress acting on the pins or bolts is found as

$$\tau_{max} = \frac{1}{2} \sqrt{{\sigma_b}^2 + 4 \, \tau^2}$$

The resultant shear stress should be less than permissible value

Step: 4 check for the dimensions of the bush

Length of the bush is taken as

$$L = G + t - \frac{2}{3} F$$

The clamping or bearing pressure acting on the bush is given as

$$P_b = \frac{P}{d_b \times L}$$

Step: 5 calculate the dimensions of the key

Take length of the key as, $l = E + t_f$

The dimensions of the sunk key is to be taken from the PSGDB

Width of the key, b =?

Height of the key, h =?

Check for shear and crushing stresses

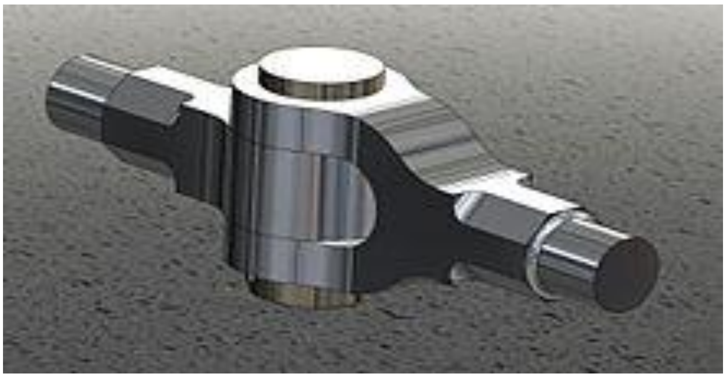
$$\tau = \frac{2 \, T}{d \, b \, l'}, \qquad N/mm^2$$

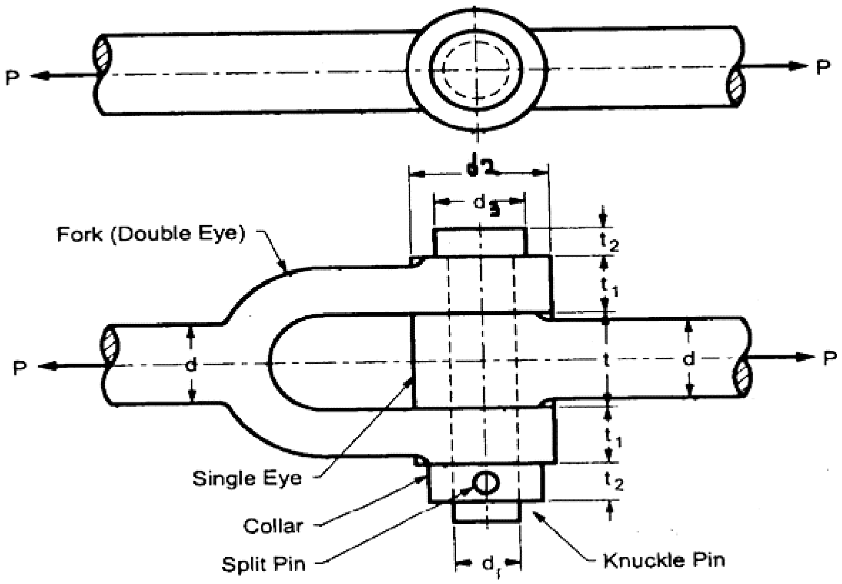
$$\sigma_c = \frac{4 \, T}{d \, h \, l'}, \qquad N/mm^2$$

2.14 DESIGN OF KNUCKLE JOINTS

- A Knuckle joint is used to connect two rods which are under the action of tensile forces, when a small amount of flexibility or angular movement is necessary.
- Examples- link of a roller chain, tension link in a bridge structure, tie rod of roof truss, tie rod of jib crank.

- It consists of three parts – an eye, a fork and a knuckle pin. The end of one rod is formed into eye and the other end of rod into fork (or double eye)





DESIGN PROCEDURE FOR KNUCKLE JOINT:

Step: 1 To find the diameter of the rod,

The rod is subjected to tensile stresses due to the applied tensile load, therefore

$$\sigma_t = \frac{P}{A} = \frac{P}{\frac{\pi}{4}\,d^2}$$

Step: 2 To find the dimensions of the knuckle joint

Diameter of the pin,	d_p	= d
Diameter of the pin head,	d_h	= 1.5 d

Outer diameter of the eye,	D	= 2 d
Thickness of the eye,	t	= 1.25 d
Thickness of the fork,	t_1	= 0.75 d
Thickness of the pin head,	t_2	= 0.5 d

Step: 3 To check for various modes of failure

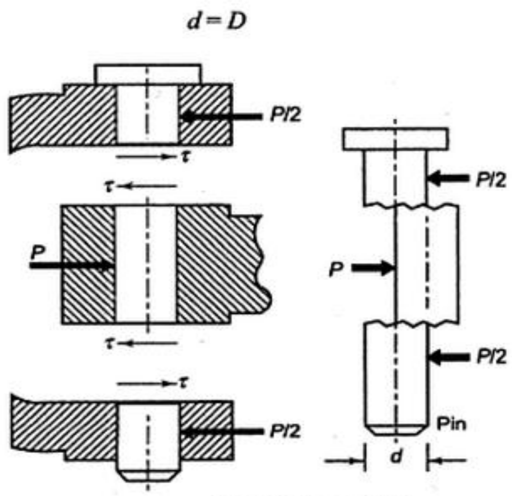
a. Failure of the solid rod in tension

$$\sigma_t = \frac{P}{A} = \frac{P}{\frac{\pi}{4}\,d^2}$$

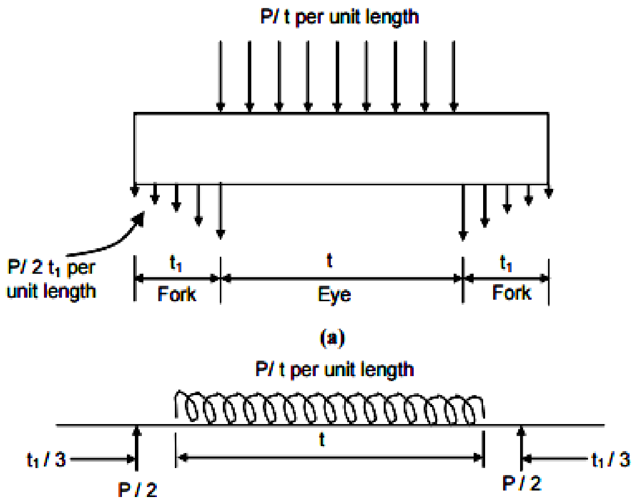
b. Failure of the knuckle pin

- By double shear

$$\tau = \frac{P}{2 \times \frac{\pi}{4}\,{d_p}^2}$$



Shear Failure of Pin



- By bending stress

$$\sigma_b = \frac{p\left(\frac{t_1}{6} + \frac{t}{8}\right)}{\frac{\pi}{32}\,{d_p}^3}$$

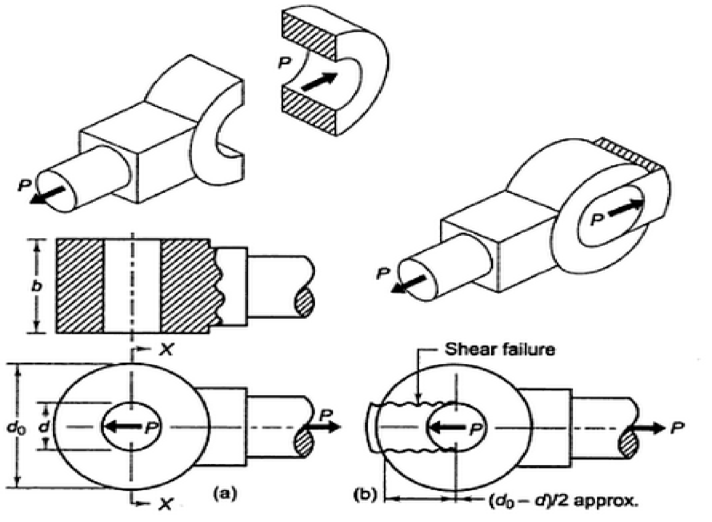
c. Failure of the single eye or rod end

- By tension

$$\sigma_t = \frac{P}{(D - d_p)\,t}$$

- By double shear

$$\tau = \frac{P}{(D - d_p)\,t}$$



(a) Tensile Failure of Eye (b) Shear Failure of Eye

- By crushing

$$\sigma_c = \frac{P}{d_p\, t}$$

d. Failure of forked end

- By tension

$$\sigma_t = \frac{P}{2(D - d_p)\, t_1}$$

- By double shear

$$\tau = \frac{P}{2(D - d_p)\, t_1}$$

- By crushing

$$\sigma_c = \frac{P}{2\, d_p\, t_1}$$

All the values obtained should not exceed the permissible limit provided for the knuckle joint material.

MULTIPLE CHOICE QUESTIONS

S.No	QUESTION	Opt 1	Opt 2	Opt 3	Opt 4	ANSWER
1	The taper on the rectangular sunk key is ________	1 in 16	1 in 32	1 in 48	1 in 100	1 in 100
2	The usual proportion for the width of the key is ___________	d/8	d/6	d/4	d/2	d/4
3	When a pulley or other mating pirce is required to slide along the shaft, a ____________sunk key is used	Rectangular	square	parallel	Round	parallel
4	A key made of cylindrical disc having segmental cross-section, is known as	Feather key	Gib head key	Wood ruf key	Flat saddle key	Flat saddle key
5	A feather key is generally	Loose in shaft and tight in hub	Tight in shaft and loose in hub	Tight in both shaft and hub	Loose in bith shaft and hub	Tight in shaft and loose in hub
6	The type of stresses developed in the key is/are	Shear stress alone	Bearing stress alone	Both shear and bearing stresses	Shearing, bearing and bending stresses	Both shear and bearing stresses
7	For a square key made of mild steel, the shear and crushing strengths are related as	Shear strength = Crushing strength	Shear strength > Crushing strength	Shear strength < Crushing strength	None of the above	Shear strength = Crushing strength
8	A key way lowers	The strength of the shaft	The rigidity of the shaft	Both the strength and rigidity of the shaft	The ductility of the material of the shaft	Both the strength and rigidity of the shaft
9	The sleeve of the muff coupling is designed as a	Thin cylinder	Thick cylinder	Solid shaft	Hollow shaft	Hollow shaft
10	Oldham coupling is used to connect two shafts	Which are perfectly designed	Which are not in exact alignment	Which have lateral misalignment	Whose axis intersect at a small angle	Which have lateral misalignment
11	The standard length of the shaft is	5m	6m	7m	All of these	All of these
12	Two shafts A and B are made of the same material. The diameter of the shaft A is twice as that of shaft B. The power transmitted by the shaft A will be ___________of shaft B	Twice	Four times	Eight times	Sixteen times	Eight times
13	Two shafts A and B of solid circular cross-section are identical except for their diameters dA and dB.The ratio of power transmitted by the shaft A to that of shaft B is	d_A/d_B	$(d_A)^2/(d_B)^2$	$(d_A)^3/(d_B)^3$	$(d_A)^4/(d_B)^4$	$(d_A)^3/(d_B)^3$
14	Two shafts will have equal strength, if	Diameter of the both shaft is same	Angle of twist of both the shafts is same	Material of both the shaft is same	Twisting moment of the both shaft is same	Twisting moment of the both shaft is same
15	\The maximum permissible deflection for the transmission shaft is taken as________	0.001 L to 0.003 L	0.01 L to 0.03 L	0.1 L to 0.3 L	0.0001 L to 0.0003 L	0.001 L to 0.003 L
16	The stiffness & strength of a hollow shaft is ______ than the solid shaft with same weight	more	equal	less	more & less	more
17	_________is an auxillary shaft used in power transmission between two shafts	Line shaft	Jack Shaft	counter shaft	spindle	Jack shaft
18	Ordinary transmission shafts are made of_________	High carbon steel	Low carbon steel	Medium carbon steel	alloy steel	medium carbon steel

19	The concept of equivalent torsional moment is used in the design of shafts based on which theory	Guest's theory	Rankine's theory	St. Venant's theory	Von mises theory	Guest's theory
20	The permissible angle of twist for machine tool applications is _______ per meter length	0.15°	0.25°	0.35°	0.45°	0.25°
21	The term 'axle' is used for a shaft that supports rotating elements like_________	Wheels	hoisting drums	rope sheeves	All of these	All of these
22	A transmision shaft subjected to the bendung loads must be designed on the basis of	Maximum normal stress theory	Maximum shear stress theory	Maximum normal and shear stress theories	Fatigue strength	Maximum normal stress theory
23	which of the following loading is considered for the design of the axles?	Bending moment only	Twisting moment only	Combined bending moment and torsion	Combined action of bending moment, twisting moment and axial thrust	Bending moment only
24	When a shaft is subjected to a bending moment M and a twisting moment T, then the equivalent twisting moment is equal to	M + T	$M^2 + T^2$	$\sqrt{M^2 + T^2}$	$\sqrt{M^2 - T^2}$	$\sqrt{M^2 + T^2}$
25	The ASME code is based on which theory	Guest's theory	Rankine's theory	St. Venant's theory	Von mises theory	Guest's theory
26	The maximum shear stress theory is used for	Brittle materials	Ductile materials	Plastic materials	Non-ferrous materials	Ductile materials
27	The maximum normal stress theory is used for	Brittle materials	Ductile materials	Plastic materials	Non-ferrous materials	Brittle materials
28	The design of shaft made of brittle materials is based on	Guest's theory	Rankine's theory	St. Venant's theory	Von mises theory	Rankine's theory
29	For two parallel shafts, the distance between whose axes is small and variable, which coupling will you use?	Muff coupling	Universal joint	Falnge coupling	Oldham's coupling	Oldham's coupling
30	Kennedy keys are used for applications like	Precision duty	Light duty	Rough and heavy services	None of the above	Rough and heavy services
31	Which key transmitts power through frictional resistance only	sunk	Kennedy	Flat	Saddle	Saddle
32	Muff coupling is used to join the two shafts which	Have lateral misalignment	Whose axes intersect at a small angle	Are not in exact alignment	Is the simplest type of rigid coupling	Is the simplest type of rigid coupling
33	The flexible coupling can tolerate ______ of lateral misalignment	0.8 mm	0.3 mm	0.5 mm	0.05 mm	0.5 mm
34	A_____ is a piece of M.S inserted between the shaft and hub	Sleeve	Key	Stud	Bolt	Key
35	Key is inserted always ____ to the axis	Parallel	Perpendicular	Towards	Axially	Parallel
36	The ________ key is easily adjustable	Sunk key	Saddle key	Tangent key	Woodruff key	Woodruff key
37	In the following keys which is the type of saddle key	Flat saddle key	Square saddle key	Tangent key	Solid saddle key	Flat saddle key
38	The ______ key is fitted in pair at right angles	Tangent keys	Saddle keys	Wood ruf key	Feather key	Tangent keys
39	The IS code used for specifying the parallel keys and keyways is	IS 2048 - 1983	IS 2292 - 1974	IS 2293 - 1974	IS 2048 - 1974	IS 2048 - 1983

40	A key atached to one member of pair and which permits relative axial movement is known as _____	Tangent keys	Saddle keys	Feather keys	Woodruff key	Feather keys
41	________affects the load carrying capacity of the shaft	Key	Key way	Hub	sleeve	Key way
42	The stress concentration is occurs in the ______	Centre	Corners	Edges	Face	Corners
43	A good shaft should be ________	Light weight	More weight	Easy to connect	Corossion resistance	Easy to connect
44	A good shaft should have ________	Projecting parts	No projecting parts	Non uniform area	No unity in cross section	No projecting parts
45	___________is used to connect two shafts rigidly connected	Sleeve coupling	Flexible coupling	Shaft coupling	Rigid coupling	Rigid coupling
46	________shafts form an integral part of the machine itself	Machine shafts	Short shaft	Transmission shaft	Rigid shaft	Transmission shaft
47	In torsion equation J = ?	Polar moment of inertia	Internal stress	Rigidity modulus	Moment of inertia	Polar moment of inertia
48	In design of the shaft main reason should be considered is _______	Strength	Roughness	Corrosive resistance	Wear resistance	Strength
49	__________ is important in design of camshaft	Torsional rigidity	Rigidity	Torque	Wear resistance	Torsional rigidity
50	Compression coupling is also called as _______	Universal joint	Muff coupling	Falnge coupling	Oldham's coupling	Muff coupling

UNIT – III

DESIGN OF FASTENERS AND WELDED JOINTS

Threaded fasteners – Design of bolted joints including eccentric loading – Design of welded joints for pressure vessels and structures – theory of bonded joints.

Introduction

In threaded joints two or more machine members are joined together with the help of threaded fastening e.g. a nut and bolt. These are non-permanent type joints i.e. members can be disassembled without damaging the component parts for the purpose of maintenance, checking and replacement. Threads are formed by cutting a helical groove on the surface of a cylindrical rod or cylindrical hole.

Advantages & Disadvantages

Threaded fasteners are standardized, and a wide variety is available for different operating conditions and applications. These are easy to manufacture, and a high accuracy can be maintained. Holes are required in the machine parts to be assembled by threaded joints, which lead to stress concentration. Another disadvantage is that, threaded joints tend to loosen up when subjected to vibrations.

Terminologies of Screw Thread

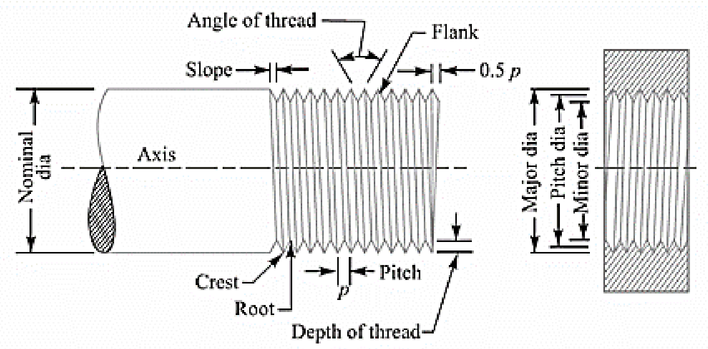


Figure. 1Terminology of Screw Threads

Figure .1 shows some important terms used in screw threads

Major diameter: It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as outside or nominal diameter.

Minor diameter: It is the smallest diameter of an external or internal screw thread. It is also known as core or root diameter.

Pitch diameter: It is the diameter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of the thread and the width of the spaces between the threads. It is also called an effective diameter. In a nut and bolt assembly, it is the diameter at which the ridges on the bolt are in complete touch with the ridges of the corresponding nut.

Pitch: It is the distance from a point on one thread to the corresponding point on the next. This is measured in an axial direction between corresponding points in the same axial plane.

Mathematically,

$$Pitch = \frac{1}{No.of\ threads\ per\ unit\ length\ of\ screw}$$

Lead: It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is equal to the pitch in case of single start threads; it is twice the pitch in double start, thrice the pitch in triple start and so on.

Crest: It is the top surface of the thread.

Root: It is the bottom surface created by the two adjacent flanks of the thread.

Depth of thread: It is the perpendicular distance between the crest and root.

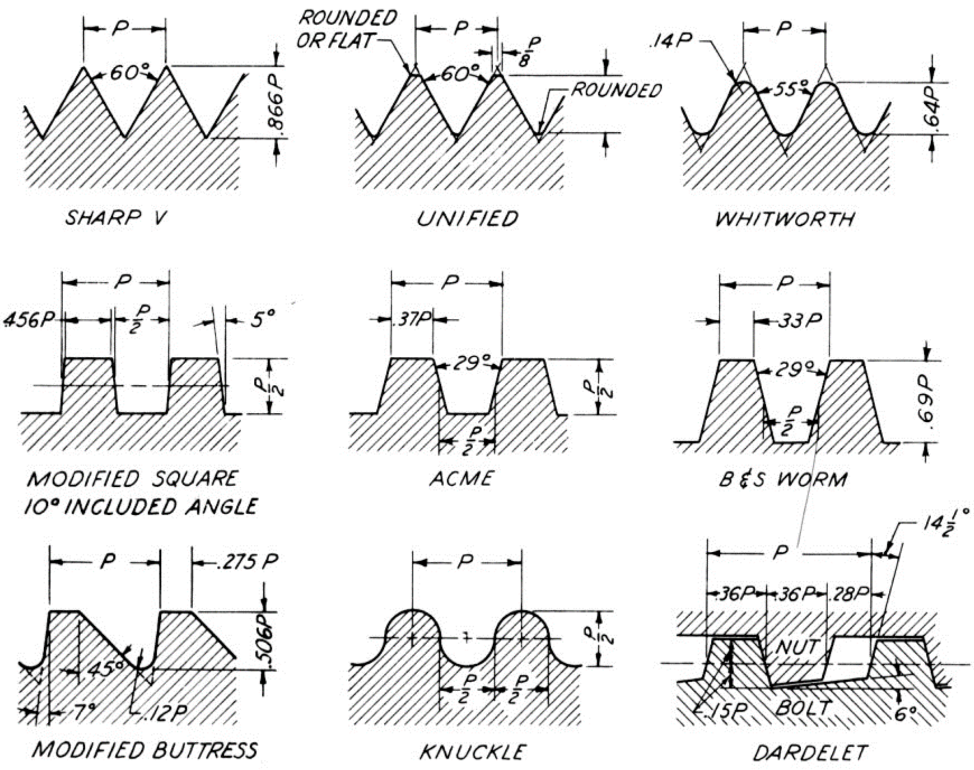
Flank: It is the surface joining the crest and root.

Angle of thread: It is the angle included by the flanks of the thread.

Slope: It is half the pitch of the thread.

Various forms of screw threads:

1. ISO Metric thread
2. ACME thread
3. Square thread
4. Buttress thread
5. B.S.W (British standard whit worth)
6. British Association
7. ANS (American National Standard)



Screw threads.

Designation of ISO Metric Screw Threads

Metric threads are divided into

1. Coarse series:

Thread profiles in both the series is similar. These are the basic series,

- Have higher static load carrying capacity, are
- Easier to cut,
- Have less effect on strength because of manufacturing errors and wear and
- Have more even stress distribution.

Coarse threads are used in members, which are free from vibrations.

Coarse thread designation:

A screw thread of coarse series is designated by the letter 'M' followed by the value of the nominal diameter in mm. For example, **'M 12'**.

2. Fine series:

fine threads have

- Greater strength against fluctuating loads and
- Have greater resistance to unscrewing because of its lower helix angle.
- Fine series threads are more dependable in terms of self-loosening.

fine threads are used in parts subjected to dynamic loads and hollow thin walled parts as the coarse threads will weaken the members considerably. Fine threads are also used in the parts where the threads are used for the purpose of adjustment.

Fine thread designation:

A screw thread of fine series is specified by the letter 'M' followed by the values of the nominal diameter and the pitch in mm and separated by the symbol '×'. For example. **M 12 × 1.25**.

Material

Threads are produced by rolling or machining. Because of cold work, the rolled threads are stronger and have better fatigue properties. Threads can also be produced using casting. Selection of material for threaded fasteners depends upon type of loading, operating environment and temperature etc. Plain Carbon Steel is used for common applications and Alloy Steels are used in high temperature applications and where high strength, better fatigue and corrosion resistance is required. Aluminium, Brass and Bronze are also used in specific applications. Generally, a factor of safety of 2 to 3 on the basis of yield strength is considered in case of carbon steels and 1.5 to 3 for alloy steels.

Types of Screw Fasteners

1. **Bolt (Through Bolt):** It is a cylindrical bar with threads for the nut at one end and head at the other end. The cylindrical part of the bolt is known as shank. It is passed through drilled holes in the two parts to be fastened together and clamped them securely to each other as the nut is screwed on to the threaded end. Bolts have hexagonal or square heads.

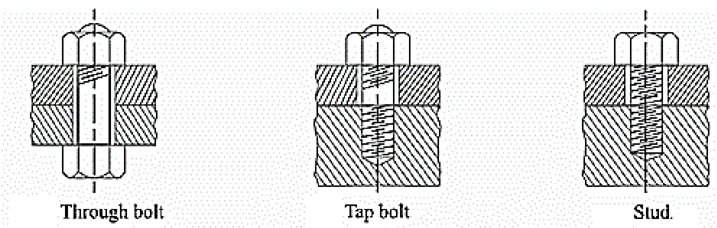
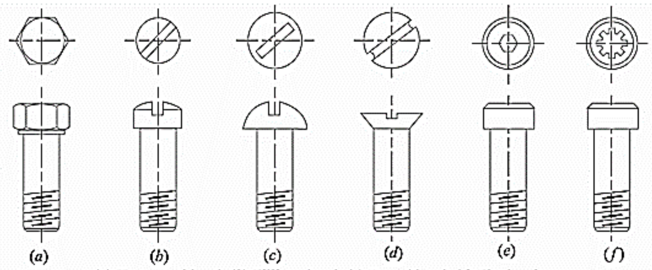


Figure 12.3 Through Bolt, Tap Bolt and Stud

2. **Tap bolts:** Tap bolt is screwed into a tapped hole of one of the parts to be fastened and nut is not used with it.
3. **Studs:** A stud is a round bar threaded at both ends. One end is screwed into a tapped hole of the parts to be fastened, while the other end receives a nut on it.
4. **Cap screws:** The cap screws are like tap bolts except that they are of small size and a variety of shapes of heads are available.



(a) Hexagonal head; (b) Fillister head; (c) Round head; (d) Flat head; (e) Hexagonal socket; (f) Fluted socket.

Types of Cap Screws

5. **Machine screws:** These are like cap screws with the head slotted for a screwdriver and are generally used with a nut.
6. **Set screws:** Set screws are used to prevent relative motion between the two parts. A set screw is screwed through a threaded hole in one part so that its point (i.e. end of the screw) presses against the other part. This resists the relative motion between the two parts by means of friction between the point of the screw and one of the parts.

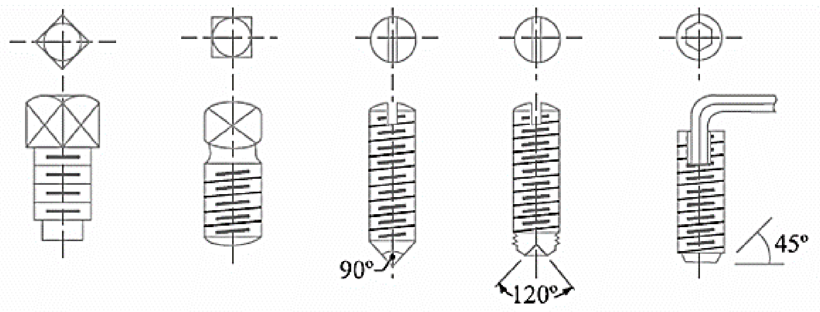
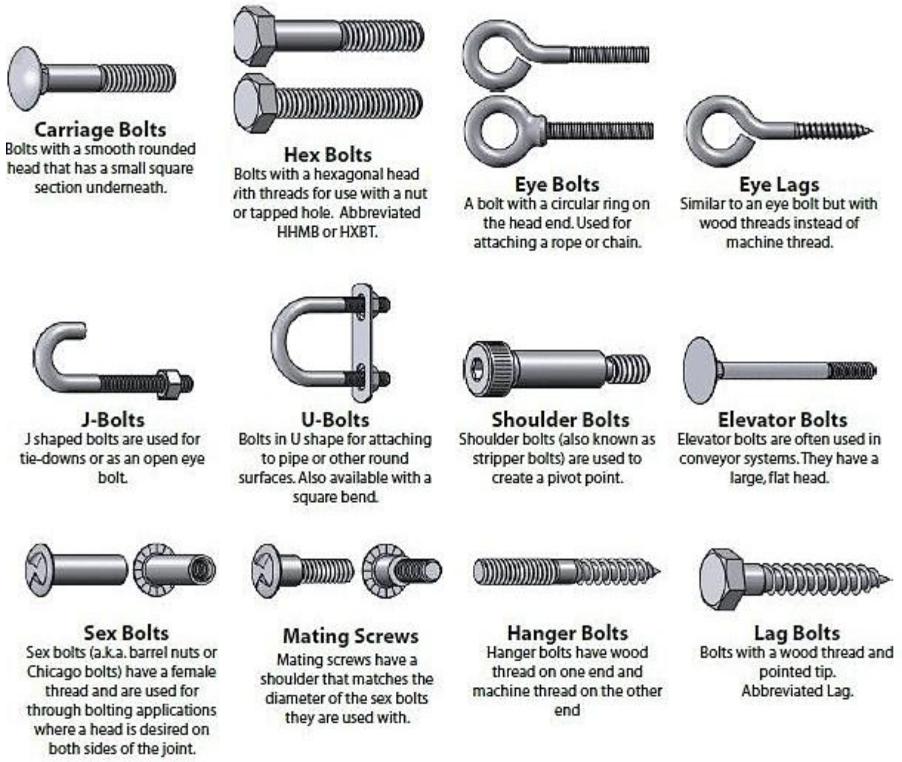
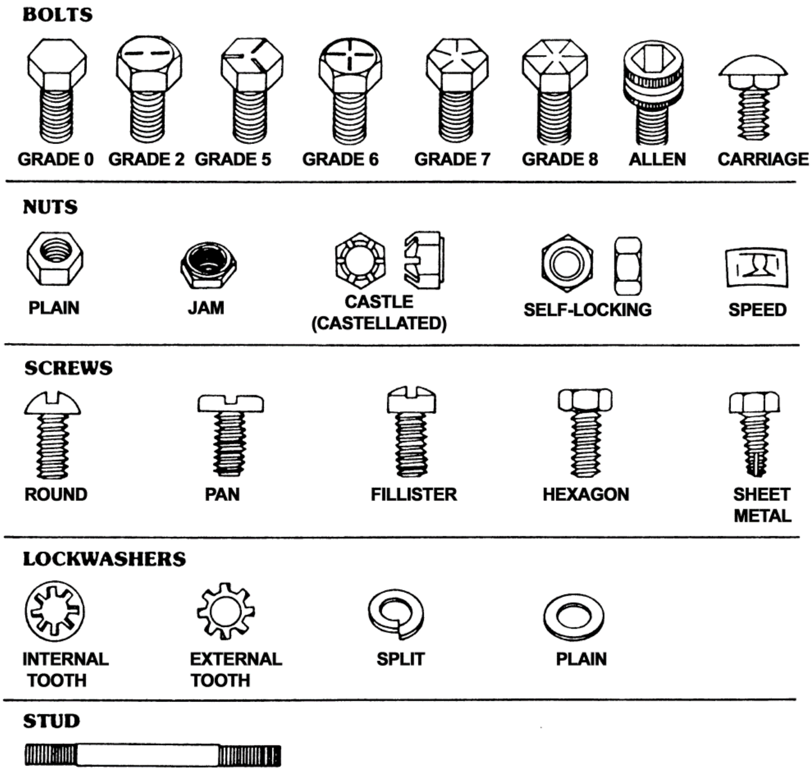


Figure 12.5 Types of Set Screws

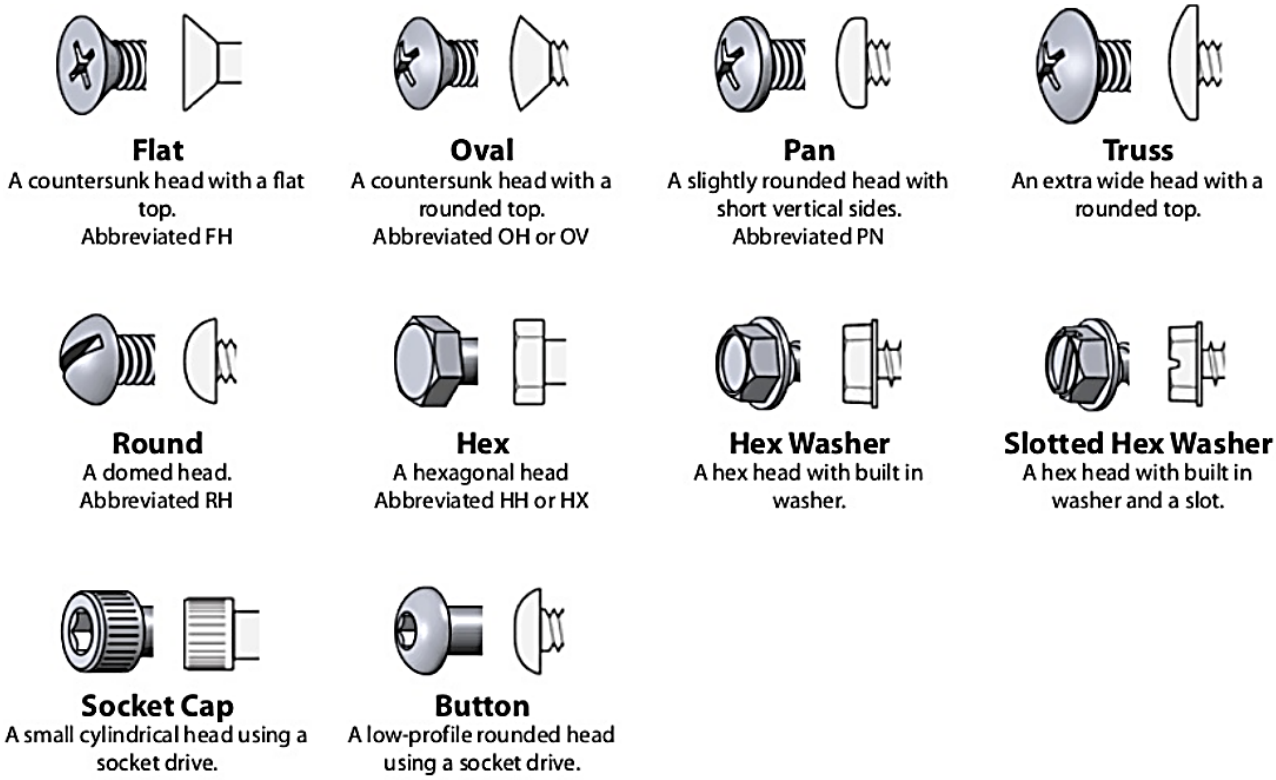
7. **Washers:** They are annular shaped metallic discs; its function is to distribute load over a larger area on the surface of the clamped parts. It prevents marring of clamped parts, bolt and nut surface during assembly. It is also used as a bearing surface over large clearance holes.

Types of Bolts & Nuts, Screws and washers:


Carriage Bolts
Bolts with a smooth rounded head that has a small square section underneath.
Hex Bolts
Bolts with a hexagonal head /ith threads for use with a nut or tapped hole. Abbreviated HHMB or HXBT.
Eye Bolts
A bolt with a circular ring on the head end. Used for attaching a rope or chain.
Eye Lags
Similar to an eye bolt but with wood threads instead of machine thread.
J-Bolts
J shaped bolts are used for tie-downs or as an open eye bolt.
U-Bolts
Bolts in U shape for attaching to pipe or other round surfaces. Also available with a square bend.
Shoulder Bolts
Shoulder bolts (also known as stripper bolts) are used to create a pivot point.
Elevator Bolts
Elevator bolts are often used in conveyor systems. They have a large, flat head.
Sex Bolts
Sex bolts (a.k.a. barrel nuts or Chicago bolts) have a female thread and are used for through bolting applications where a head is desired on both sides of the joint.
Mating Screws
Mating screws have a shoulder that matches the diameter of the sex bolts they are used with.
Hanger Bolts
Hanger bolts have wood thread on one end and machine thread on the other end
Lag Bolts
Bolts with a wood thread and pointed tip. Abbreviated Lag.


BOLTS
GRADE 0
GRADE 2
GRADE 5
GRADE 6
GRADE 7
GRADE 8
ALLEN
CARRIAGE
NUTS
PLAIN
JAM
CASTLE (CASTELLATED)
SELF-LOCKING
SPEED
SCREWS
ROUND
PAN
FILLISTER
HEXAGON
SHEET METAL
LOCKWASHERS
INTERNAL TOOTH
EXTERNAL TOOTH
SPLIT
PLAIN
STUD

Types of Screw Heads:



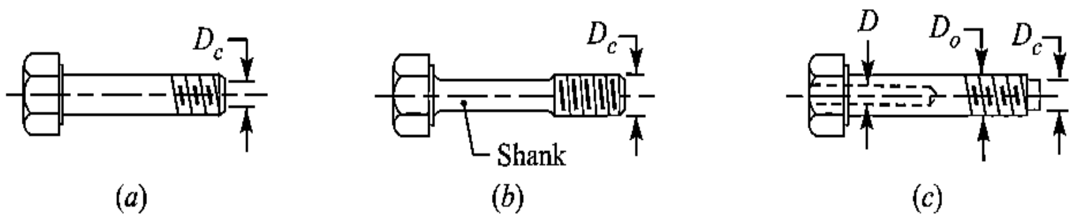
Manufacturing of Bolts:

They are manufactured either by means of thread cutting or thread rolling. In thread rolling, there is no wastage of material, also it gives radius to root and crest. They in turn minimizes the stress concentration. This improves the residual compressive stress which increases the fatigue strength of the bolt.

Bolts of Uniform strength:

When bolts are subjected to impact loading, the normal bolt does not have that high resilience capacity to withstand the shock loads. In order to increase that capacity, the following methods are adopted

1. The shank diameter of the bolt is reduced to the core diameter of the thread or even less.
2. The length of the shank can be increased which increases the modulus of resilience.
3. An axial hole is drilled in the center of the bolt, reducing the cross-sectional area of the bolt.



Bolts of Uniform strength

Stresses acting on the screw fasteners:

The stresses acting on the screw threads due to static loading are as follows

1. Initial stresses:

When the bolt is screwed up tightly, they are subjected to

a. Tensile stress, σ_t due to stretching of bolts
b. Compression or bearing stress on the thread
c. Shearing stress across the threads
d. Torsional shear stress due to friction between the threads
e. Bending stress due to non-parallel surface mating of bolt with the clamped surface

2. Stresses due to external forces

If the bolt is subjected to axial tensile load, the weakest section will be at the root of the thread, due to its minimum diameter,

$$\sigma_t = \frac{External\ forcce}{core\ area\ or\ stress\ area} = \frac{P}{A_c} = \frac{P}{n \times \frac{\pi}{4} \times d_c^2}$$

Where, n – no of bolts, d_c= core diameter.

3. Preloading of the bolts and combined stresses:

In applications like pressure vessel and cylinder covers, it is essential to apply in an intial tightening torque to the make the joint leak proof. For such joints, the initial tension in the bolt is given by

$$P_i = 2840\ d, \qquad N$$

If the joint does require any leak proof joint, then

$$P_i = 1420\ d, \qquad N$$

Where, d is the nominal diameter of the bolt.

When the bolts are subjected to above combined stresses (initial & external) it undergoes elongation and compression for certain rate. This is included in design by considering stiffness of the bolt $q_b$and joining part material q_p,

$$Total\ load\ acting\ on\ the\ bolt, P_T = Preload, P_i + Increase\ in\ load, \Delta P$$

Where,

$$\Delta P = P \times \left(\frac{q_b}{q_p + q_b}\right)$$

Where, P is the external load or force, N

Design of bolts under easy situation:

The following empirical relation is used to calculate the stress or core area of the bolt, by knowing this from PSGDB pg.no. 5.42, the bolt size can be found out.

$$stress\ Area, A_c = \left(\frac{60 \times P_T}{\sigma_Y}\right)^{2/3}, \qquad mm^2$$

The above relation is applicable for d < 45 mm, and for d > 45 mm, the below shown relation is used

$$stress\ Area, A_c = \left(\frac{40 \times P_T}{\sigma_Y}\right)^{2/3}, \qquad mm^2$$

Where, P_T is the total load, σ_Y is the yield stress of the bolt material, N/mm^2

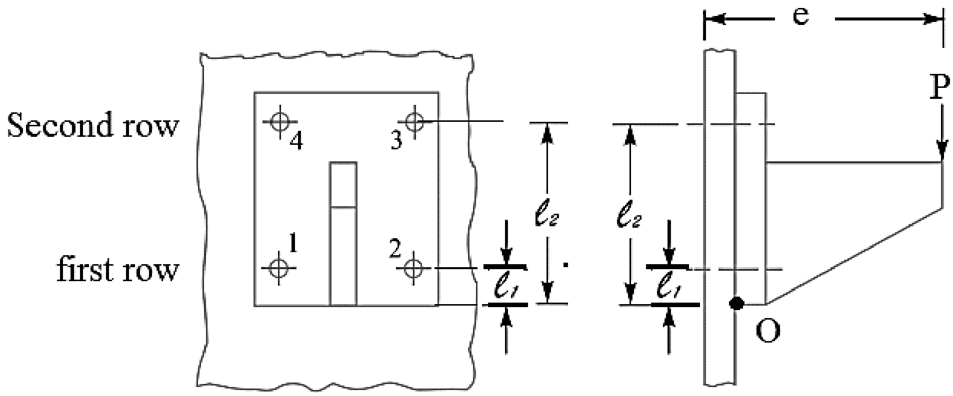
Design of bolts subjected to eccentric loading:

The bolts are subjected to two different forms of eccentric loading based on the plane in which the external load acting,

i. loading of bolts in a plane different from the plane of the bolts

ii. loading of bolts in a plane of the bolts

1. Eccentric loading in different plane (perpendicular to the bolts):



The above figure shows the eccentric loading of bolts in plane different to the plane of the bolts. That is the eccentric load is acting perpendicular to the bolts.

P - Eccentric load, N

e - Eccentric distance, mm

l_1 & l_2 - Distance between the first row and second row of bolts from the fulcrum point, 'O'

n_1 & n_2 - no of bolts in the first and second row

q - Stiffness of the bolt material (load per unit distance), N/mm

F_1 & F_2 - force acting on the first and second row of bolts, N

The following relation is used to find the forces acting on the bolts

$$F_1 = \frac{P \times e \times l_1}{n_1\, l_1^2 + n_2\, l_2^2}, \qquad N$$

$$F_2 = \frac{P \times e \times l_2}{n_1\, l_1^2 + n_2\, l_2^2}, \qquad N$$

Due to these forces, the bolts are subjected to

1. Tensile stress,

$$\sigma_t = \frac{\max of\ F_1 or\ F_2}{A_c}, \qquad N/mm^2$$

2. Shear stress,

$$\tau = \frac{P}{(n_1 + n_2) \times A_c}, \qquad N/mm^2$$

To find the maximum principal normal stress and shear stress

$$\sigma_{maxi} = \frac{1}{2}\left[\sigma_t + \sqrt{\sigma_t^2 + 4\,\tau^2}\right], \qquad N/mm^2$$

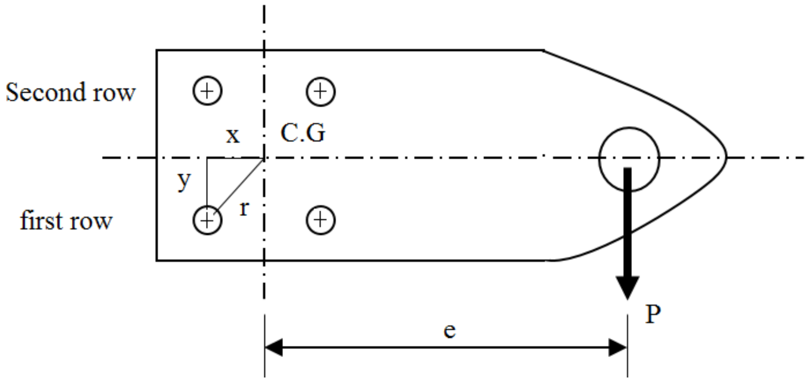
$$\tau_{maxi} = \frac{1}{2}\sqrt{\sigma_t^2 + 4\,\tau^2}, \qquad N/mm^2$$

For safe design,

$$\sigma_{maxi} = \frac{\sigma_y}{fos}\ \&\ \tau_{maxi} = \frac{0.5 \times \sigma_y}{fos}, \qquad N/mm^2$$

By solving the above equations, stress area can be found out, then from PSGDB pg.no. 5.42, for the stress area A_c, mm^2, the bolts size can be selected.

2. Eccentric loading in same plane (parallel to the bolts):



The above figure shows the eccentric loading of bolts in plane of the bolts. That is the eccentric load is acting parallel to the bolts.

P	- Eccentric load, N
e	- Eccentric distance, mm
x & y	- Distance from the C.G point of the bolts in the x and y direction, mm
$n = n_1 + n_2$	- no of bolts
r	- radius of the bolt center from the C.G point, mm

$$r = \sqrt{x^2 + y^2}, \qquad mm$$

The bolts are subjected to direct shear stress due to the eccentric loading, P and secondary shear stress due to secondary shear force, F

i. Direct or primary shear stress, τ_1

$$\tau_1 = \frac{P}{A_c}, \qquad N/mm^2$$

ii. Secondary shear stress, τ_2

$$\tau_2 = \frac{F}{A_c}, \qquad N/mm^2$$

Where,

$$F = \frac{P \times e}{n \times r}, \qquad N$$

If all the bolts are having same radii, for bolts of different radii,

$$F_1 = \frac{P \times e \times r_1}{r_1^2 + r_2^2 + r_3^2 + \cdots}, \qquad F_2 = \frac{P \times e \times r_2}{r_1^2 + r_2^2 + r_3^2 + \cdots}, \quad etc.$$

Maximum of the above will be considered for the design.

Since there are two shear stresses acting on the bolts, it is required to find resultant shear stress,

$$\tau_r = \sqrt{\tau_1^2 + \tau_2^2 + 2\,\tau_1\,\tau_2 \cos\theta}, \qquad N/mm^2$$

Where,

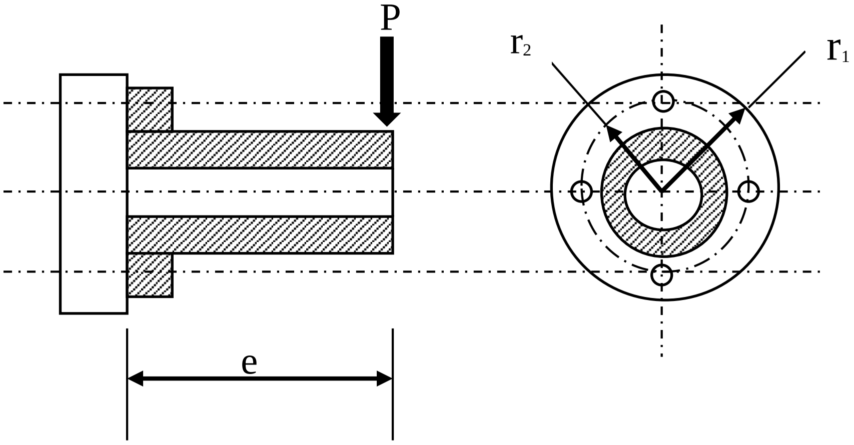
$$\theta = \tan^{-1}\left(\frac{y}{x}\right)$$

For safe design,

$$\tau_r = \tau_{all} = \frac{0.5 \times \sigma_y}{fos}, \qquad N/mm^2$$

By solving the above equations, stress area can be found out, then from PSGDB pg.no. 5.42, for the stress area A_c, mm^2, the bolts size can be selected.

Design of bolts in circular base:



The above figure shows the loading of bolts in circular base.

P - Eccentric load, N

e - Eccentric distance, mm

r_1 - radius of the flange, mm

r_2 - radius of the pitch circle of the bolts, mm

n - no of bolts

θ - angular distribution of the bolts in the circular base,

When the bolts are arranged in circular pattern for joining to a circular base, the force acting on the bolt situated in position 1, is given by

$$F_1 = \frac{P \times e \times (r_1 - r_2\, cos\theta)}{4\, r_1^2 + 2\, r_2^2} \; and\ so\ on\ for\ other\ bolts$$

when the direction of the load P changes with relation to the bolts as in the case of pillar crane. Then, the maximum force acting on the 'n' no of bolts is given as,

$$F_{maxi} = \frac{2 \times P \times e\,(r_1 + r_2)}{n\,(2\,r_1^2 + \,r_2^2)}, \qquad N$$

When the direction of load P does not change with relation to bolts, then

$$F_{maxi} = \frac{2 \times P \times e\,\left[r_1 + r_2 \cos\left(\frac{180}{n}\right)\right]}{n\,(2\,r_1^2 + \,r_2^2)}, \qquad N$$

Due to this force, the bolts are subjected to tensile stress and it is given by

$$\sigma_t = \frac{F_{maxi}}{A_c}, \qquad N/mm^2$$

For safe design,

$$\sigma_t = \frac{\sigma_y}{fos}, \qquad N/mm^2$$

By solving the above equations, stress area can be found out, then from PSGDB pg.no. 5.42, for the stress area A_c, mm^2, the bolts size can be selected.

Introduction

Welding is a process for joining two similar or dissimilar metals by fusion and provides a permanent joint. In welding, the parts are coalesced at their contacting surfaces by a suitable application of heat and/or pressure, with or without the addition of a filler metal. Welding provides a permanent joint but it normally affects the metallurgy of the components. It is therefore usually accompanied by post weld heat treatment for most of the critical components. The welding is widely used as a fabrication and repairing process in industries. Some of the typical applications of welding include the fabrication of ships, pressure vessels, automobile bodies, bridges, welded pipes, sealing of nuclear fuel and explosives, etc.

Advantages

1. Welding is more economical and is much faster process as compared to other processes (riveting, bolting, casting etc.)
2. Welding, if properly controlled results permanent joints having strength equal or sometimes more than base metal.
3. Large number of metals and alloys both similar and dissimilar can be joined by welding.
4. General welding equipment is not very costly.
5. Portable welding equipment's can be easily made available.
6. Welding permits considerable freedom in design.
7. Welding can join welding jobs through spots, as continuous pressure tight seams, end-to-end and in a number of other configurations.
8. Welding can also be mechanized.

Disadvantages

1. It results in residual stresses and distortion of the work pieces.
2. Welded joint needs stress relieving and heat treatment.
3. Welding gives out harmful radiations (light), fumes and spatter.
4. Jigs and fixtures may also be needed to hold and position the parts to be welded
5. Edges preparation of the welding jobs are required before welding
6. Skilled welder is required for production of good welding
7. Heat during welding produces metallurgical changes as the structure of the welded joint is not same as that of the parent metal.

Types of Welding

Welding processes can be broadly classified in two groups: fusion welding and solid-state welding.

Fusion Welding Processes

Fusion Welding processes use heat to melt the base metals. In fusion welding operations, a filler metal is generally added to the molten pool. Fusion welding processes can further be subdivided into following types:

Arc Welding: Arc welding refers to a group of welding processes in which heating of the metals is accomplished by an electric arc.

Resistance welding: Resistance welding achieves coalescence using heat from electrical resistance to the flow of a current passing between the faying surfaces of two parts held together under pressure.

Oxyfuel Gas Welding: These joining processes use an oxyfuel gas, such as a mixture of oxygen and acetylene, to produce a hot flame for melting the base metal.

Other welding processes that produce fusion of the metals joined include electron beam welding and laser beam welding.

Solid-State Welding

Solid-state welding refers to joining processes in which coalescence results from application of pressure alone or a combination of heat and pressure. If heat is used, the temperature in the process is below the melting point of the metals being welded. No filler metal is utilized. Some welding processes in this group are:

Diffusion welding: Two surfaces are held together under pressure at an elevated temperature and the parts coalesce by solid-state fusion.

Friction welding: Coalescence is achieved by the heat of friction between two surfaces.

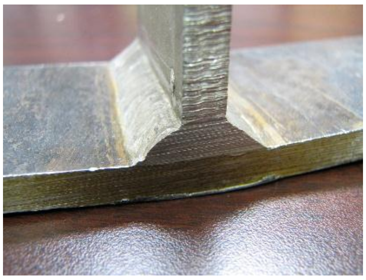
Ultrasonic welding: Moderate pressure is applied between the two parts and an oscillating motion at ultrasonic frequencies is used in a direction parallel to the contacting surfaces. The combination of normal and vibratory forces results in shear stresses that remove surface films and achieve atomic bonding of the surfaces.

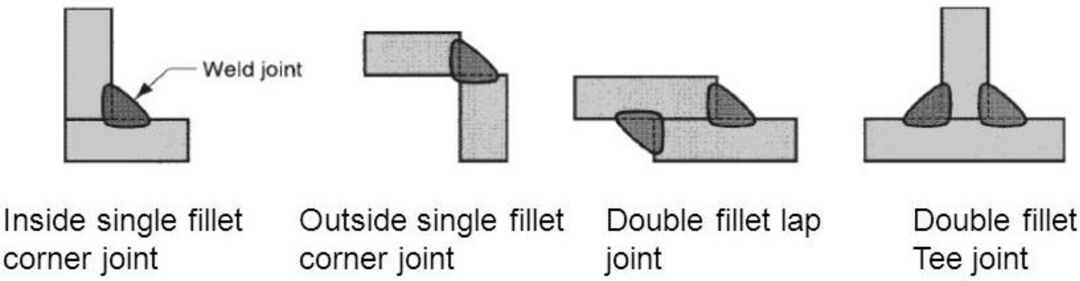
Types of Welded Joints

Welded joints are primarily of two types: 1. Lap joint or fillet joint, and 2. Butt joint.

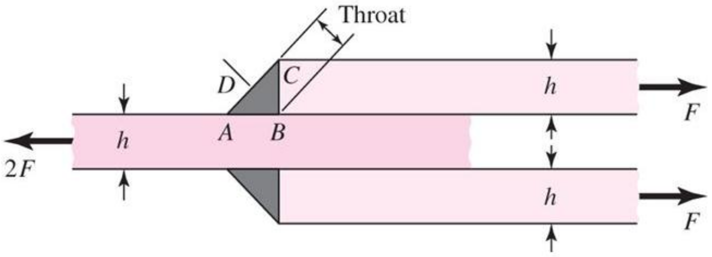
Lap Joint

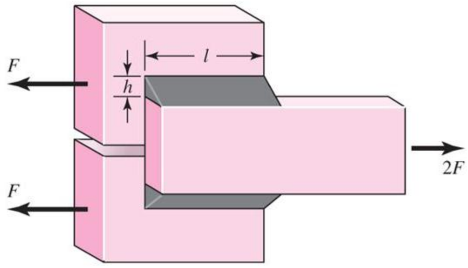
The lap joint or the fillet joint is obtained by overlapping the plates and then welding the edges of the plates. The cross-section of the fillet is approximately triangular. The fillet joints are of three types: - Single transverse fillet, Double transverse fillet and Parallel fillet joints. These are shown in Figure 10.1.





Fillet welds



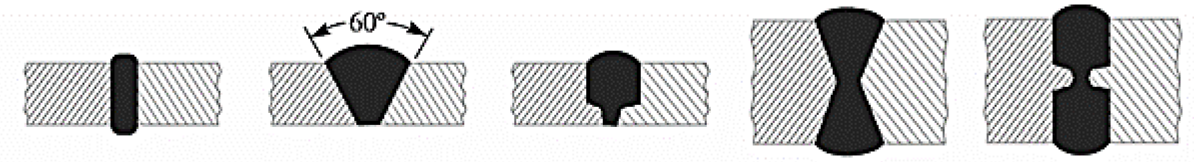


Types of

Lap or Fillet Joint

Butt Joint

The butt joint is obtained by placing the plates edge to edge as shown in figure. In butt welds, the plate edges do not require bevelling if the thickness of plate is less than 5 mm. On the other hand, if the plate thickness is 5 mm to 12.5 mm, the edges should be bevelled to V or U-groove on both sides. Figure 10.2 shows the types of butt joints.



(*a*) Square butt joint. (*b*) Single *V*-butt joint. (*c*) Single *U*-butt joint. (*d*) Double *V*-butt joint. (*e*) Double *U*-butt joint.

Types of Butt Joint

Corner joint, edge joint and T-joint are some other types of welded joints.

(a) Corner joint (b) Edge joint (c) T - joint

Some Other Types of Welded Joints

Welded joints:

It is defined as the process of joining two metallic parts heating them to a suitable temperature with or without application of pressure.

Advantages and Disadvantages:

- It forms a stronger and lighter joint
- The cost of welding is less
- It forms a tight and leak proof joint
- The welded part looks smooth and pleasant when compared riveted joints.
- It is highly subjected to damped vibrations which leads to thermal distortion resulting in residual stresses.
- Heat treatment is required to relieve the residual stresses
- The strength of the weld depends on the skill of the labour.
- The inspection cost of welding is high.

Welding process:

It is broadly classified into two types:

1. welding process that use heat alone to join the two parts. Example: Thermit welding, gas welding, electric arc welding

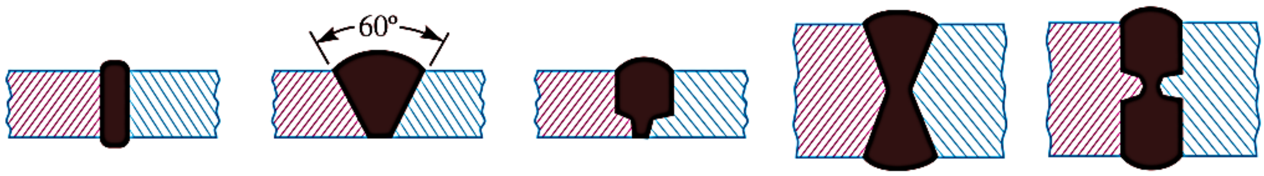
2. welding process that use combination of heat and pressure to join the two parts. Example: forge welding, electric resistance welding, friction stir welding

Types of welded joints:

(1) Butt joint:

It is defined as the joint between two components lying approximately in the same plane. This kind of joints are used in applications like pressure vessels and cylinders.

Types of butt joints:



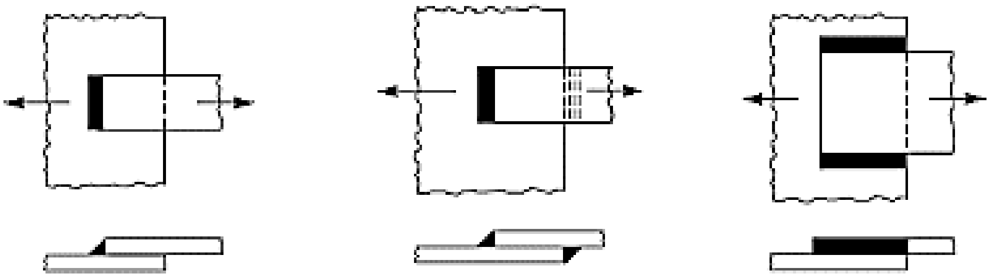
(*a*) Square butt joint. (*b*) Single *V*-butt joint. (*c*) Single *U*-butt joint. (*d*) Double *V*-butt joint. (*e*) Double *U*-butt joint.

(2) Fillet joint:

It is defined as the joint between two overlapping plates or components. They are also called as lap joint. A fillet weld has an approximate triangular cross section joining the two surfaces at right angles to each other.

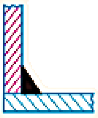
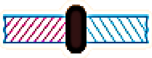
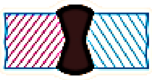
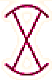
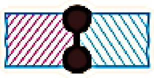
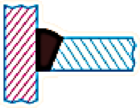
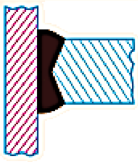
Types of fillet joints:

a. Transverse fillet weld – (single and double)
b. Parallel fillet weld – (single and double)



(*a*) Single transverse. (*b*) Double transverse. (*c*) Parallel fillet.

Weld symbols:

S. No.	Form of weld	Sectional representation	Symbol
1.	Fillet		
2.	Square butt		
3.	Single-*V* butt		
4.	Double-*V* butt		
5.	Single-*U* butt		
6.	Double-*U* butt		
7.	Single bevel butt		
8.	Double bevel butt		

Design of single V and Double V butt joint:

When plates welded with single and double V butt joint subjected to tensile load, results in tensile stress in the weld cross section.

$$\sigma_t = \frac{P}{A_w}, \qquad N/mm^2$$

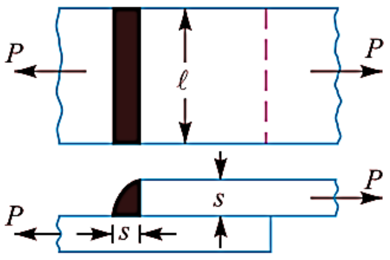
Where, A_w is the cross-sectional area of the weld,

$$for\ single\ V\ butt\ joint, \qquad A_w = l \times h, \qquad mm^2$$

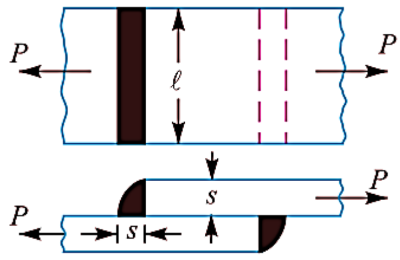
$$for\ double\ V\ butt\ joint, \qquad A_w = l \times \frac{h}{2}, \qquad mm^2$$

Where, l = length of weld, h = thickness of the weld (equal to thickness of the plate being welded)

Design of Transverse fillet weld:



(*a*) Single transverse fillet weld.



(*b*) Double transverse fillet weld.

The transverse fillet weld subjected to axial loading as shown in the figure, will be subjected to tensile stress and it is given by

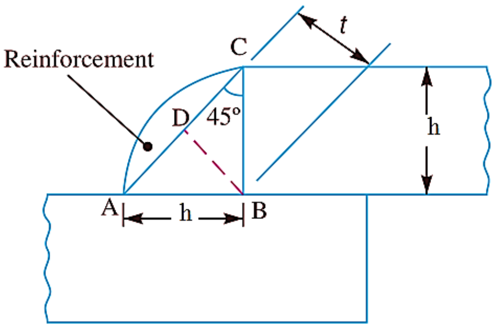
$$\sigma_t = \frac{P}{A_w}, \qquad N/mm^2$$

The area of the weld, A_w is given by

$$A_w = l \times t, \qquad mm^2$$

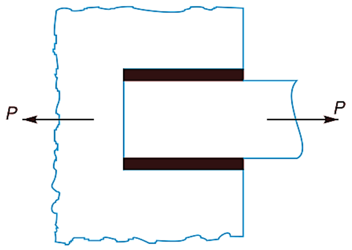
Where, l = length of the weld, t = throat thickness, mm

$$t = 0.707\, h, \qquad mm$$



, h = thickness of the plate, mm

Design of parallel fillet weld:



Double parallel fillet weld.

The parallel fillet weld subjected to axial loading as shown in the figure results in shear stress and it is given by

$$\tau = \frac{P}{A_w}, \qquad N/mm^2$$

The area of the weld, A_w is given by

$$A_w = l \times t, \qquad mm^2$$

Where, l = length of the weld, t = throat thickness, mm

$$t = 0.707\ h, \qquad mm$$

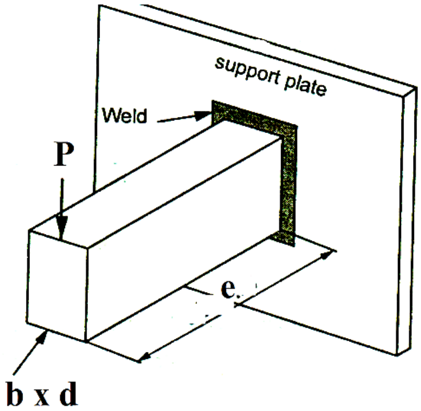
Design of welded joints under fatigue loading:

For above types of welded joints when subjected to fatigue loading, the stress concentration factor is to be considered while calculating the permissible stress of the weld section. The following table shows the stress concentration factor values for different types of weld.

Type of joint	*Stress concentration factor*
1. Reinforced butt welds	1.2
2. Toe of transverse fillet welds	1.5
3. End of parallel fillet weld	2.7
4. T-butt joint with sharp corner	2.0

Design of welded joints subjected to eccentric loading:

1. welded joints subjected to moment in the plane of the weld:



Such welded joints are subjected to

(i) Direct shear stress, τ

$$\tau = \frac{P}{A_w}, \qquad N/mm^2$$

Where, area of the weld A_w is given by

$$A_w = l \times t, \qquad mm^2$$

Where, throat thickness, $t = 0.707\ h,\ mm$

(ii) Bending stress, σ_b

$$\sigma_b = \frac{M_b \times y}{I_w} = \frac{M_b}{I_w/y} = \frac{M_b}{z_w}, \qquad N/mm^2$$

Where, bending moment, $M_b = P \times e\ ,\ N.mm$; Sectional modulus, Z_w is taken from PSGDB pg.no. 11.5 & 11.6.

According to maximum principal stress theory and shear stress theory,

$$\sigma_{maxi} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\,\tau^2}\right], \qquad N/mm^2$$

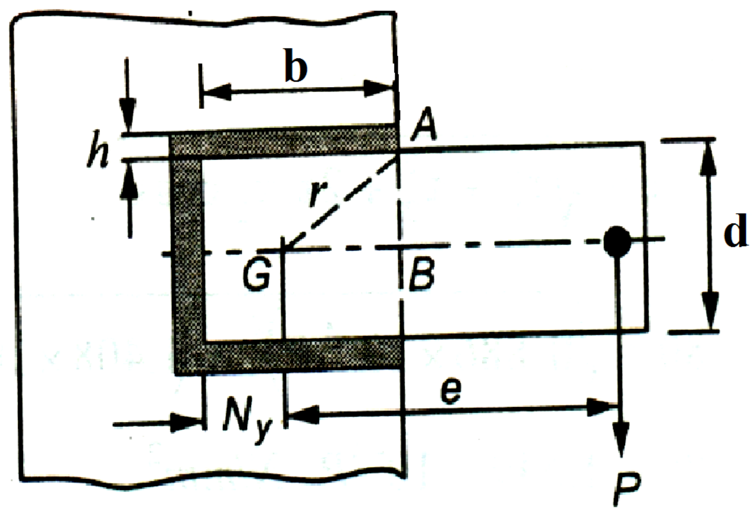
$$\tau_{maxi} = \frac{1}{2}\sqrt{\sigma_b^2 + 4\,\tau^2}, \qquad N/mm^2$$

For safe design,

$$\sigma_{maxi} = \frac{\sigma_Y}{fos}\ and\ \tau_{maxi} = \frac{0.5\ \sigma_Y}{fos}$$

Where, σ_Y is the yield value of the weld material in N/mm^2

2. welded connection subjected to torsion in the plane of the weld:



The welded joint shown in the figure is subjected to

i. Direct shear stress, τ_1

$$\tau_1 = \frac{P}{A_w}, \qquad N/mm^2$$

Where, area of the weld A_w is given by

$$A_w = l \times t, \qquad mm^2$$

Where, throat thickness, $t = 0.707\ h,\ mm$, length of the weld, $l = (2\,b + d)\ ,mm$

ii. Torsional shear stress or secondary shear stress, τ_2

$$\tau_2 = \frac{T \times r}{J_w} = \frac{P \times e \times r}{J_w}, \qquad N/mm^2$$

Where, J_w is the polar moment of inertia of the weld section, it can be found using the relations provided in PSGDB pg.no. 11.5 & 11.6

Where, the radius of rotation, $r = \sqrt{GB^2 + AB^2}, mm$

The resultant shear stress for the primary and secondary shear stress based on vector approach is given by,

$$\tau_R = \sqrt{\tau_1^2 + \tau_2^2 + 2\,\tau_1\,\tau_2 \cos\theta}\ , \qquad N/mm^2$$

For safe design,

$$\tau_R = \frac{0.5 \times \sigma_Y}{fos}, \qquad N/mm^2$$

Problems:

1. A plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of two parallel fillet welds. The plates are subjected to a load of 50 kN. Find the length of the weld so that the maximum stress does not exceed 56 MPa. Consider the joint first under static loading and then under fatigue loading.

Given Data:

h = 12.5 mm; P = 50 x 10^3 N, τ = 56 N/mm^2

To find:

$l = ?$

Solution:

Case (i) static loading

w.k.t the parallel fillet weld is subjected to shear stress, τ

$$\tau = \frac{P}{A_w}, \qquad N/mm^2$$

The area of the weld,

$$A_w = l \times t = l \times 0.707\, h = l \times 0.707 \times 12.5$$

$$A_w = 8.84\, l \;\; mm^2$$

The length of the weld, l

$$\tau = \frac{P}{8.84\, l}$$

$$56 = \frac{50 \times 10^3}{8.84\, l} = \frac{5656.1}{l}$$

$$l = \frac{5656.1}{56} = 101\ mm$$

Where, $l = 101\ mm$ is the total length of the weld, for one parallel fillet weld,

$$b = \frac{l}{2} = \frac{101}{2} = 50.5\, mm$$

Case 2: for fatigue loading

$$\tau = \frac{56}{stress\ concentration\ factor} = \frac{56}{2.7} = 20.74\ N/mm^2$$

The length of the weld, l

$$\tau = \frac{P}{8.84\, l}$$

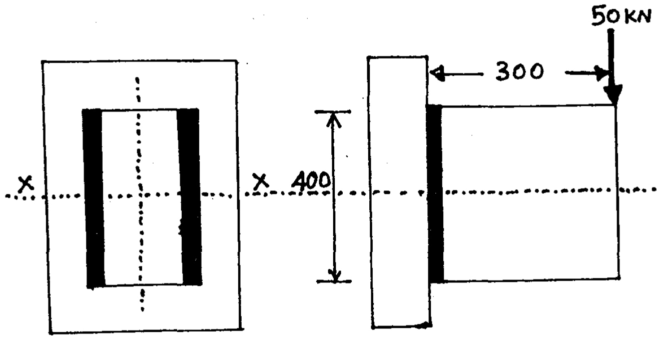
$$20.74 = \frac{50 \times 10^3}{8.84\, l} = \frac{5656.1}{l}$$

$$l = \frac{5656.1}{20.74} = 272.7 \approx 273\ mm$$

Where, $l = 101\ mm$ is the total length of the weld, for one parallel fillet weld,

$$b = \frac{l}{2} = \frac{273}{2} = 136.5\ mm$$

2. A bracket is welded to the vertical plate by means of two fillet welds as shown in figure. Determine the size of the welds, if the permissible shear stress is limited to 80 N/mm^2.



NOTE : DIMENSIONS ARE IN 'mm'

Given data:

$\tau = 80\ N/mm^2$

From figure, $P = 50 \times 10^3\ N; e = 300\ mm,\ d = 400\ mm$

To find: size of the weld, h =?

Solution:

The given weld section is subjected to

(i) Direct shear stress, τ

$$\tau = \frac{P}{A_w}, \qquad N/mm^2$$

Where, area of the weld A_w is given by

Where, throat thickness, $t = 0.707\ h,\ \ mm$

$$A_w = l \times t = 2\,d \times 0.707\ h = 2 \times 400 \times 0.707 \times h$$

$$\boldsymbol{A_w = 565.5\ h\ mm^2}$$

(ii) Bending stress, σ_b

$$\sigma_b = \frac{M_b}{z_w} \quad N/mm^2$$

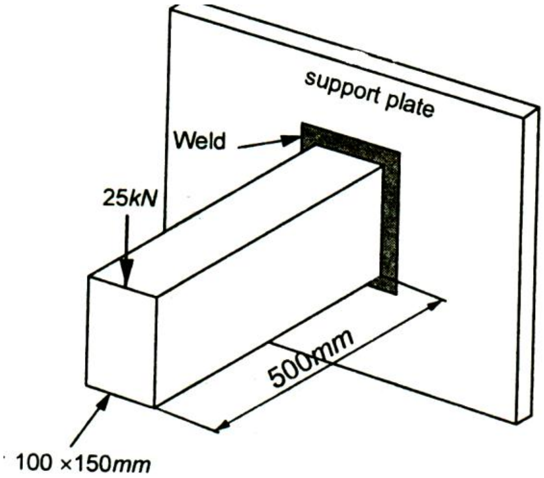
Where, bending moment, $M_b = P \times e\ ,\ N.mm$; Sectional modulus, Z_w is taken from PSGDB pg.no. 11.5 & 11.6.

According to maximum principal stress theory and shear stress theory,

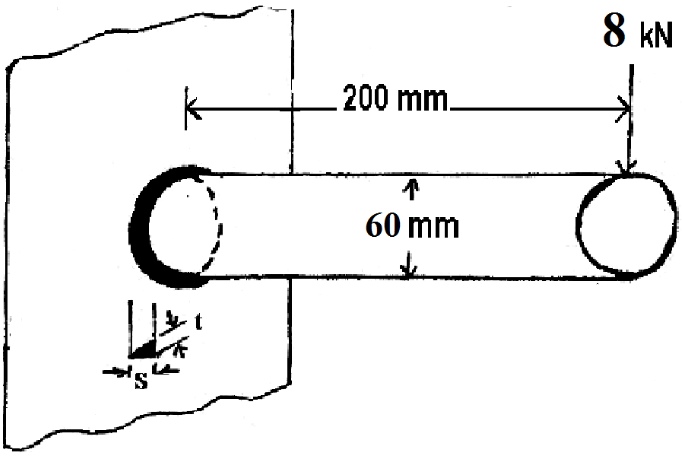
$$\sigma_{maxi} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b^2 + 4\,\tau^2}\right], \qquad N/mm^2$$

$$\tau_{maxi} = \frac{1}{2}\sqrt{\sigma_b^2 + 4\,\tau^2}, \qquad N/mm^2$$

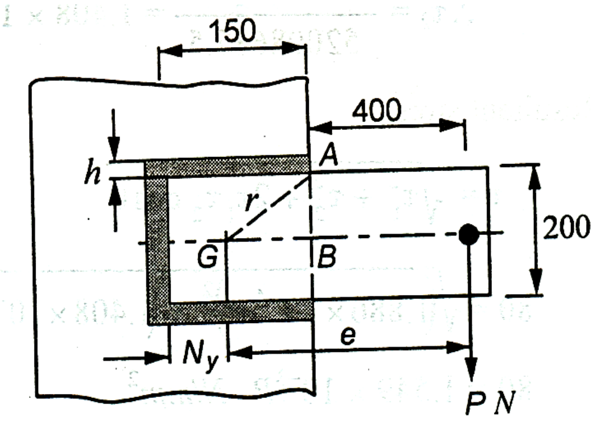
3. A shaft of rectangular cross section is welded to a support plate by means of fillet weld on its one end as shown in figure. The other end is loaded by 25 kN. If the size of weld is 6 mm, find the maximum normal and shear stress in the weld.



4. A 60 mm diameter solid shaft is welded to a flat plate as shown in Figure. If the size of the weld is 20 mm, find the maximum normal and shear stress in the weld.



5. A bracket shown in the figure carries a load of 'P' kN. Calculate the value of P, if the weld size is 25 mm and the allowable stress is not to exceed 120 N/mm^2.



MULTIPLE CHOICE QUESTIONS

S.No	QUESTIONS	Opt 1	Opt 2	Opt 3	Opt 4	Answer
1	In a fusion welding process	Only heat is used	Only pressure is used	Combunation of heat and pressure is used	None of these	Only heat is used
2	The electric arc welding is a type of__________ welding	Forge	Fusion	Thermite welding	Gas welding	Fusion
3	The principle of applying heat and pressure is widely used in	Spot welding	Seam welding	Projection welding	All of these	All of these
4	In transverse fillet welded joint, the size of the weld is equal to	0.5 X Throat of the weld	Throat of the weld	√2 X Throat of the weld	2 X throat of the weld	√2 X Throat of the weld
5	The transverse fillet welded joints are designed for	Tensile strength	Compressive strength	Bending strength	Shear strength	Tensile strength
6	The parallel fillet welded joints is designed for	Tensile strength	Compressive strength	Bending strength	Shear strength	Shear strength
7	The size of the weld in butt welded joints is equal to	0.5 X Throat of the weld	Throat of the weld	√2 X Throat of the weld	2 X throat of the weld	Throat of the weld
8	A double fillet welded joint with parallel fillet weld of lemgth l and leg s is subjected to the tensile force p. Assuming uniform stress distribution, the shear stress in weld is given by	(√2 p)/(s.l)	p/(2s.l)	p/(√2s.l)	(2p)/(s.l)	p/(√2s.l)
9	When a circular rod welded to a rigid plate by a circular fillet weld is subjected to a twisting moment T, then the maximum shear stress is given by	(2.83T)/(π sd2)	(4.242T)/(π sd2)	(5.66T)/(π sd2)	None of these	(2.83T)/(π sd2)
10	For a parallel load on a fillet weld of equal legs, the plane of maximum shear occurs at	22.5°	30°	45°	60°	45°
11	The largest diameter of the external or internal diameter screw thread is known as	Minor diameter	Major diameter	Pitch diameter	None of these	Major diameter

12	The pitch diameter is the __________ diameter of an external or internal screw thread	Effective	Smallest	Largest	Medium	Effective
13	A screw is specified by its	Major diameter	Minor diameter	Pitch diameter	Pitch	Major diameter
14	The railway carriage coupling have	Square threads	Acme threads	Knuckle threads	Buttress threads	Buttress threads
15	The square threads are usually found on	Spindles of bench vise	Railway carriage couplings	Feed mechanism of machine tools	Screw cutting lathes	Feed mechanism of machine tools
16	A locking device in which the bottom cylindrical portion is recessed to receive the tip of the locking set screw, is called	Castle nut	Jam nut	Ring nut	Screw nut	Ring nut
17	Which one is not a positive locking device?	Spring washer	Cotter pin	Tongued washer	Spring wire lock	Spring washer
18	The washer is generally specified by its	Outer diameter	Hole diameter	Thickness	Mean diameter	Hole diameter
19	A locking device extensively used in automobile is a	Jam nut	Castle nut	Screw nut	Ring nut	Castle nut
20	A bolt of M24 X 2 means that	Pitch of the thread is 24mm and depth is 2mm	The cross sectional area of the thread is 24mm2	Nominal diameter of the bolt is 24mm and pitch 2mm	Effective diameter of the bolt is 24mm and there are two threads per cm	Nominal diameter of the bolt is 24mm and pitch 2mm
21	When a nut is tigntened by placing a washer below it, the bolt will be subjected to	Tensile stress	Compressive stress	shear stress	None of these	Tensile stress
22	The eye bolts are used for	Transmission of power	Locking devices	Lifting and transporting heavy machines	Absorbing shocks and vibrations	Lifting and transporting heavy machines
23	The shock absorbing capacity of the bolt may be increased by	Increasing its shank diameter	Increasing its length	Tightening the bolt properly	Making the shank diameter equal to the core diaketer of the thread	Increasing its length
24	The resilense of a bolt may be increased by	Increasing its shank diameter	Increasing its length	Decreasing its shank diameter	Decreasing its length	Increasing its length
25	Design for Manufacture and assemble suggests _____ number of threaded fasteners	Maximum	minimum	equal	None of these	minimum
26	A bolt of uniform strength can be developed by	Keeping the core diameter of threads equal to the diameter of unthreaded portion of the bolt	Keeping the core diameter of threads smaller than the diameter of unthreaded portion of the bolt	Keeping the nominal diameter of threadsequal to the diameter of unthreaded portion of bolt	None of these	Keeping the core diameter of threads equal to the diameter of unthreaded portion of the bolt
27	_____ bolt has rough shank	Machine bolt	eyebolt	automobile bolts	carriage bolts	Machine bolt
28	A bolt	Has a head on one end and	Has head at one end and other end fits	Has both ends threaded	Is provided with	Has a head on one end and

		nut fitted to other	in to a tapped hole in the other part to be joined		poointed threads	nut fitted to other
29	_______ cap screw has cylindrical head with a slot for the screw driver	button head	Fillister head	Hexagonal head	Socket head	Fillister head
30	______ is used to prevent the relative motion between two parts	Setscrew	Capscrew	Studs	Tap bolts	Setscrew
31	The normal height of a hexagonal or square head of bolt is taken as _______ of the major diameter of the thread of screw	0.8	1.5	0.7	0.5	0.7
32	The crest diameter of the screw thread is same as	Major diameter	Minor diameter	Pitch diameter	Core diameter	Major diameter
33	If the diameter of the bolt hole then for a flanged pipe loint to be leak proof, the circulferential pitch of the bolts should be	10√d	10√d tp 15√d	15√d tp 20√d	20√d tp 30√d	20√d tp 30√d
34	The included angle in unified of American National threads is	60°	55°	47.5°	29°	60°
35	The included angle in unified of Acme threads is	60°	55°	47.5°	29°	29°
36	An allen bolt is	self locking bolt	Same as stud	Provided with hexagonal depression in the head	Used in high speed componenets	Provided with hexagonal depression in the head
37	A self locking screw has	Fine threads	Coarse threads	coefficient friction ≥Tangent of load angle	Hole for inserting split pin	coefficient friction ≥Tangent of load angle
38	Machine screws are	Similar to small size tap bolts except that a greater variety of shapes of heads are available	slotted for a screw driver and generally used with a nut	Similar to stud	None of these	slotted for a screw driver and generally used with a nut
39	What is the stress concentration factor for ststic load?	1	1.01	1.11	1.01	1
40	A screw made by cutting a single helical groove on the cylinder is known as _______	Single threaded screw	Double threaded screw	Right hand thread screw	Left hand thread screw	Single threaded screw
41	The smallest diameter of an external or internal screw is known as _________	Minor diameter	Major diameter	Pitch diameter	None of these	Minor diameter
42	The top surface of thread________	Root	crest	Slope	Pitch	crest
43	The distance from a point on one thread to corresponding point on next next is ______	Slope	crest	Pitch	Root	Pitch
44	B.S.W stands for___________	British Standard Withworth	British Standard Work	Britain Standard Withworth	Britian Standard Work	British Standard Withworth
45	ANST stands for ___________	American National	American National	America's National	America's National	American National

		society of thread	Standard thread	Standard Thread	Standard Thread	Standard thread
46	light loaded bolts are made of ________	free cutting steel	alloy steel	high carbon steel	None of these	free cutting steel
47	_____ can be defined as a joint between two components lying approximately in same plane.	Butt joint	lap joint	fillet joint	All of these	Butt Joint
48	when the thickness of the weld is more than 20 mm_______ welded joint is used	square butt joint	V - butt joint	Double V - butt joint	U - butt joint	U - butt joint
49	The permissible shear stress for fillet welds is taken as ____ N/mm^2 as per AWS	120	200	94	87	94
50	The permissible stress for a Butt weld in tension load is _____N/mm^2	110	120	140	125	110

UNIT IV- DESIGN OF SPRINGS AND FLYWHEEL

A spring is defined as an elastic machine element, which deflects under the action of load and returns to its original shape when the load is removed.

Functions of spring

A) Spring act as a flexible joint in between two parts or bodies

1. Cushioning, absorbing, or controlling of energy due to shock and vibration

Eg: Car springs or railway buffers

B) To control energy, springs-supports and vibration dampers.

2. Control of motion

C) Maintaining contact between two elements (cam and its follower)

D) Creation of the necessary pressure in a friction device (a brake or a clutch)

E) Restoration of a machine part to its normal position when the applied force is withdrawn (a governor or valve)

3. Measuring forces- Spring balances, gages

4. Storing of energy - In clocks or starters

Types of springs:

Classification based on the shape of the spring:

1. Helical spring
   a. Compression spring or open coil helical spring
   b. Tension spring or extension spring or closed coil helical spring
2. Torsion spring
3. Laminated leaf spring
4. Spiral spring
5. Disc or Belleville spring

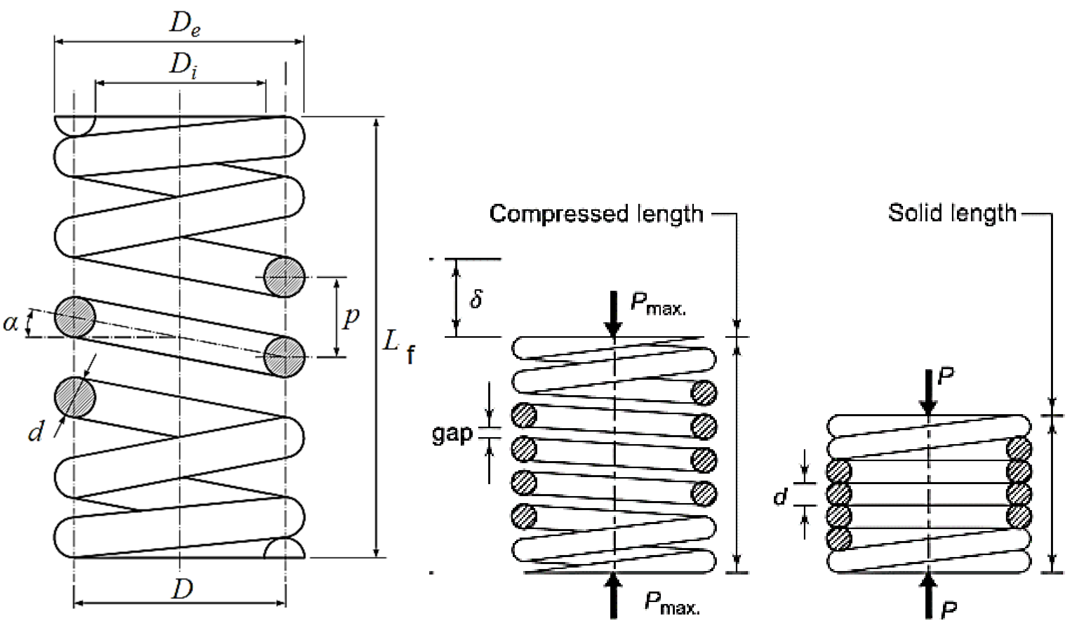


Helical compression spring **Laminated leaf spring** **Torsion spring**

Belleville spring

Spiral spring

Terminologies of Helical compression spring:



Where,

D – Mean coil diameter

d – wire diameter

D_i & D_e – Inner and outer diameter of the coil

1. Spring index (C):

It is defined as the ratio of mean coil diameter to the wire diameter

$$C = \frac{D}{d}$$

This value ranges from 4 to 12

2. Solid length:

It is the axial length of ths spring, when the spring is compressed in such a way that the adjacent coils touch each other.

$$L_s = n_t\, d, \qquad mm$$

Where, n_t – total no of coils or turns

3. Free length:

It is the axial length of an unloaded spring

$$L_f = L_s + 0.15\, y_{maxi} + y_{maxi}$$

Where, y_{maxi} – maximum deflection of the spring

4. Pitch of the coil:

It is the axial distance between the adjacent coils in uncompressed state of the spring

$$p = \frac{L_f}{n_t - 1}, \qquad mm$$

5. Stiffness (q):

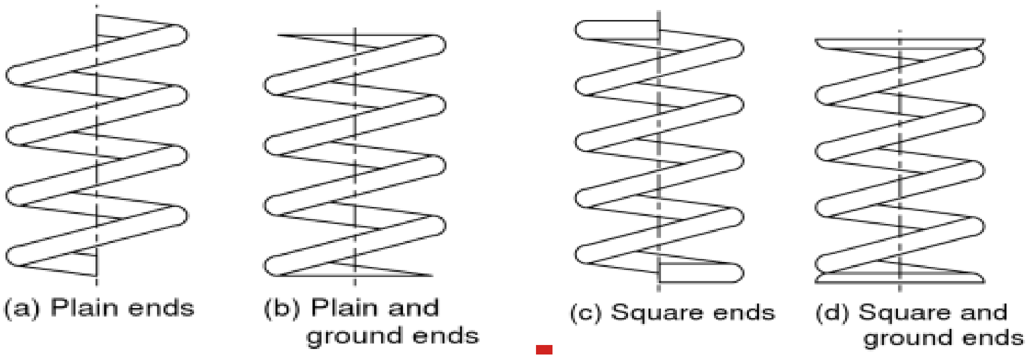
It is the ratio of the force and deflection, that is force per unit deflection.

$$q = \frac{P}{y}$$

6. Active coils (n) and inactive coils (n'):

The coils which contributes to the spring action are active coils and those which doesnot involve in the spring action are meant as inactive coils.

End condition of the spring:



Design of helical compression spring:

A helical compression spring subjected to axial force of 'P', the spring results in

1. Direct shear stress in the spring wire
2. Torsion shear stress due to torsional moment

The equivalent of these two stresses is given by

$$\tau = k_s \frac{8\,P\,D}{\pi\,d^3} = k_s \frac{8\,P\,C}{\pi\,d^2}, \qquad N/mm^2$$

The axial deflection of the spring is given by

$$y = \frac{8\,P\,D^3 n}{G\,d^4} = \frac{8\,P\,C^3\,n}{G\,d}, \qquad mm$$

Where, G – modulus of rigidity, n – no of active coils

The strain energy stored by the spring is given as

$$U = \frac{1}{2}\,P\,y\,, \qquad joules$$

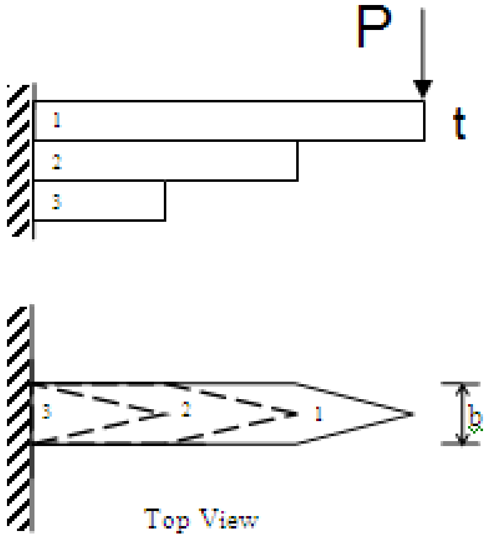
Leaf spring:

Simply supported beam and cantilever beam maybe used as springs because under certain amount of load, these beams get deflected and thus absorbs energy. These types of springs are known as leaf spring or flat spring.

Types of leaf spring:

1. Laminated leaf spring
   a. Cantilever types
   b. Simply supported type
2. Semi elliptic leaf spring

Design of Laminated cantilever leaf spring



Laminated leaf spring

The leaf springs are subjected to bending stress and it is given by

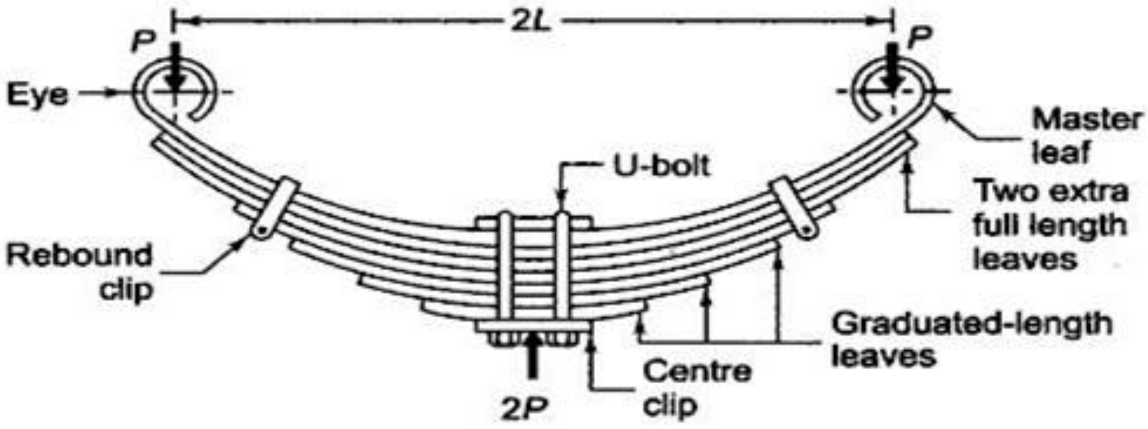
$$\sigma_b = \frac{6\,P\,L}{b\,t^2}, \qquad N/mm^2$$

And the deflection of the spring is given by

$$y = \frac{6\,P\,L^3}{E\,n\,b\,t^3}, \qquad mm$$

Refer PSGDB pg.no 7.104

Design of semi elliptic leaf spring:



Semi elliptic leaf spring

The semi elliptic leaf spring consists of two types of leaves

1. Extra full length / master leaves (n_e)
2. Graduated leaves (n_g)

This type of spring is also subjected to bending stress and it is given by

$$\sigma_{bg} = \frac{12\,P\,L}{b\,t^2\,(3\,n_e + 2\,n_g)}, \qquad N/mm^2$$

$$\sigma_{be} = \frac{18\,P\,L}{b\,t^2\,(3\,n_e + 2\,n_g)}, \qquad N/mm^2$$

The deflection of the spring is given by

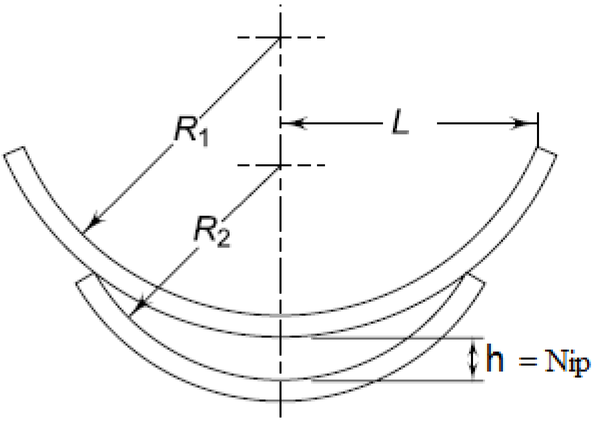
$$y = \frac{12\,P\,L^3}{E\,b\,t^3\,(3\,n_e + 2\,n_g)}, \qquad mm$$

Nipping:

It is the process of prestressing the leaf springs by providing different radii of curvature for full length and graduated leaves.

Nip: it is the initial gap between the leaves which is adjusted under maximum load condition so that stresses in all leaves are same.

$$h = \frac{2\,P\,L^3}{E\,n\,b\,t^3}, \qquad mm$$



The amount of load required to close the initial gap while assembly of the leaves by means of U-bolts is given by

$$P_b = \frac{2\,P\,n_e\,n_g}{n\left(3\,n_e + 2\,n_g\right)}, \qquad N$$

The effective length of the spring for semi elliptic type is given by

$$2L = length\,of\;the\;spring - width\,of\;the\;central\;band$$

Problems:

1. Design a spring for spring loaded safety valve for the following condition:

Operating pressure	= 1.2 MPa
Diameter of the valve seat	= 100 mm
Design shear stress for the spring	= 380 MPa.
Shear modulus ' G '	= 84 GPa.

The spring is to be kept in the casing of 130 mm inner diameter and 400 mm long. The spring should be at maximum lift of 6 mm when the pressure is 1.3 MPa.

2. A spring-loaded safety valve for a boiler is required to blow-off at a pressure of 1.5 N/mm^2. The diameter of the valve is 100 mm. Design a suitable compression spring for the safety valve, if spring index is 6 and initial compression is 25 mm. The maximum lift of the valve is 15 mm. The shear stress in the spring material is to be limited to 450 MPa. Take G = 0.84 GPa.

3. A rail wagon of mass 8 tones is moving with a velocity of 2 m/s. It is brought to rest by two buffers with springs of 250 mm diameter. The maximum deflection of springs is 200 mm. The allowable shear stress in the spring material is 600 MPa. Design the spring for the buffers.

4. Design a Leaf spring for a truck to the following specifications.

Maximum load on springs	= 150 kN
Number of springs	= 4
Spring material	= chromium vanadium steel
Permissible tensile stress	= 600 MPa
Maximum number of leaves	= 10
Span of the spring	= 1200mm

Permissible deflection = 80 mm
Young's Modulus for the material = 210 GPa.

5. A truck spring has 15 number of leaves, two of which are full lengthy leaves. The spring supports are 1.5 m apart and the central band is 100 mm wide. The central load is to be 8 kN with a permissible stress of 300 MPa. Determine the thickness and width of the steel spring leaves. The ratio of the total depth to the width of the spring is 3. Also determine the deflection of the spring.

6. A semi elliptic leaf spring is of 5 m long and is required to resist a load of 60 kN. The spring has 15 leaves, of which three are full length leaves. The width of central band is 100 mm. All the leaves are to be stressed to 440 MPa. The ratio of total depth to width is 3. Take E = 2 x 10^5 MPa. Determine
   a. The thickness and width of the leaves.
   b. The initial gap that should be provided between the full lengths and graduated leaves before assembly.
   c. The load exerted on the band for the assembly.

MULTIPLE CHOICE QUESTIONS:

S.No	Questions	Option A	Option B	Option C	Option D	Answer
1	A spring used to absorb shocks and vibrations is	Closely-coiled helical spring	Open-coiled helical spring	Conical spring	Leaf spring	
2	The spring mostly used in gramophones is	Helical spring	Conical spring	Laminated spring	Flat spiral spring	Flat spiral spring
3	Which of the following spring is used in a mechanical wrist watch ?	Helical compression spring	Spiral spring	Torsion spring	Bellevile spring	Torsion spring
4	When a helical compression spring is subjected to an axial compressive load, the stress induced in the wire is	Tensile stress	Compressive stress	Shear stress	Bending stress	Shear stress
5	In a close coiled helical spring, the spring index is given by D/d where D and d are the mean coil diameter and wire diameter respectively. For considering the effect of curvature, the Wahl's stress factor K is given by	(4C-1/4C+4) + (0.615/C)	(4C-1/4C-4) + (0.615/C)	(4C+1/4C-4) - (0.615/C)	(4C+1/4C+4) - (0.615/C)	(4C-1/4C-4) + (0.615/C)
6	When helical spring is cut into halves, the stiffness of the resulting spring will be	Same	Double	One-half	One-fourth	Double
7	Two close coiled helical springs with stiffness k1 and k2 respectively are connected in series. The stiffness of an equivalent spring is given by,	k1.k2/k1+k2	k1-k2/k1+k2	k1+k2/k1.k2	k1-k2/k1.k2	k1.k2/k1+k2
8	When two concentric coil springs made of the same material, having same length and compressed equally by an axial load, the load shared by the two springs will be ______ to the square of the diameters of the wires of the two springs.	Directly proportional	Inversely proportional	Equal to	None of these	Directly proportional

9	A leaf spring in automobiles is used	To apply forces	To measure forces	To absorb shocks	To store strain energy	To absorb shocks
10	In leaf springs, the longest leaf is known as	Lower leaf	Master leaf	Upper leaf	None of these	Master leaf
11	When the compression spring is compressed until the coils come in contact with each other, then the spring is said to be	Solid	Liquid	Spring	None of these	Solid
12	Leaf springs are also known as	Suspension springs	Flat spiral springs	Flat springs	None of these	Flat springs
13	Helical springs are not subjected to	Hoop stress	Force	Deflection	Torsional shear stress	Hoop stress
14	The spring rate of conical and volute springs, with increase in load	Remains constant	Decreases	Increases	Increases after the largest active coil starts to "bottom"	Increases after the largest active coil starts to "bottom"
15	The deflection of helical spring is directly and inversely proportional respectively to	D2, d2	D3, d2	D4, d3	D3, d4	D3, d4
16	Concentric helical springs should be	Wound in same direction	Wound with opposite hand helices	Could be wound in any direction	None of the above	Wound with opposite hand helices
17	Which is true statement about Belleville springs	These are used for dynamic loads	These are composed of coned discs which may be stacked up to obtain variety of load-deflection characteristics	These are commonly used in clocks and watches	These take up torsional loads	These are composed of coned discs which may be stacked up to obtain variety of load-deflection characteristics
18	Type of spring suited for space limitations and for providing variable stiffness.	Belleville spring	Helical spring	Conical helical spring	Conical spring	Conical helical spring
19	Type of spring for high compression capacity and to fit into small space	Belleville spring	Helical spring	Conical helical spring	Conical spring	Belleville spring
20	Springs are used for	Transfering energy	Generating energy	Storing energy	None of these	Storing energy
21	The cost index for a music wire based spring material is	3.5	2.5	1.5	0.5	3.5
22	The mimimum tensile strength for oil hardened and tempered spring steel wire diameter of 5 mm is ________	1440 Mpa	1480 Mpa	1520 Mpa	1250 Mpa	1440 Mpa
23	The factor of safety for springs based on torsional yield strength is taken as______	2.5	1.2	1.5	1	1.5
24	IS code used for Helical compression spring is _______	IS 7907 - 1975	IS 4454 - 1981	IS 7906 - 1975	IS 7906 - 1981	IS 7906 - 1975
25	The pulsating shear stress for patented and cold drawn steel wires is taken as-----	0.21 Sut	0.42 Sut	0.22 Sut	0.45 Sut	0.21 Sut
26	____ type of spring is used in door hinges, door locks etc.	Helical compression spring	Helical torsion spring	Helical tension spring	Helical concentric springs	Helical torsion spring
27	The spring index for helical torsion spring is taken as	5 to 10	5 to 8	5 to 15	5 to 12	5 to 15

28	______ spring is also known as power spring	Helical compression spring	Helical torsion spring	Spiral spring	Leaf spring	Spiral spring
29	Spiral springs are widely used in	Watches	cameras	instruments	All of these	All of these
30	The factor of safety for an automobile suspension based on yield strength is	1 to 1.5	2 to 2.5	3 to 3.5	0.5 to 1.5	2 to 2.5
31	The maximum fluctuation of speed is the	Difference of minimum fluctuation of speed and the mean speed	Difference of the maximum and minimum speeds	Sum of the maximum and minimum speeds	Variations of speed above and below the mean resisting torque line	Difference of the maximum and minimum speeds
32	The coefficient of fluctuation of speed is the___________of maximum fluctuation of speed and the mean speed.	Product	Ratio	Sum	Difference	Ratio
33	In a turning moment diagram, the variations of energy above and below the mean resisting torque line is called	Fluctuation of energy	Maximum fluctuation of energy	Coefficient of fluctuation of energy	None of these	Fluctuation of energy
34	If E= Mean kinetic energy of the flywheel, Cs = Coefficient of the fluctuation of speed and ΔE= Maximum fluctuation of energy, then	ΔE= E/Cs	ΔE= E2*Cs	ΔE= E*Cs	ΔE= 2E*Cs	ΔE= 2E*Cs
35	The ratio of the maximum fluctuation of the energy to the ______ is called coefficient of fluctuation of energy.	Minimum fluctuation of energy	Workdone per cycle	Maximum fluctuation of energy	None of these	Workdone per cycle
36	Due to the centrifugal force acting on the rim, the flywheel arms will be subjectes to	Tensile stress	Compressive stress	Shear stress	None of these	Tensile stress
37	The tensile stress in the flywheel rim due to the centrifugal force acting on the rim is given by	ρ.v2/4	ρ.v2/2	3ρ.v2/4	ρ.v2	ρ.v2
38	The cross-section of the flywheel arms is usually	Elliptical	Rectangular	I-section	L-section	Elliptical
39	In order to find the maximum bending moment on the arms, it is assumed as a	Simply supported beam carrying a uniformly distributed arm	Fixed at both ends and carrying a uniformly distributed load over the arm	Cantilever beam fixed at the hub and carrying a concentrated load at the free end of the rim.	None of these	Cantilever beam fixed at the hub and carrying a concentrated load at the free end of the rim.
40	The diameter of the hub of the flywheel is usually taken	Equal to the diameter of the shaft	Twice the diameter of the shaft	Three times the diameter of the shaft	Four times the diameter of the shaft	Twice the diameter of the shaft
41	The reciprocal of coefficient of fluctuation of speed is known as	Coefficient of steadiness	Fluctuation of energy	Coefficient of fluctuation of energy	None of these	Coefficient of steadiness
42	Tensile stress in flwheel arms is due to __________ acting on the rim	Radial force	Axial force	Centrifugal force	None of these	Centrifugal force
43	Bending stress in flywheel arms due to the ________ transmitted from the rim to the shaft or vice versa	Moment	Torque	Force	None of these	Torque

44	The fluctuation of energy may be determined by the ___________________ for one complete ycle of operation	Turning moment diagram	Bending moment diagram	Shear force diagram	None of these	Turning moment diagram
45	When flywheel absorbs energy its speed	Increases	Decreases	Remain constant	None of these	Increases
46	When flywheel gives up energy its speed	Increases	Decreases	Remain constant	None of these	Decreases
47	Cefficient of fluctuation of speed is	N1-N2 / N	N1*N2 / N	N1+N2 / N	None of these	N1-N2 / N
48	Mean speed during the cycle in r.p.m. is	N1-N2 / N	N1*N2 / N	N1+N2 / N	None of these	N1+N2 / N
49	Maximum fluctuation of energy, ΔE =	Max energy - Min energy	Max energy + Min energy	Max energy / Min energy	Max energy * Min energy	Max energy - Min energy
50	Mean angular speed during the cycle in rad/sec =	ω1 - ω2 / 2	ω1 + ω2 * 2	ω1 + ω2 / 2	None of these	ω1 + ω2 / 2

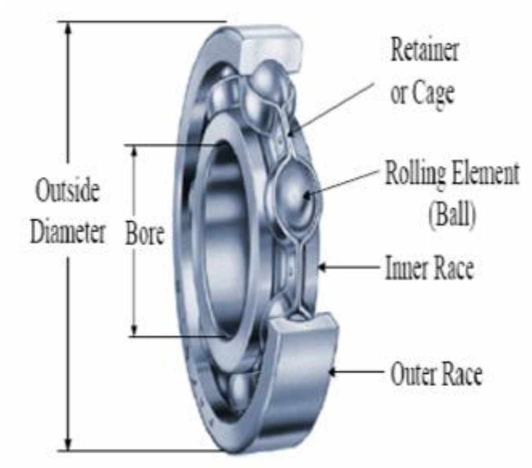
UNIT V

DESIGN OF BEARINGS AND LEVERS

Rolling contact bearings

- The purpose of a bearing is to support a load while permitting relative motion between two elements of a machine.
- The term rolling contact bearings refers to the wide variety of bearings that use spherical balls or some other type of roller between the stationary and the moving elements.
- The most common type of bearing supports a rotating shaft, resisting purely radial loads or a combination of radial and axial (thrust) loads.
- The components of a typical rolling contact bearing are the inner race, the outer race, and the rolling elements.

Rolling contact bearings



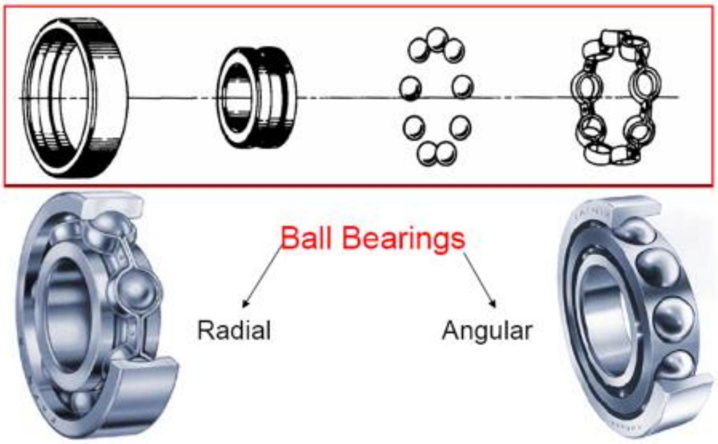
Contact-rolling bearings usually consists of four parts:-

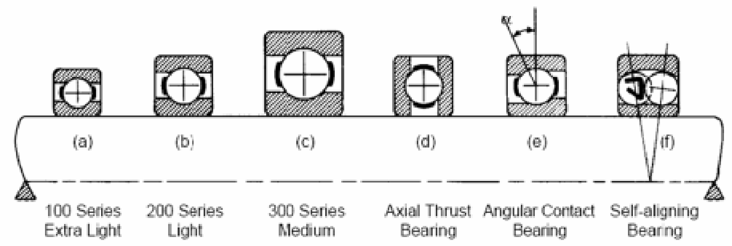
- ❖An inner ring
- ❖An outer ring
- ❖The balls
- ❖The cage or separator.

Types of Rolling Contact Bearings

- Radial loads act toward the center of the bearing along a radius.
- Such loads are typical of those created by power transmission elements on shafts such as spur gears, V-belt drives, and chain drives.
- Thrust loads are those that act parallel to the axis of the shaft.
- The axial components of the forces on helical gears, worms and worm gears, and bevel gears are thrust loads.
- Misalignment refers to the angular deviation of the axis of the shaft at the bearing from the true axis of the bearing itself.

Types of Rolling Contact Bearings



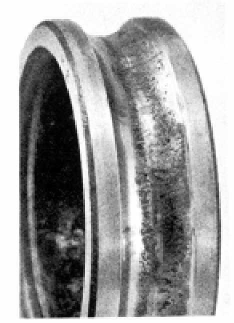


Bearing Failure Modes – Surface Fatigue

Surface fatigue is the dominant failure mode

• The cyclic subsurface Hertzian shear stresses produced by the curved surfaces in rolling contact may initiate and propagate cracks that ultimately dislodge particles and generate surface pits

• Typically, the raceways pit first, resulting in noise, vibration, and heat

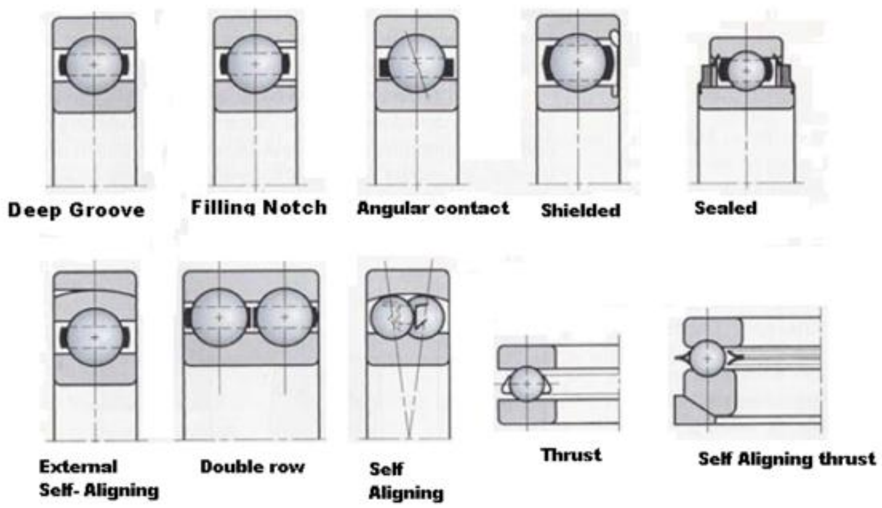


Ball Bearings

Ball bearings are made in a wide variety of types and sizes:

- Single-row radial (carry mostly radial loads, but can also carry axial loads).
- Angular contact bearing (Will take both axial and radial load).
- Axial thrust bearing (When load is directed entirely along the axis, thrust type of bearing should be used).
- Self-aligning bearing (will take care of large amount of misalignment).
- An increase in radial capacity may be secured by using rings with deep grooves, or by employing a double-row radial bearing.

Ball Bearings

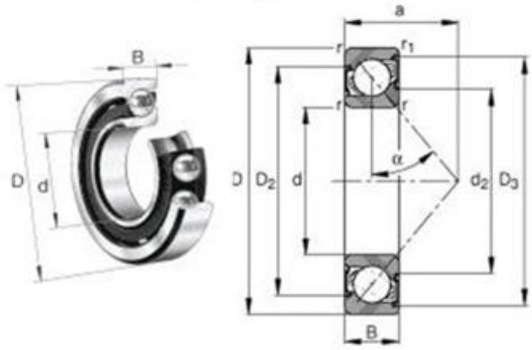


Single row deep groove ball bearing

•The single-row, deep groove ball bearing is what most people think of when the term ball bearing is used.

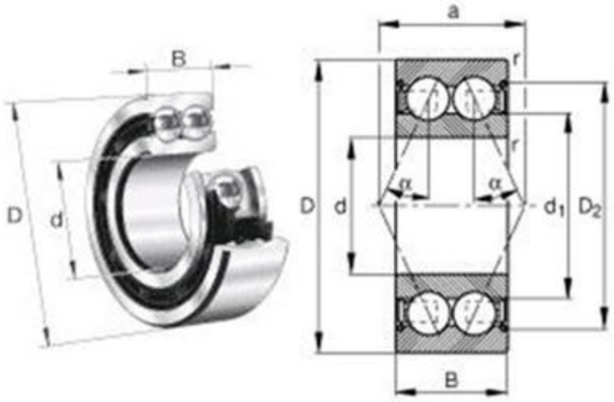
• The inner race is typically pressed on the shaft at the bearing seat with a slight interference fit to ensure that it rotates with the shaft.

• The spherical rolling elements, or balls, roll in a deep groove in both the inner and outer races. The spacing of the balls is maintained by retainers, or cages.

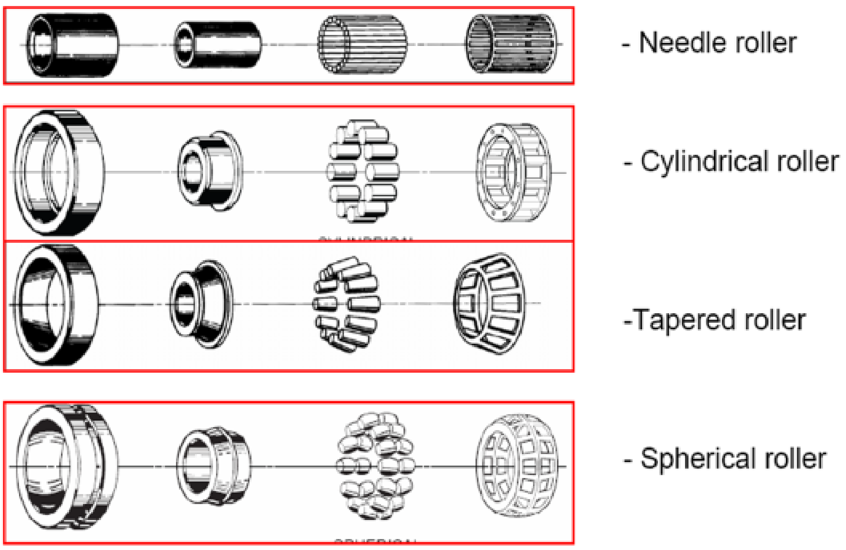


Double row deep groove ball bearing

- Adding a second row of balls increases the radial load-carrying capacity of the deep-groove type of bearing compared with the single-row design because more balls share the load.



Types of Roller bearing



Cylindrical Roller Bearing

- Replacing the spherical balls with cylindrical rollers with corresponding changes in the design of the races, gives a greater radial load capacity.
- The resulting contact stress levels are lower than for equivalent-sized ball bearings, allowing smaller bearings to carry a given load or a given size bearing to carry a higher load.
- Thrust load capacity is poor because any thrust load would be applied to the side of the rollers, causing rubbing, not true rolling motion.

Needle Bearing

- Needle bearings are actually roller bearings, but they have much smaller-diameter rollers.
- A smaller radial space is typically required for needle bearings to carry a given load than for any other type of rolling contact bearing.
- This makes it easier to design them into many types of equipment and components such as pumps, universal joints, precision instruments, and household appliances.

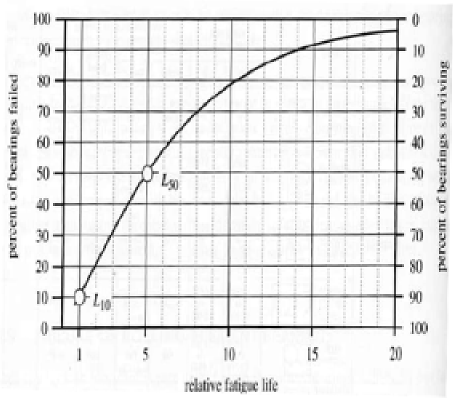
Rolling Contact Bearing Materials

- High-carbon chromium steel 52100, 440C stainless steel and M50 steel are used for balls and rings, and are treated to high strength and hardness. T (360 - 600ºF).
- Silicon nitride is used if high T (2200ºF) and HRC 78.
- The surface are smooth ground and polished. Minimum accepted hardness for bearing components is HRC 58.

Bearing Life: Static Load Capacity

- Static Load Capacity (C_0):
- The static capacity is ordinarily defined as the maximum allowable static load that does not impair the running characteristics of the bearing to make it unusable.
- The bearing is not rotating when the measurement is made.
- The life of a ball bearing is the life in hours at some known speed, or the number of revolutions, that the bearing will attain before the first evidence of fatigue appears on any of the moving part.
- Following nomenclature and definitions are used in the testing of bearing.
- Rate life (L_{10})is the life at which 10 percent of bearing have failed and 90% of them are still good.

Bearing load Life at Rated Reliability



- Median Life (L_{50})is the life at which 50% of the bearings failed and 50% are still good. It is generally not more than 5 time the rate life L_{10}.
- Basic Load Rating (C)For angular or radial contact ball bearing is the calculate, constant, radial load which a group of apparently identical bearings with stationary outer ring can theoretically endure for rating life of one million revolutions of the inner ring.
- For thrust ball bearing it is the calculated, constant, centric, thrust load which a group of apparently identical bearing can theoretically endure for a rating life of one million revolution of one of the bearing washers.

Load/ Life Relationship

$$FL^{1/a} = C$$

C = constant
a = 3 for ball bearing
a = 10/3 for roller bearing

$$F_1 L_1^{1/a} = F_2 L_2^{1/a}$$

Associating F_1 with C_{10} and L_1 with L_{10} , the equation becomes:

$$C_{10} L_{10}^{1/a} = FL^{1/a}$$

The units of L is revolutions

Basic Dynamic Load rating (C_{10})

–The basic Dynamic load rating (C_{10}) is that load which will cause 10% of sample of bearings to fail at or before 1 millions revolutions.

–90% of the bearings would achieve at least 1million revolutions at this load.

- ***Bearing load***
- If two groups of identical bearings are tested under loads P^1 and P_2 for respective lives of L_1 and L_2, then,

$$\frac{L_1}{L_2} = \left(\frac{P_2}{P_1}\right)^a$$

- Where,
- L : life in millions of revolution or life in hours
- a : constant which is 3 for ball bearings and 10/3 for roller bearings
- ***Basic load rating***
- It is that load which a group of apparently identical bearings can withstand for a rating life of one million revolutions.
- if say, L_1 is taken as one million then the corresponding load is,

$$C = P(L)^{\frac{1}{a}}$$

- Where, C is the basic or dynamic load rating

Equivalent radial load

- The load rating of a bearing is given for radial loads only. Therefore, if a bearing is subjected to both axial and radial load, then an equivalent radial load is estimated as,

$$P_e = VP_r \quad \text{or}$$
$$P_e = XVP_r + YP_a$$

Where,

P_e : Equivalent radial load

P_r : Given radial load

P_a : Given axial load

V : Rotation factor (1.0, inner race rotating; 1.2, outer race rotating)

X : A radial factor

Y : An axial factor

- The values of X and Y are found from the chart whose typical format and few representative values are given below.

$\frac{P_a}{C_o}$	e	$\frac{P_a}{P_r} \le e$		$\frac{P_a}{P_r} \ge e$	
		X	Y	X	Y
0.021	0.21	1.0	0.0	0.56	2.15
0.110	0.30	1.0	0.0	0.56	1.45
0.560	0.44	1.0	0.0	0.56	1.00

- The factor, C_o is obtained from the bearing catalogue

The selection procedure

- Depending on the shaft diameter and magnitude of radial and axial load a suitable type of bearing is to be chosen from the manufacturer's catalogue, either a ball bearing or a roller bearing. The equivalent radial load is to be determined
- If it is a tapered bearing then manufacturer's catalogue is to be consulted for the equation given for equivalent radial load.
- The value of dynamic load rating C is calculated for the given bearing life and equivalent radial load.
- From the known value of C, a suitable bearing of size that conforms to the shaft is to be chosen.

Introduction

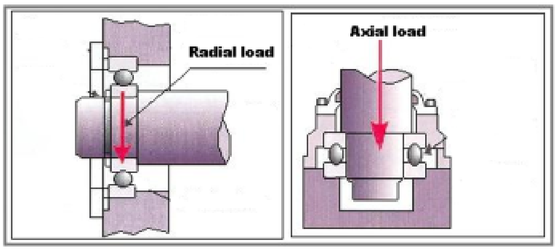
- A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load.
- A little consideration will show that due to the relative motion between the contact surfaces, a certain amount of power is wasted in overcoming frictional resistance and if the rubbing surfaces are in direct contact, there will be rapid wear.
- In order to reduce frictional resistance and wear and in some cases to carry away the heat generated, a layer of fluid (known as lubricant) may be provided.
- The lubricant used to separate the journal and bearing is usually a mineral oil refined from petroleum, but vegetable oils, silicon oils, greases etc., may be used.

Classification of Bearings

- **Depending upon the direction of load to be supported.**

The bearings under this group are classified as:

(a) Radial bearings, and (b) Thrust bearings.



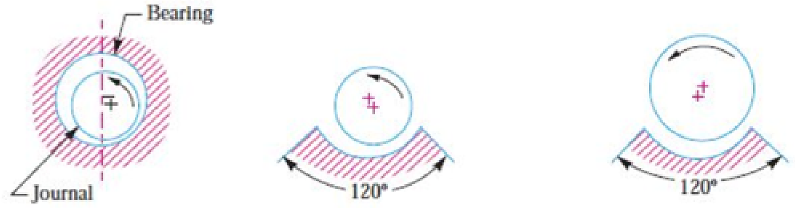
- Depending upon the nature of contact.

The bearings under this group are classified as :

(a) Sliding contact bearings, and (b) Rolling contact bearings.

Types of Sliding Contact Bearings

- The sliding contact bearings in which the sliding action is guided in a straight line and carrying radial loads, is called slipper or guide bearings. Such type of bearings are usually found in crosshead of steam engines.
- The sliding contact bearings in which the sliding action is along the circumference of a circle or an arc of a circle and carrying radial loads are known as ***journal or sleeve bearings.*** When the angle of contact of the bearing with the journal is 360° as shown in Fig, then the bearing is called a ***full journal bearing.*** This type of bearing is commonly used in industrial machinery to accommodate bearing loads in any radial direction.



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

Types of Sliding Contact Bearings

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

1. Thick film bearings.

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

2. *Thin film bearings.*

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

3. *Zero film bearings.*

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

4. *Hydrostatic or externally pressurized lubricated bearings.*

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

Hydrodynamic Lubricated Bearings

- In hydrodynamic lubricated bearings, there is a thick film of lubricant between the journal and the bearing. A little consideration will show that when the bearing is supplied with sufficient lubricant, a pressure is build up in the clearance space when the journal is rotating about an axis that is eccentric with the bearing axis.
- The load can be supported by this fluid pressure without any actual contact between the journal and bearing.
- The load carrying ability of a hydrodynamic bearing arises simply because a viscous fluid resists being pushed around.
- Under the proper conditions, this resistance to motion will develop a pressure distribution in the lubricant film that can support a useful load.

The load supporting pressure in hydrodynamic bearings arises from either

1. the flow of a viscous fluid in a converging channel ***(known as wedge film lubrication),***

- The load carrying ability of a wedge-film journal bearing results when the journal and/or the bearing rotates relative to the load. The most common case is that of a steady load, a fixed (non-rotating) bearing and a rotating journal.

2. The resistance of a viscous fluid to being squeezed out from between approaching surfaces ***(known as squeeze film lubrication).***

- If the load is uniform or varying in magnitude while acting in a constant direction, this becomes a thin film or possibly a zero film problem. But if the load reverses its direction, the squeeze film may develop sufficient capacity to carry the dynamic loads without contact between the journal and the bearing.

Properties of Sliding Contact Bearing Materials

- 1. Compressive strength.
- 2. Fatigue strength.
- 3. Comformability.
- 4. Embeddability.
- 5. Bondability.
- 6. Corrosion resistance.
- 7. Thermal conductivity.
- 8. Thermal expansion.

Properties of Sliding Contact Bearing Materials

Bearing material	*Fatigue strength*	*Comfor-mability*	*Embed-dability*	*Anti scoring*	*Corrosion resistance*	*Thermal conductivity*
Tin base babbit	Poor	Good	Excellent	Excellent	Excellent	Poor
Lead base babbit	Poor to fair	Good	Good	Good to excellent	Fair to good	Poor
Lead bronze	Fair	Poor	Poor	Poor	Good	Fair
Cooper lead	Fair	Poor	Poor to fair	Poor to fair	Poor to fair	Fair to good
Aluminium	Good	Poor to fair	Poor	Good	Excellent	Fair
Silver	Excellent	Almost none	Poor	Poor	Excellent	Excellent
Silver lead deposited	Excellent	Excellent	Poor	Fair to good	Excellent	Excellent

Materials used for Sliding Contact Bearings

1. Babbit metal. The tin base and lead base babbits are widely used as a bearing material, because they satisfy most requirements for general applications.

Tin base babbits : Tin 90% ; Copper 4.5% ; Antimony 5% ; Lead 0.5%.

Lead base babbits : Lead 84% ; Tin 6% ; Anitmony 9.5% ; Copper 0.5%.

2. Cast iron. The cast iron bearings are usually used with steel journals. Such type of bearings are fairly successful where lubrication is adequate and the pressure is limited to 3.5 N/mm2 and speed to 40 meters per minute.

3. Silver. The silver and silver lead bearings are mostly used in aircraft engines where the fatigue strength is the most important consideration.

Materials used for Sliding Contact Bearings

4. Non-metallic bearings. The various non-metallic bearings are made of carbon-graphite, rubber, wood and plastics.

- The carbon-graphite bearings are self lubricating, dimensionally stable over a wide range of operating conditions, chemically inert and can operate at higher temperatures than other bearings.
- The soft rubber bearings are used with water or other low viscosity lubricants, particularly where sand or other large particles are present. In addition to the high degree of Embeddability and Comformability, the rubber bearings are excellent for absorbing shock loads and vibrations.
- The wood bearings are used in many applications where low cost, cleanliness, in attention to lubrication and anti-seizing are important.

Properties of Lubricants

- 1. Viscosity. It is the measure of degree of fluidity of a liquid. It is a physical property by virtue of which an oil is able to form, retain and offer resistance to shearing a buffer film-under heat and pressure. The greater the heat and pressure, the greater viscosity is required of a lubricant to prevent thinning and squeezing out of the film.

$$\tau = \frac{P}{A} \propto \frac{dV}{dy} \quad \text{or} \quad \tau = Z \times \frac{dV}{dy}$$

- where Z is a constant of proportionality and is known as absolute viscosity (or simply viscosity) of the lubricant.

$$Z = \tau \times \frac{h}{V} = \frac{\mathrm{N}}{\mathrm{m}^2} \times \frac{\mathrm{m}}{\mathrm{m/s}} = \mathrm{N\text{-}s/m^2}$$

Properties of Lubricants

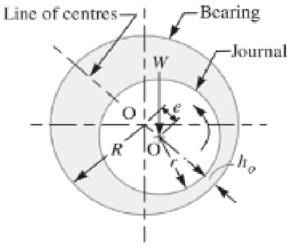
- Oiliness: It is a joint property of the lubricant and the bearing surfaces in contact. It is a measure of the lubricating qualities under boundary conditions where base metal to metal is prevented only by absorbed film. There is no absolute measure of oiliness.
- Density: This property has no relation to lubricating value but is useful in changing the kinematic viscosity to absolute viscosity. Mathematically
  - Absolute viscosity = ρ × Kinematic viscosity (in m2/s)
  - where ρ = Density of the lubricating oil.
- Viscosity index. The term viscosity index is used to denote the degree of variation of viscosity with temperature.

Properties of Lubricants

- Flash point:
  - It is the lowest temperature at which an oil gives off sufficient vapour to support a momentary flash without actually setting fire to the oil when a flame is brought within 6 mm at the surface of the oil.
- Fire point:
  - It is the temperature at which an oil gives off sufficient vapour to burn it continuously when ignited.
- Pour point or freezing point:
  - It is the temperature at which an oil will cease to flow when cooled.

Terminologies used in Hydrodynamic Journal Bearings

- A hydrodynamic journal bearing is shown in Fig, in which O is the centre of the journal and O′ is the centre of the bearing.
- D = Diameter of the bearing,
- d = Diameter of the journal, and
- *l = Length of the bearing.*



- Diametral clearance: It the difference between the diameters of the bearing and the journal. Mathematically, diametral clearance,

$$c = D - d$$

- Note : The diametral clearance (c) in a bearing should be small enough to produce the necessary velocity gradient, so that the pressure built up will support the load.

Terminologies used in Hydrodynamic Journal Bearings

- Radial clearance: It is the difference between the radii of the bearing and the journal. Mathematically, radial clearance,

$$c_1 = R - r = \frac{D - d}{2} = \frac{c}{2}$$

- Diametral clearance ratio. It is the ratio of the diametral clearance to the diameter of the journal. Mathematically, diametral clearance ratio

$$= \frac{c}{d} = \frac{D - d}{d}$$

- Eccentricity: It is the radial distance between the centre (O) of the bearing and the displaced centre (O′) of the bearing under load. It is denoted by e.

Terminologies used in Hydrodynamic Journal Bearings

- Minimum oil film thickness: It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by h_0 . Its value may be assumed as c/4.
- Attitude or eccentricity ratio. It is the ratio of the eccentricity to the radial clearance. Mathematically, attitude or eccentricity ratio,

$$\varepsilon = \frac{e}{c_1} = \frac{c_1 = h_0}{c_1} = 1 - \frac{h_0}{c_1} = 1 - \frac{2h_0}{c}$$

- Short and long bearing. If the ratio of the length to the diameter of the journal (i.e. l / d) is less than 1, then the bearing is said to be short bearing. On the other hand, if l/d is greater than 1, then the bearing is known as long bearing

Bearing Characteristic Number and Bearing Modulus for Journal Bearings

- The coefficient of friction in design of bearings is of great importance, because it affords a means for determining the loss of power due to bearing friction. It has been shown by experiments that the coefficient of friction for a full lubricated journal bearing is a function of three variables, i.e.

(i) $\frac{ZN}{p}$; (ii) $\frac{d}{c}$; and (iii) $\frac{l}{d}$

Therefore the coefficient of friction may be expressed as

$$\mu = \phi\left(\frac{ZN}{p}, \frac{d}{c}, \frac{l}{d}\right)$$

- μ = Coefficient of friction,
- p = Bearing pressure on the projected bearing
  - = Load on the journal ÷ l × d
- φ = A functional relationship,
- Z = Absolute viscosity of the lubricant kg / m s
- d = Diameter of the journal,
- N = Speed of the journal in r.p.m.,
- l = Length of the bearing, and c = Diametral clearance.

Bearing Characteristic Number and Bearing Modulus for Journal Bearings

- The factor ZN/p is termed as bearing characteristic number and is a dimensionless number.
- Coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k$$

- k = Factor to correct for end leakage. It depends upon the ratio of length to the diameter of the bearing (i.e. l/d). = 0.002 for l/d ratios of 0.75 to 2.8.
- The operating values of ZN / p should be compared with values given in Tables of PSG data Book

Critical Pressure of the Journal Bearing

- The pressure at which the oil film breaks down so that metal to metal contact begins, is known as critical pressure or the minimum operating pressure of the bearing. It may be obtained by the following empirical relation, i.e.
- Critical pressure or minimum operating pressure,

$$p = \frac{ZN}{4.75 \times 10^6}\left(\frac{d}{c}\right)^2\left(\frac{l}{d+l}\right) \text{ N/mm}^2 \qquad \text{(when Z is in kg / m-s)}$$

Sommerfeld Number

- The Sommerfeld number is also a dimensionless parameter used extensively in the design of journal bearings. Mathematically,

$$\text{Sommerfeld number} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2$$

- For design purposes, its value is taken as follows :

$$\frac{ZN}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6$$

- (when Z is in kg / m-s and p is in N / mm^2)

Heat Generated in a Journal Bearing

- The heat generated in a bearing is due to the fluid friction and friction of the parts having relative motion. Mathematically, heat generated in a bearing,

$$Q_g = \mu . W . V \text{ N-m/s or J/s or watts}$$

- μ = Coefficient of friction,
- W = Load on the bearing in N,
  = Pressure on the bearing in N/mm^2 × Projected area of the bearing in mm^2
  = p (l × d),
- V = Rubbing velocity in m/s d is in metres, $= \frac{\pi d . N}{60}$
- N = Speed of the journal in r.p.m.

Heat Generated in a Journal Bearing

- After the thermal equilibrium has been reached, heat will be dissipated at the outer surface of the bearing at the same rate at which it is generated in the oil film. The amount of heat dissipated will depend upon the temperature difference, size and mass of the radiating surface and on the amount of air flowing around the bearing.
- However, for the convenience in bearing design, the actual heat dissipating area may be expressed in terms of the projected area of the journal.

Heat dissipated by the bearing, $Q_d = C.A\,(t_b - t_a)$ J/s or W

- C = Heat dissipation coefficient in $W/m^2/°C$,
- A = Projected area of the bearing in m^2 = l × d,
- t_b = Temperature of the bearing surface in °C, and
- t_a = Temperature of the surrounding air in °C.

Heat Generated in a Journal Bearing

- The average values of C (in $W/m^2/°C$), for journal bearings may be taken as follows :
- For unventilated bearings (Still air)
  $= 140$ to $420\ W/m^2/°C$
- For well ventilated bearings
  $= 490$ to $1400\ W/m^2/°C$
- It has been shown by experiments that the temperature of the bearing (t_b) is approximately mid-way between the temperature of the oil film (t_0) and the temperature of the outside air (t_a). In other words,

$$t_b - t_a = \frac{1}{2}(t_0 - t_a)$$

Heat Generated in a Journal Bearing

- For well designed bearing, the temperature of the oil film should not be more than 60°C, otherwise the viscosity of the oil decreases rapidly and the operation of the bearing is found to suffer. The temperature of the oil film is often called as the operating temperature of the bearing.
- In case the temperature of the oil film is higher, then the bearing is cooled by circulating water through coils built in the bearing.
- The mass of the oil to remove the heat generated at the bearing may be obtained by equating the heat generated to the heat taken away by the oil. We know that the heat taken away by the oil,

$$Q_t = m.S.t \quad J/s \text{ or watts}$$

where

- m = Mass of the oil in kg / s,
- S = Specific heat of the oil. Its value may be taken as 1840 to 2100 J / kg / °C,
- t = Difference between outlet and inlet temperature of the oil in °C.

Design Procedure for Journal Bearing

The following procedure may be adopted in designing journal bearings, when the bearing load, the diameter and the speed of the shaft are known.

- 1. Determine the bearing length by choosing a ratio of l/d from related Table.
- 2.Check the bearing pressure, p = W / l.d from related Table for probable satisfactory value.
- 3. Assume a lubricant from Table and its operating temperature (t_0). This temperature should be between 26.5°C and 60°C with 82°C as a maximum for high temperature installations such as steam turbines.
- 4. Determine the operating value of ZN / p for the assumed bearing temperature and check this value with corresponding values in related Table, to determine the possibility of maintaining fluid film operation.
- 5. Assume a clearance ratio c/d from related Table.

Design Procedure for Journal Bearing

- 6. Determine the coefficient of friction (μ) by using the relation as discussed in previous slides.
- 7. Determine the heat generated by using the relation as discussed in previous slides.
- 8. Determine the heat dissipated by using the relation as discussed in previous slides.
- 9. Determine the thermal equilibrium to see that the heat dissipated becomes at least equal to the heat generated. In case the heat generated is more than the heat dissipated then either the bearing is redesigned or it is artificially cooled by water.

MULTIPLE CHOICE QUESTIONS:

S.No	Questions	Option A	Option B	Option C	Option D	Answer
1	In a full journal bearing, the angle of contact of the bearing with the journal is	120o	180o	270o	360o	360o
2	A sliding bearing which can support steady loads without any relative motion between the journal and the bearing is called	Zero film bearing	Boundary lubricated bearing	Hydrodynamic lubricated bearing	Hydrostatic lubricated bearing	Hydrostatic lubricated bearing
3	In a boundary lubricated bearing, there is a ________ of lubricant between the journal and the bearing.	Thick film	Thin film	Wide film	None of these	Thin film
4	When a shaft rotates in anticlockwise direction at slow speed in a bearing, then it will	Have contact at the lowest point of bearing	Move towards right of the bearing making metal to metal contact	Move towards left of the bearing making metal to metal contact	Move towards rigtht of the bearing making no metal to metal contact	Move towards left of the bearing making metal to metal contact
5	The property of a bearing material which has the ability to accommodate small particles of dust, grit etc., without scoring the material of the journal, is called	Bondability	Embeddability	Conformability	Fatigue strength	Embeddability
6	Teflon is used for bearings because of	Low coefficient of friction	Better heat dissipation	Smaller space consideration	All of these	Smaller space consideration
7	When the bearing is subjected to large fluctuations of load and heavy impacts, the bearing characteristic number should be _______ the bearing modulus.	5 times	10 times	15 times	20 times	15 times
8	When the length of the journal is equal to the diameter of the journal, then the bearing is said to be a	Short bearing	Long bearing	Medium bearing	Square bearing	Square bearing
9	If Z= Absolute viscosity of the lubricant in kg/m-s, N- Speed of the journal in r.p.m., and p= Bearing pressure in N/mm2, then the bearing characteristic number is	ZN/p	Zp/N	Z/pN	pN/Z	ZN/p
10	In thrust bearings, the load acts	along the axis of rotation	Parallel to the axis of rotation	Prependicular to the axis of rotation	In any direction	along the axis of rotation
11	The rolling contact bearings are known as	Thick lubricated bearings	Plastic bearings	Thin lubricated berainga	Antifriction bearings	Antifriction bearings
12	The bearings of medium series have capacity ________ over the light series.	10 to 20 %	20 to 30 %	30 to 40 %	40 to 50 %	30 to 40 %
13	The bearing of heavy series have capacity __________ over the medium series.	10 to 20 %	20 to 30 %	30 to 40 %	40 to 50 %	20 to 30 %
14	The ball bearings are usually made from	Low carbon steel	Medium carbon steel	High speed steel	Chrome nickel steel	Chrome nickel steel
15	The tapered roller bearings can take	Radial load only	Axial load only	Both radial and axial loads	None of these	Both radial and axial loads

16	The piston pin bearings in heavy duty diesel engines are	Needle roller bearings	Tapered roller bearings	Spherical roller bearings	Cylindrical roller bearings	Needle roller bearings
17	Which of the following is antifriction bearing ?	Journal bearing	Pedestal bearing	Collar bearing	Needle bearing	Needle bearing
18	Ball and roller bearings in comparison to sliding bearings have	More accuracy in alignment	Small overall dimensions	Low starting and running friction	all of these	all of these
19	A bearing is designated by the number 405. It means that a bearing is of	Light series with bore of 5mm	medium series with bore of 15mm	Heavy series with bore of 25mm	Light series with width of 20mm	Heavy series with bore of 25mm
20	The listed life of a rolling bearing, in a catalogue, is the	Minimum expected life	Maximum expected life	Average life	None of these	Minimum expected life
21	________type of bearing is used in agricultural machinery	Deep groove ball bearing	cylindrical roller bearing	Angular contact bearing	Taper roller bearing	Angular contact bearing
22	________type of bearing is used in rail road axle boxes	Deep groove ball bearing	cylindrical roller bearing	Angular contact bearing	Taper roller bearing	Taper roller bearing
23	________type of bearing is used in worm gear boxes	cylindrical roller bearing	Angular contact bearing	Taper roller bearing	Thrust ball bearing	Thrust ball bearing
24	The cages of the rolling conatct bearing is made of	Low carbon steel	high carbon chromium steel	Alloy steel	case hardened steel	Low carbon steel
25	The minimum hardness level of balls used in rolling contact bearings	78 Rockwell C	58 Rockwell C	52 Rockwell C	68 Rockwell C	58 Rockwell C
26	The IS code used for Rolling bearings for static load ratings is	IS 3824 - 1988	IS 3822 - 1988	IS 3823 - 1988	IS 3821 - 1988	IS 3823 - 1988
27	The force required to produce a given permanent deformation of the ball is given by	kd	kd^3	kd^2	k/d^2	kd^2
28	The bearing life for a wheel bearing in trolley cars is _______ million revolutions	50	100	500	1000	500
29	The bearing life for a wheel bearing in Automobile cars is _______ million revolutions	50	100	500	1000	50
30	To avoid the seizure of a sliding conatct bearing the bearing characteristic number should be atleast _______ times bearing modulus when the coefficient of friction is minimum	3 to 4	4 to 5	5 to 6	6 to 7	5 to 6
31	In levers, the leverege is the ratio of	Load lifted to the effort applied	Mechanical advantage to the velocity ratio	load arm to the effort arm	Effort arm to the load arm	Effort arm to the load arm
32	In the levers of first type, the mechanical advantage is ____________one.	Less than	Euqal to	More than	Not equal to	More than
33	The bell crank levers used in railway signalling arrangement are of	First type of levers	Second type of levers	Third type of levers	None of these	Third type of levers
34	The rocker arm in internal combustion engines are of ____________type of levers	First	Second	Third	None of these	First
35	The cross-section of the arm of the bell crank lever is	Rectangular	Elliptical	I-section	Any one of these	Any one of these

36	All the types of levers are subjected to	Twisting moment	Bending moment	Direct axial load	Combined twisting and bending moment	Bending moment
37	The method of manufacturing usually adopted for levers is	Casting	Fabrication	Forging	Machining	Forging
38	An I-section is more suitable for a	Rocker arm	Cranked lever	Foot lever	Lever of lever safety valve	Rocker arm
39	The design of the pin of a rocker arm of an I.C. Engine is based on	Tensile, creep and bearing failure	creep, bearing and shearing failure	Bearing, shearing and bending failure	None of these	Bearing, shearing and bending failure
40	In designing a rocker arm for operating the exhaust valve, the ratio of the length to the diameter of the fulcrum and roller pin is taken as	1.25	1.5	1.75	2	1.25
41	A lever is a rigid rod or bar capable of turning about a fixed point called	Pivot	Fulcrum	Hinge	None of these	Fulcrum
42	The ratio of load lifted to the effort applied is called	Simple advantage	Electrical advantage	Mechanical advantage	None of these	Mechanical advantage
43	The perpendicular distance between the load point and fulcrum (l1) is known as	Load arm	Effort arm	Fulcrum	None of these	Load arm
44	The perpendicular distance between the effort point and fulcrum (l2) is known as	Load arm	Effort arm	Fulcrum	None of these	Effort arm
45	The ratio of the effort arm to the load arm (l1/l2) is called	Fulcrum	leverage	arm	None of these	leverage
46	In order to obtain a great leverage, ________________may be used.	Foot levers	Cranked levers	Compound levers	Hand levers	Compound levers
47	The load and effort cause moments in ________directions about the fulcrum.	Parallel	Opposite	Perpendicular	None of these	Opposite
48	The height of the lever is usually taken as times the thickness of the lever.	2 to 5	5 to 10	10 to 15	15 to 20	2 to 5
49	In the bell crank lever, the two arms of the lever are at	90o	120o	360o	180o	90o
50	The rocker arm is usually of	L-section	I-section	T-section	None of these	I-section