MACHINE DESIGN

It is define as the use of scientific principles, technical information and imagination in the description of a machine or mechanical system to perform specific function with maximum economy and efficiency.

Unit I
GENERAL CONSIDERATIONS

1. Type of load and stress
2. Motion of the parts
3. Selection of materials
4. Form and size of the part
5. Convenient and economical features
6. Use of standard part
7. Safety of operation
8. Workshop facilities
9. Numbers of machines to be manufactured
10. Assembling

General Design Procedure

- Recognition of need
- Synthesis (Mechanism)
- Analysis of forces
- Material selection
- Design of elements (size and stress)
- Modification
- Detailed drawing
- Production

Classification of Engg. materials

1. Metals and their alloys- iron, steels, copper

- Ferrous metals- iron as main constituent

  cast iron, wrought iron, steels

- Non ferrous metals- other than iron

  copper, aluminum, brass etc.

2. Non- metals- glass, rubber, plastic

Selection of Materials

The best materials is one which serve the desired objective at the minimum cost.

- Availability of materials
- Suitability of material for the working condition in service,
- The cost of materials
- Gives the proper strength to withstand the applied loads

Indian standard designation of materials

A) **Cast iron designation on the basis of mechanical properties**
B) **Steel designation on the basis of mechanical properties**
C) **Steel designation on the basis of Chemical composition**

Mechanical Properties

- Strength
- Elasticity
- Plasticity
- Stiffness
- Resilience
- Toughness
- Malleability
- Ductility
- Brittleness
- hardness

Design against static load

- Static load: force that is gradually applied and does not changes its magnitude and direction w. r. t. time.

Modes of failure

1. Failure by elastic deflection
2. Failure by general yielding
3. Failure by fracture

1. Failure by elastic deflection

- e.g. Buckling of columns, transmission shaft supporting gears etc.
- Lateral or torsional rigidity is considered as the criteria of design.
- The moduli of elasticity and rigidity are the important properties

2. Failure by general yielding

- e.g. mechanical components made of ductile material fails due to a large amount of plastic deformation.

- Considerable portion of the component is subjected to plastic deformation called general yielding.

- Yield strength is an important property

3. Failure by fracture

- e.g. mechanical components made of brittle materials fails due to fracture without any plastic deformation.

- Ultimate tensile strength is an important property.

Factor of safety

- While designing a component it is necessary to provide sufficient reserve strength in case of an accident.

it is achieved by a suitable factor of safety

$$Fs = \frac{Failure\,stress}{Allowable\,stress} = \frac{Failure\,load}{Working\,load}$$

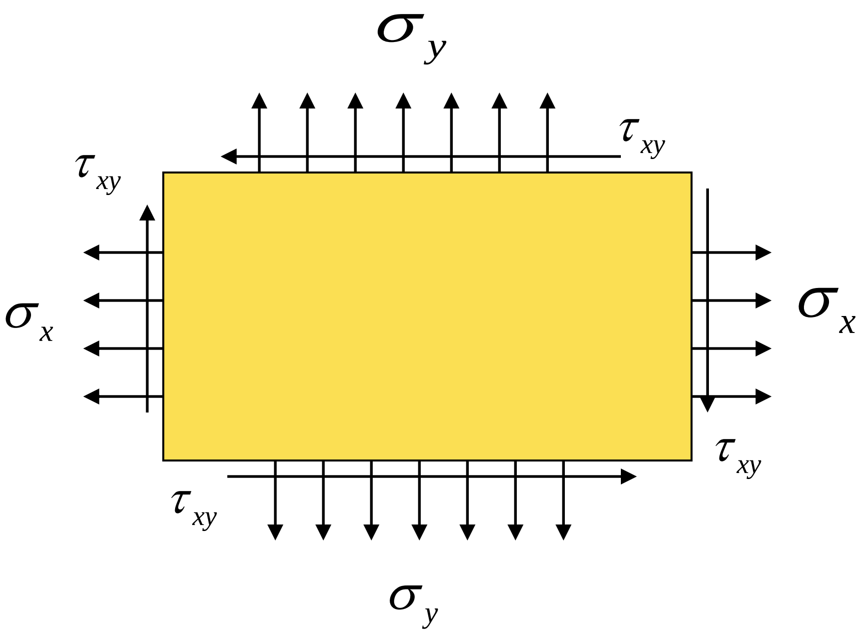
$$Allowable\,Stress\;\; \sigma = \frac{S_{yt}}{Fs},\; for\,ductile\,materials$$

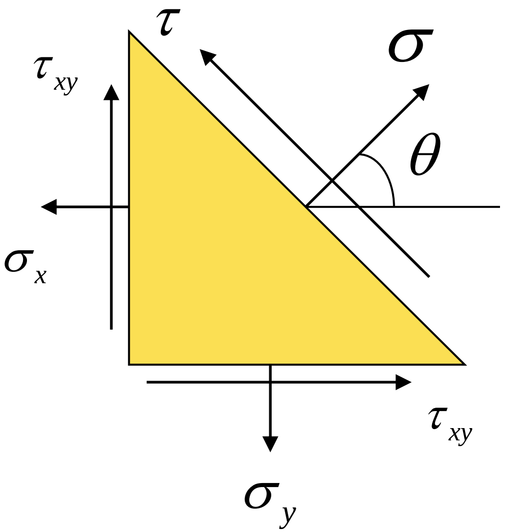
$$\sigma = \frac{S_{ut}}{Fs},\; for\,brittle\,materials$$

Factor of safety depends upon:-

1. Effect of failure
2. Type of load
3. Degree of accuracy in force analysis
4. Material of component
5. Reliability of component
6. Cost of component
7. Testing of machine element
8. Service conditions
9. Quality of manufacture

Stresses on an oblique plane





$$\boxed{\sigma = \frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\cos 2\theta + \tau_{xy}\sin 2\theta}$$

$$\boxed{\tau = \frac{1}{2}\left(\sigma_x - \sigma_y\right)\sin 2\theta - \tau_{xy}\cos 2\theta}$$

Principal stresses and planes

- No shear stress i.e. $\tau = 0$

$$\boxed{\tan 2\theta = \frac{2\tau_{xy}}{\sigma_x - \sigma_y}}$$

- Principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \frac{1}{2}\sqrt{\left(\sigma_x - \sigma_y\right)^2 + 4\tau_{xy}{}^2}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \frac{1}{2}\sqrt{\left(\sigma_x - \sigma_y\right)^2 + 4\tau_{xy}{}^2}$$

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

Theories of failure under static load

1. Maximum principal or normal stress theory (Rankine’s theory)
2. Maximum shear stress theory (Guest’s or Tresca’s theory)
3. Maximum principal strain theory (Saint Venant’s theory)
4. Maximum strain energy theory (Haighs theory)
5. Maximum distortion energy theory (Hencky and von-mises theory)

1. Maximum principal or normal stress theory (Rankine's theory)

- Failure or yielding occurs at a point when the maximum principal or normal stress reaches the limiting strength of the material.

$$\boxed{\sigma_1 = \frac{\sigma_{yt}}{fs} \qquad \text{For ductile material}}$$

$$\boxed{\sigma_1 = \frac{\sigma_{ut}}{fs} \qquad \text{For brittle material}}$$

- Mostly used for brittle material.

2. Maximum shear stress theory (Guest's or Tresca's theory)

- Failure or yielding occurs at a point when the maximum shear stress reaches a value equal to the shear stress at yield point in a simple tension test.

$$\boxed{\tau_{\max} = \frac{\tau_{yt}}{fs} = \frac{0.5\sigma_{yt}}{fs}}$$

- Mostly used for ductile material.

3. Maximum principal strain theory (Saint Venant's theory)

- Failure or yielding occurs at a point when the maximum principal strain reaches a limiting value of strain.

$$\in_{\max} = \frac{\sigma_1}{E} - \frac{\sigma_2}{mE}$$

$$\frac{\sigma_1}{E} - \frac{\sigma_2}{mE} = \frac{\sigma_{yt}}{E\ fs}$$

$$\boxed{\sigma_1 - \frac{\sigma_2}{m} = \frac{\sigma_{yt}}{fs}}$$

4. Maximum strain energy theory (Haigh's theory)

- Failure or yielding occurs at a point when strain energy per unit volume reaches a limiting value of strain energy per unit volume as determined from simple tension test.

$$U_1 = \frac{1}{2E}\left[(\sigma_1)^2 + (\sigma_2)^2 - \frac{2\sigma_1\sigma_2}{m}\right]$$

$$U_2 = \frac{1}{2E}\left[\frac{\sigma_{yt}}{fs}\right]^2$$

$$\boxed{(\sigma_1)^2 + (\sigma_2)^2 - \frac{2\sigma_1\sigma_2}{m} = \left(\frac{\sigma_{yt}}{fs}\right)^2}$$

- May be used for ductile material.

5. Maximum distortion energy theory (Hencky and von-mises theory)

- Failure or yielding occurs when the distortion energy (also called shear strain energy) per unit volume reaches the limiting value of distortion energy.

$$\boxed{(\sigma_1)^2 + (\sigma_2)^2 - 2\sigma_1\sigma_2 = \left(\frac{\sigma_{yt}}{fs}\right)^2}$$

- Mostly used for ductile material in place of maximum strain energy theory.

Problem 1. The load on a bolt consists of an axial pull of 10 kN together with a transverse shear force of 5 kN. Find the diameter of bolt required according to

(a) Maximum principal stress theory

(b) Maximum shear stress theory

(c) Maximum principal strain theory

(d) Maximum strain energy theory

(e) Maximum distortion energy theory.

Take permissible tensile stress at elastic limit = 100 MPa and poisons ratio = 0.3.

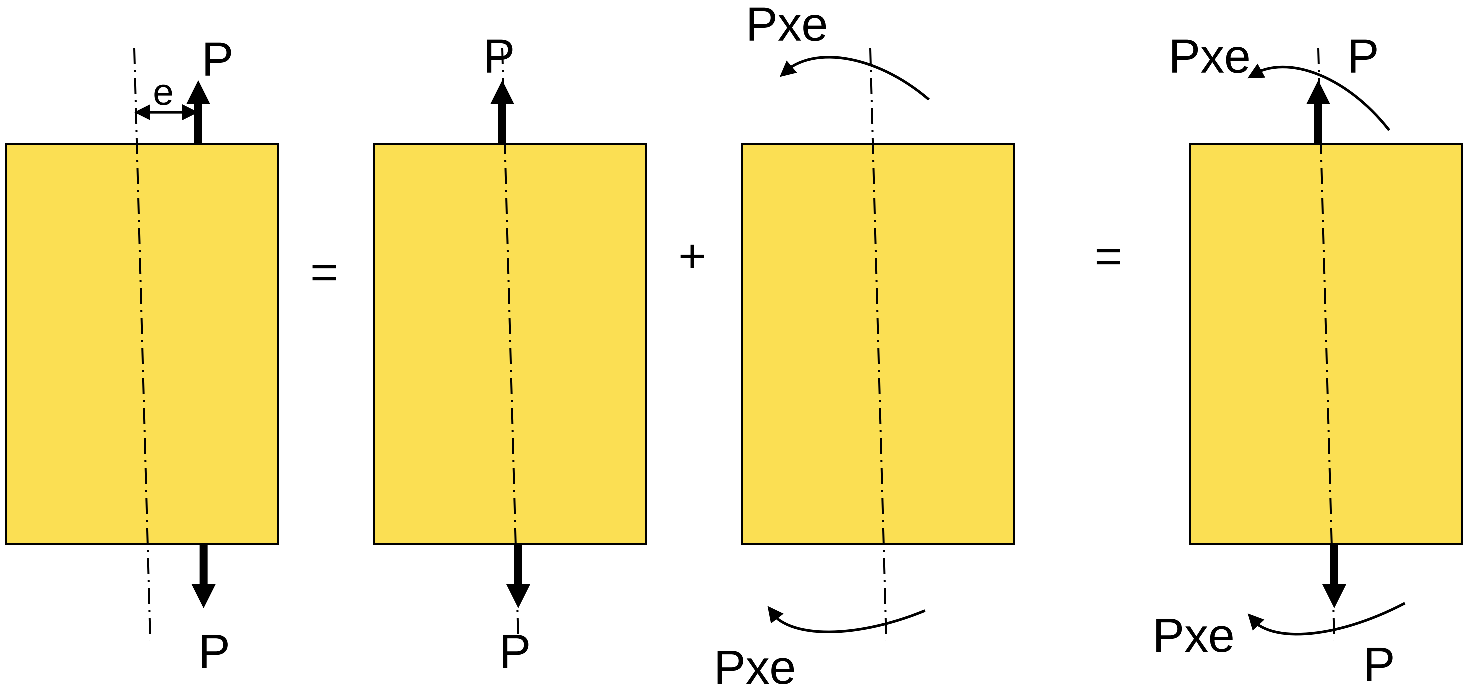
Problem 2. The forces acting on a bolt consists of two components an axial pull of 12 kN transverse shear force of 6 kN. The bolt is made of steel (S_Y= 310 N/mm^2) and the factor of safety is 2.5. Determine the diameter of bolt using the maximum shear stress theory of failure.

Problem 3 A cylindrical shaft made of steel of yield strength 700 MPa is subjected to static loads consisting of bending moment 10 kN-m and a torsional moment of 30 kN-m . Determine the diameter of the shaft using two different theories of failure and assuming factor of safety =2. Take E=210 GPa and poison ratio= 0.25.

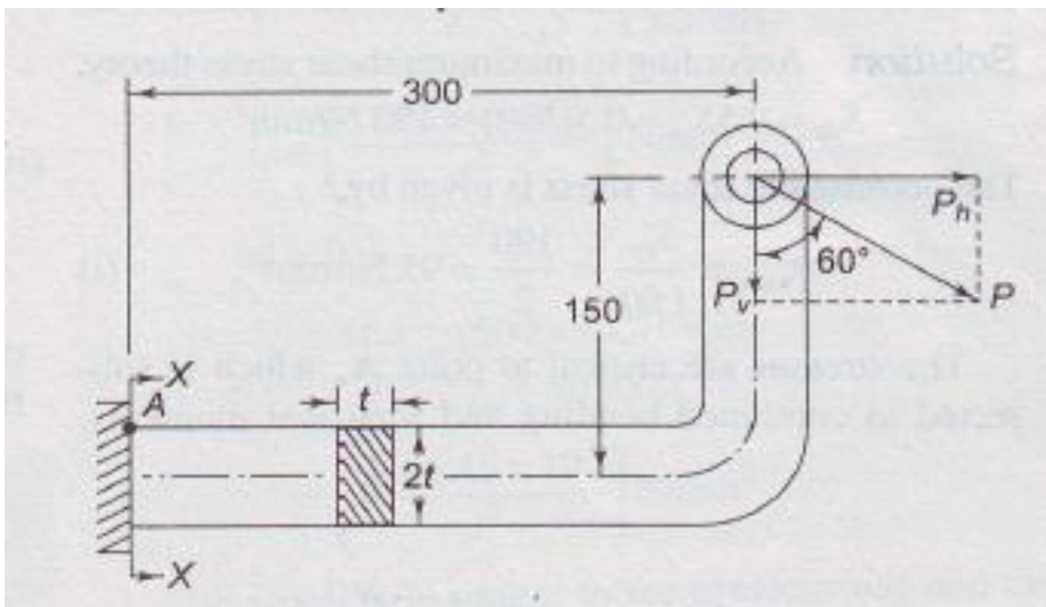
Problem 4 A mild steel shaft of 50 mm diameter is subjected to a bending moment of 2000 N-m and a torque T. If the yield point of the steel in tension is 200 MPa find the maximum value of this torque without causing yielding of the shaft according to

1. maximum principal stress theory
2. maximum shear stress theory
3. maximum distortion energy theory.

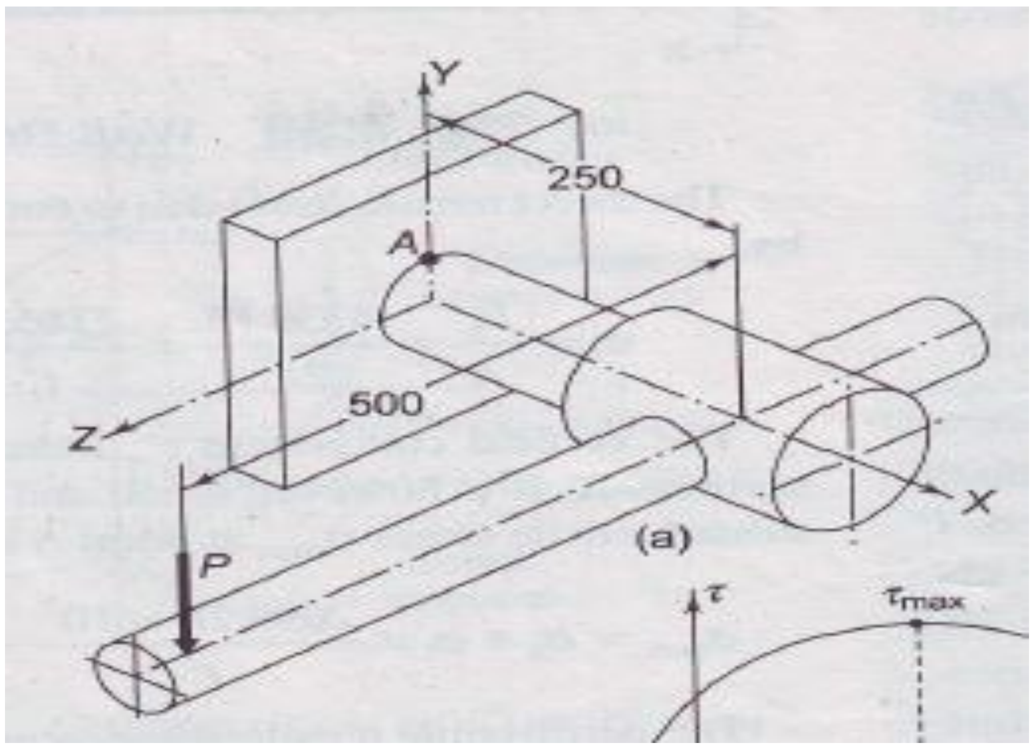
Eccentric axial loading
(direct and bending stress combined)



Problem.5 A wall bracket with a rectangular cross section is shown in fig. The depth of the cross section is twice the width. The force P acting on bracket at 60^0 to the vertical is 5 kN. The material of the bracket is grey cast iron FG 200 and factor of safety is 3.5. Determine the dimensions of the cross section of the bracket. Assume maximum principal stress theory of failure.



Problem 6 The shaft of an overhang crank subjected to a force of 1 kN is shown in fig. The shaft is made of plain carbon steel 45C8 and the tensile yield strength is 380 N/mm^2. The factor of safety is 2. Determine the diameter of the shaft using the maximum shear stress theory.



Design against fluctuating load

- Stress Concentration:

it is defined as the localization of high stresses due to the irregularities present in the component and abrupt changes of the cross section.

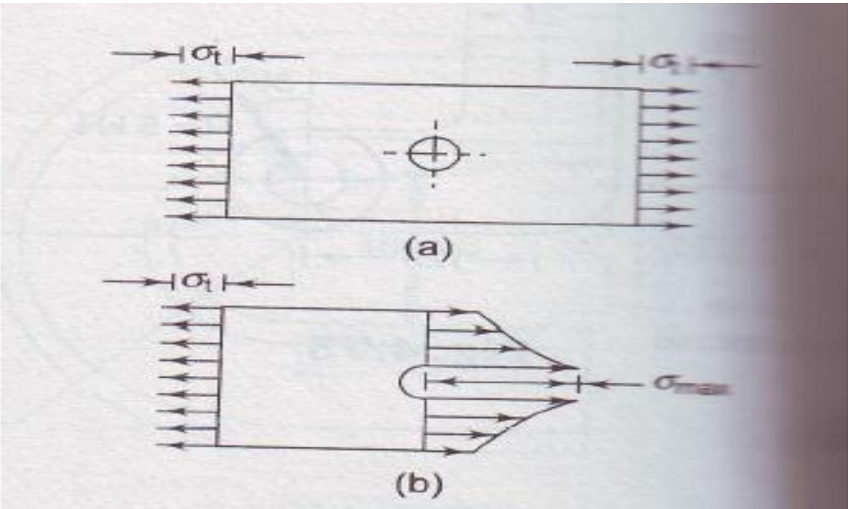


Fig. 5.1 *Stress Concentration*

Causes of stress concentration

1. variation in properties of material
2. Load application
3. Abrupt changes in section
4. Discontinuity in the component
5. Machining scratches

Reduction of stress concentration

1. Additional notches and holes in tension member:

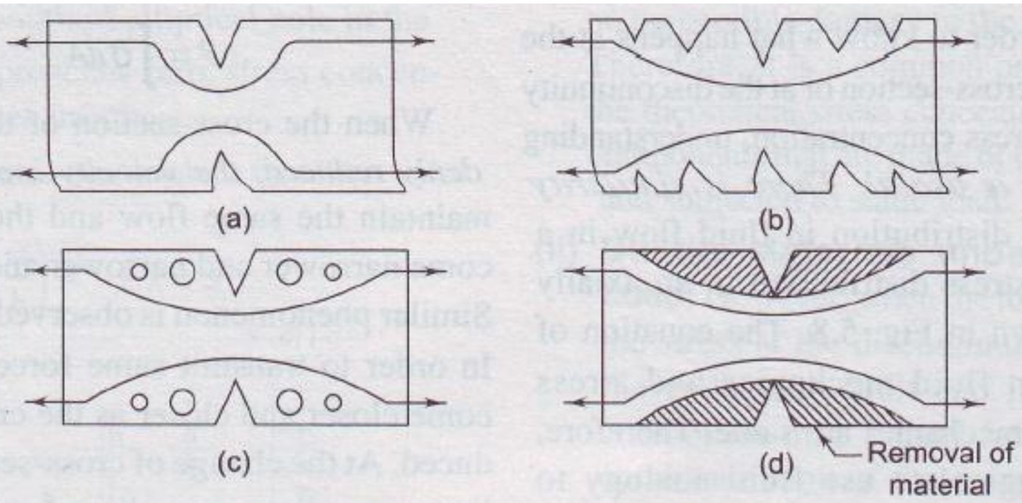


Fig. 5.9 *Reduction of Stress Concentration due to V-notch (a) Original Notch (b) Multiple Notches (c) Drilled Holes (d) Removal of Undesirable Material*

2. Fillet radius, undercutting and notch for member in bending:

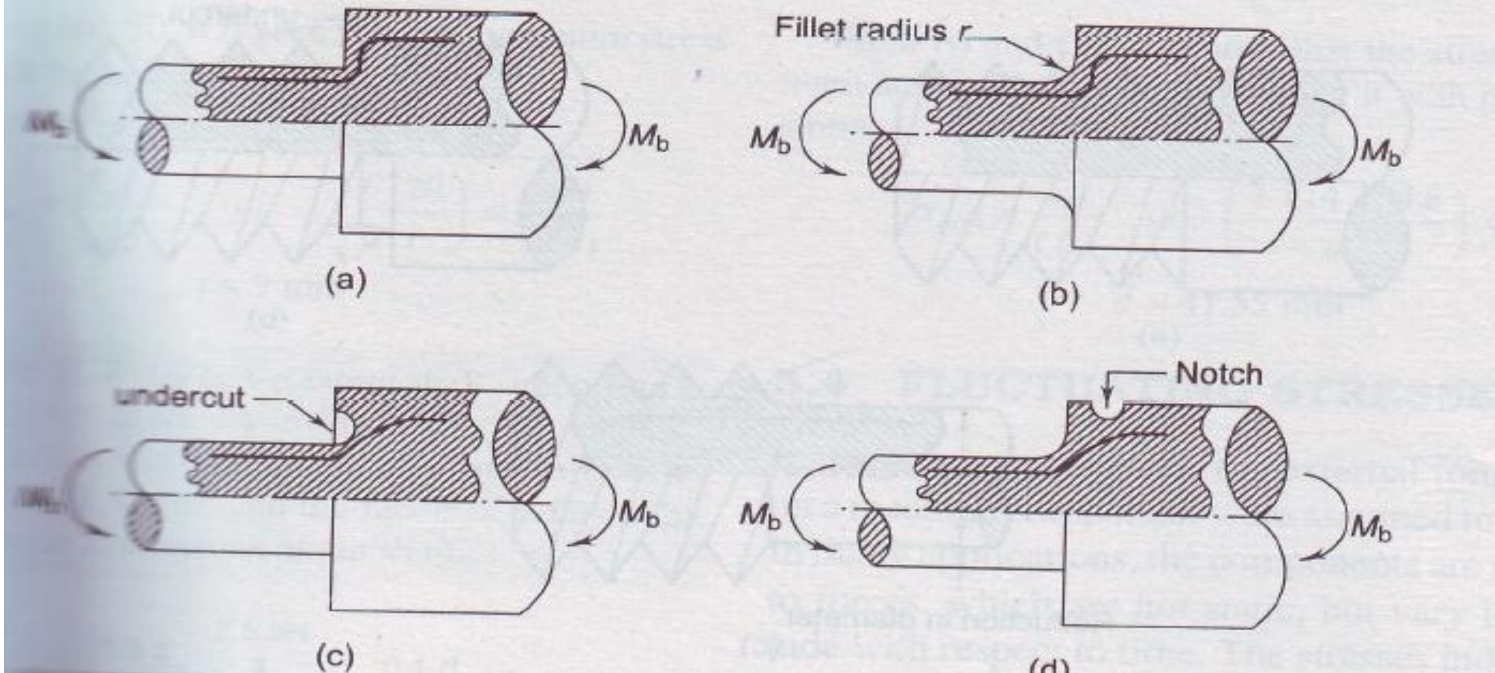
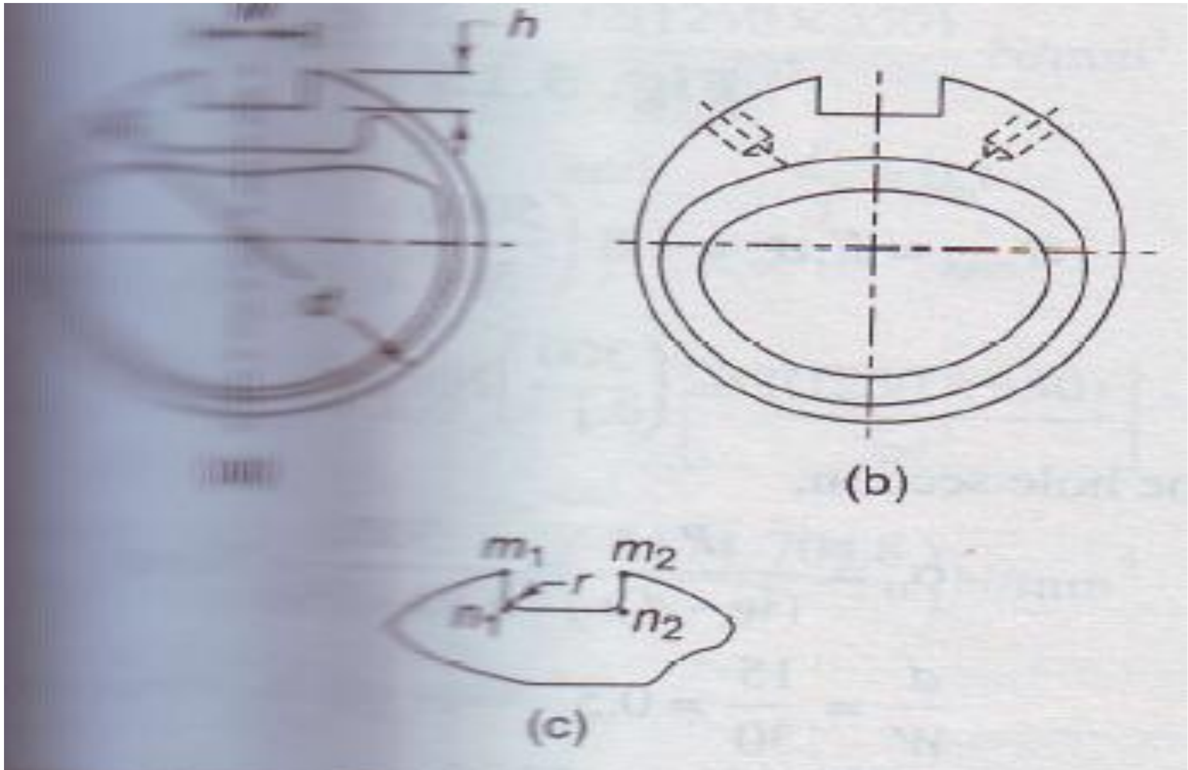


Fig. 5.10 *Reduction of Stress Concentration due to Abrupt Change in Cross-section (a) Original Component (b) Fillet Radius (C) Undercutting (d) Addition of Notch*

3. Drilling additional holes for shaft:



Reduction of Stress Concentration in Shaft with Keyway (a) Original Shaft (b) Drilled Holes (c) Fillet Radius

4. Reduction of stress concentration in threaded members:

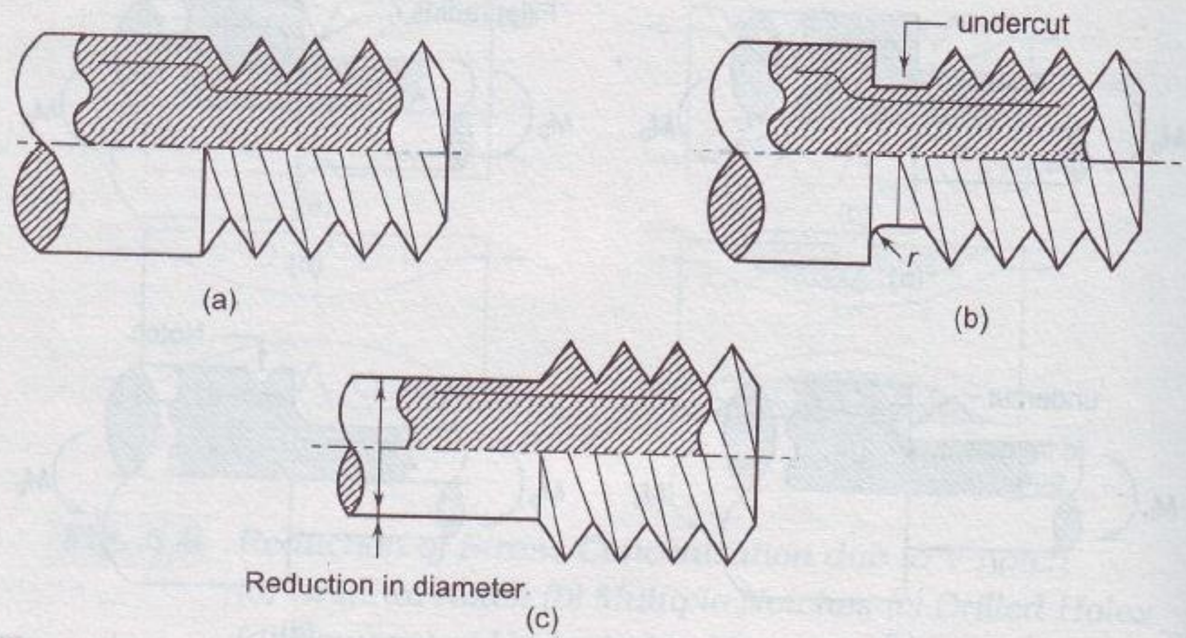


Fig. 5.12 *Reduction of Stress Concentration in Threaded Components (a) Original Component (b) Undercutting (c) Reduction in Shank Diameter*

Stress concentration factor

Theoretical stress concentration factor is defined as the ratio of maximum stress to the nominal stress. It is given by K_t. It depends upon the material and geometry.

$$K_t = \frac{\sigma_{\max}}{\sigma_o} = \frac{\tau_{\max}}{\tau_o}$$

Where σ_0 and ζ_0 are stresses obtained by elementary equations for minimum cross section. And σ_{max} and ζ_{max} are localized stresses at the discontinuity.

Stress concentration factors

- For a rectangular plate with transverse hole

  Nominal stress $\sigma_0 = \dfrac{P}{(w-d)t}$

- For a flat plate with a shoulder fillet

  Nominal stress $\sigma_0 = \dfrac{P}{dt}$

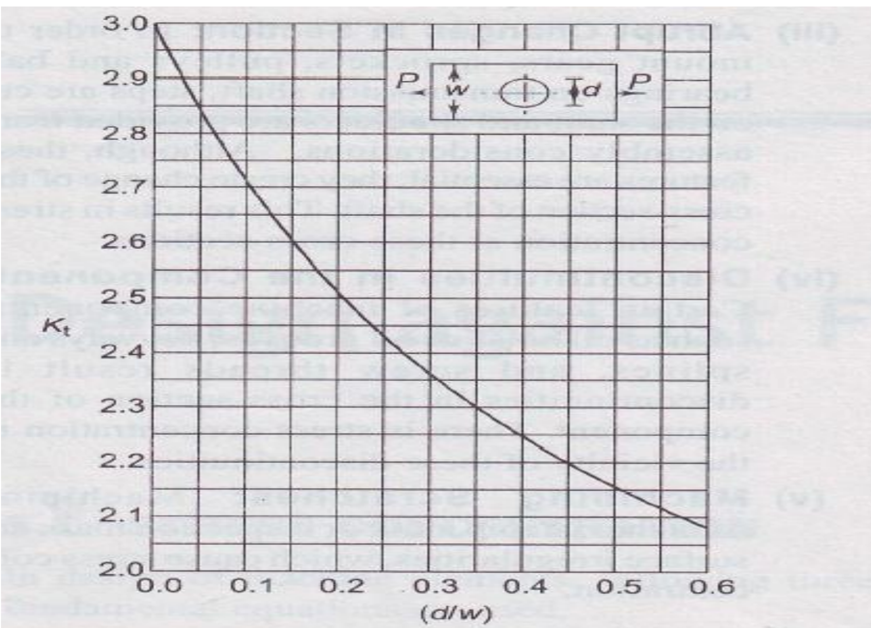


Fig. 5.2 *Stress Concentration Factor (Rectangular Plate with Transverse Hole in Tension or Compression)*

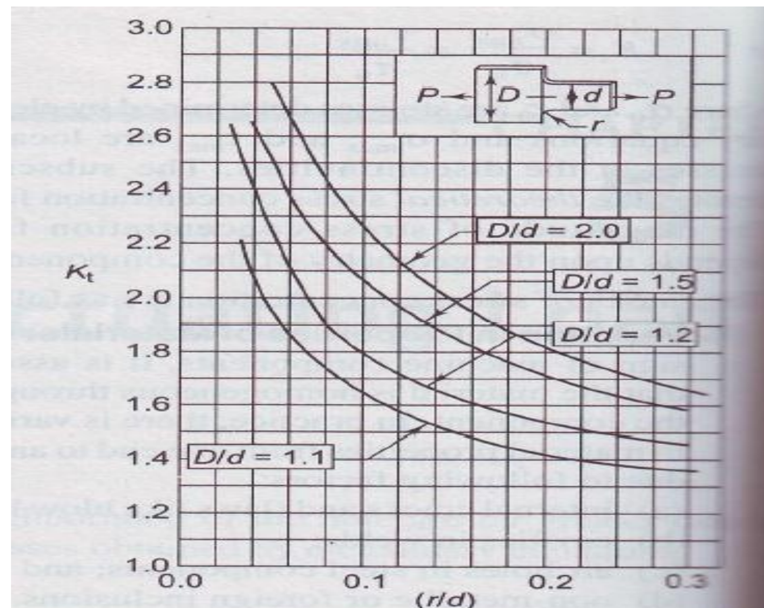


Fig. 5.3 *Stress Concentration Factor (Flat Plate with Shoulder Fillet in Tension or Compression)*

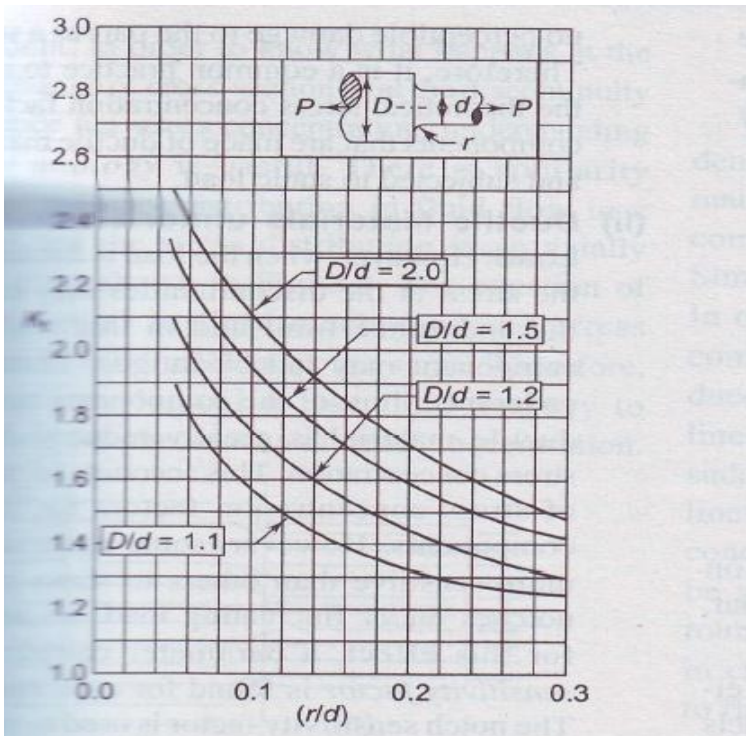
For round shaft with shoulder fillet

Nominal stress for tensile force

$$\sigma_0 = \frac{P}{\left(\frac{\pi}{4} d^2\right)}$$

Nominal stress for bending moment

$$\sigma_0 = \frac{M_b y}{I}$$



5.4 Stress Concentration Factor (Round Shaft with Shoulder Fillet in Tension)

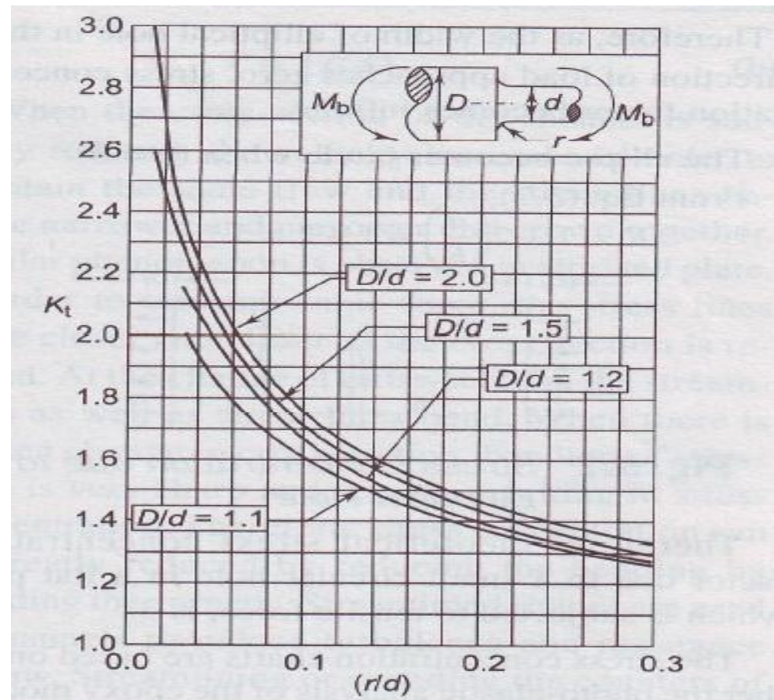
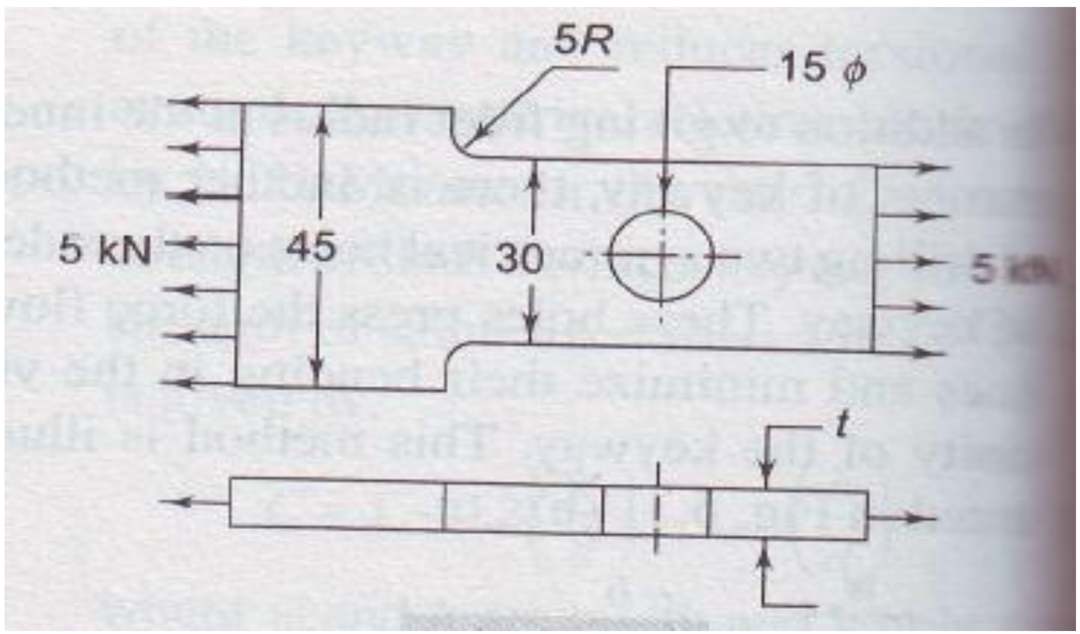
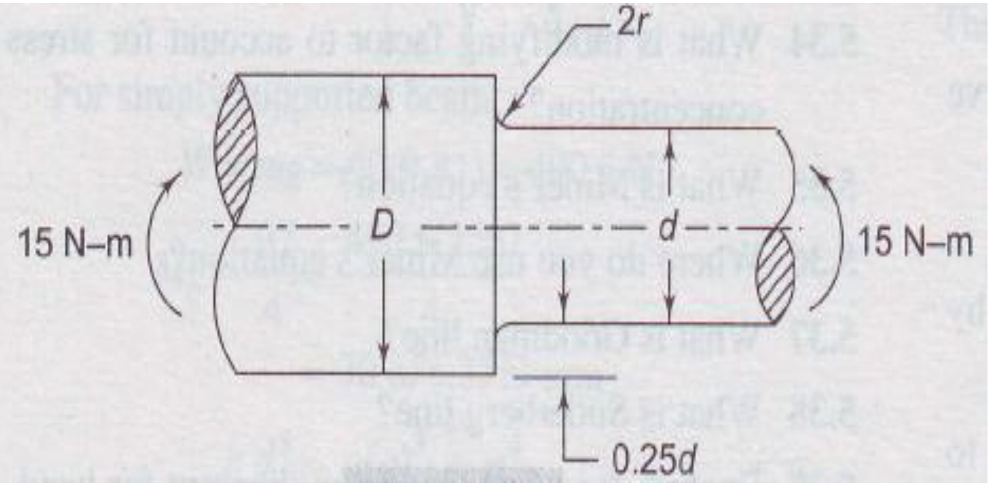


Fig. 5.5 Stress Concentration Factor (Round Shaft with Shoulder Fillet in Bending)

Problem 1 A flat plate subjected to a tensile force of 5 kN as shown in fig. The plate material is grey cast iron FG 200 and the factor of safety is 2.5. Determine the thickness of the plate. (Nov. –Dec. 2009)



Problem 2 A round shaft made of a brittle material and subjected to a bending moment of 15 N-m. The stress concentration factor at the fillet is 1.5 and the ultimate tensile strength of the shaft material is 200 N/mm^2. Determine the diameter d, the magnitude of the stress at the fillet and the factor of safety.

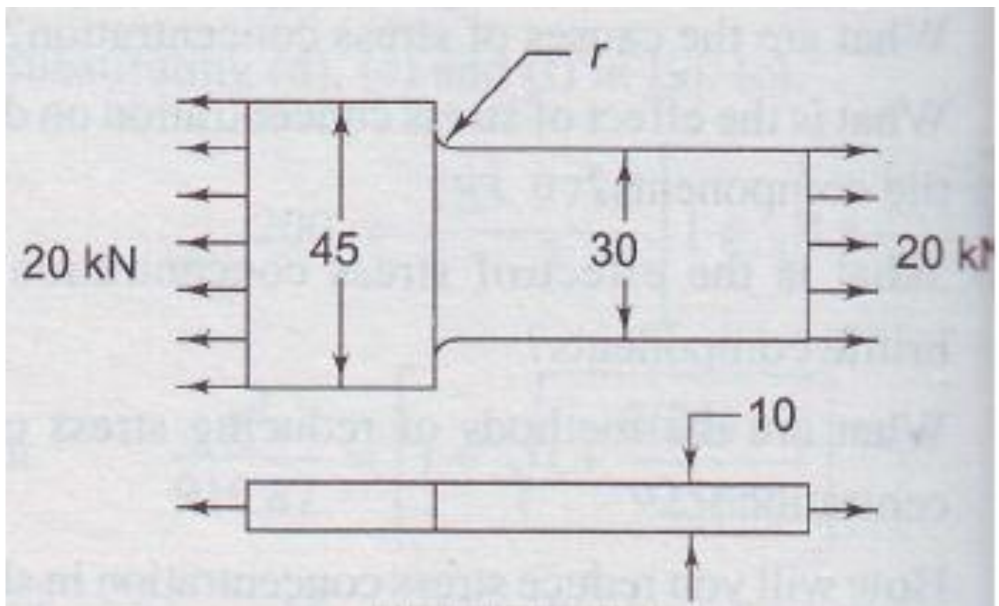


Problem 3 A Non-rotating shaft supporting a load of 2.5 kN is shown in fig. The shaft is made of brittle material with an ultimate tensile strength of 300 N/mm^2 . The factor of safety is 3. Determine the dimensions of the shaft

(Nov- Dec 2010)

Problem 4 A rectangular plat, 15 mm thick made of a brittle material is shown in fig. Calculate the stresses at each of three holes of 3, 5 and 10 mm diameter. (Apr- May 2011)

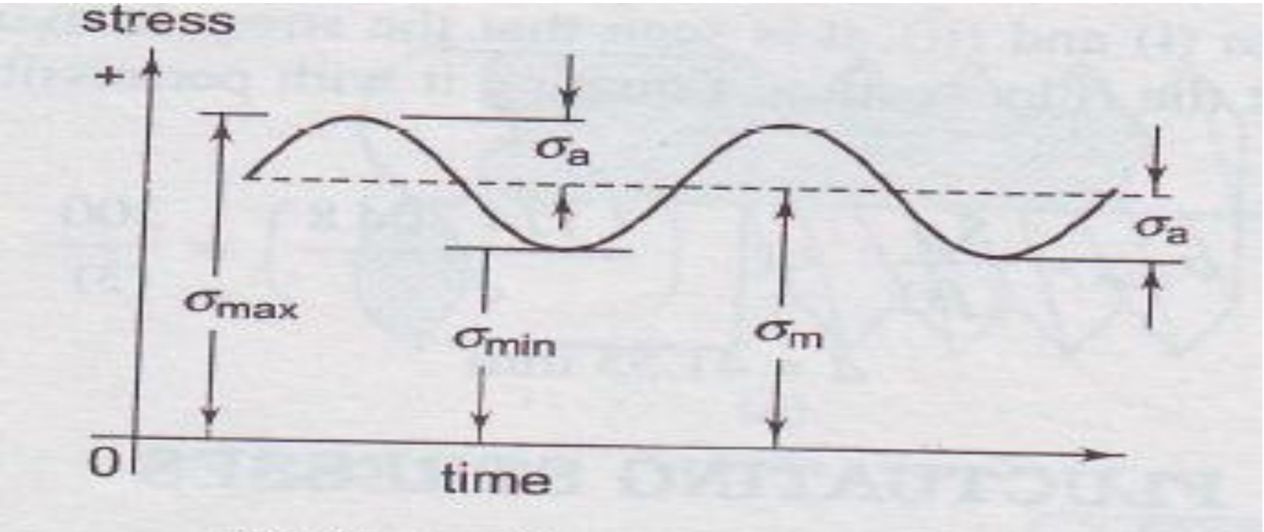
Problem 5 A plate, 10 mm thick, subjected to a tensile load of 20 kN is shown in fig below. The plate is made of cast iron (S_{ut}=350 N/mm^2) and the factor of safety is 2.5. Determine the fillet radius.



Fluctuating stresses

- External force vary in magnitude with respect to time is called fluctuating load and stresses induced due to these loads are called fluctuating stresses.
- 80 % failure in machine component occurs because of fatigue failure.
- Types of mathematical models for cyclic stress:-
  1. fluctuating or alternating stresses
  2. Repeated stresses
  3. Reversed stresses

1. Fluctuating or alternating stress

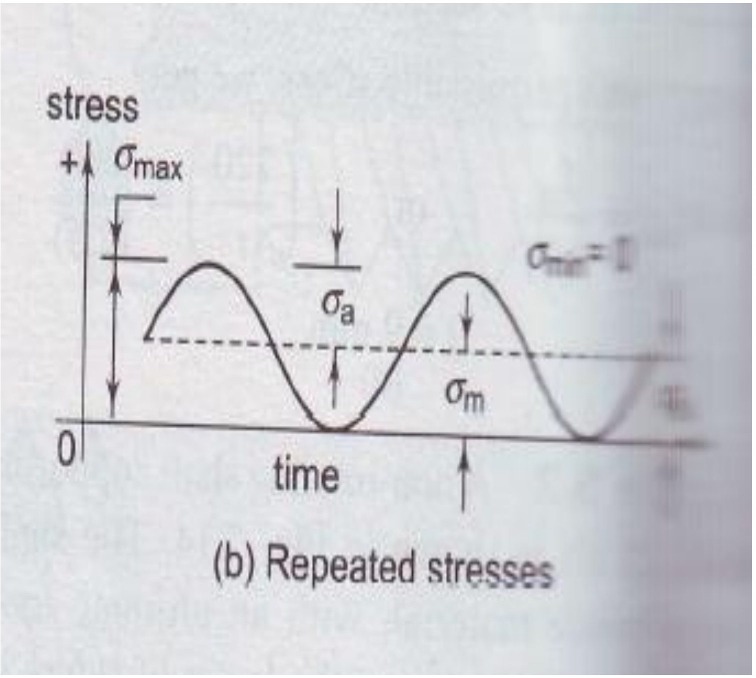


(a) Fluctuating stresses

$$\sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min})$$

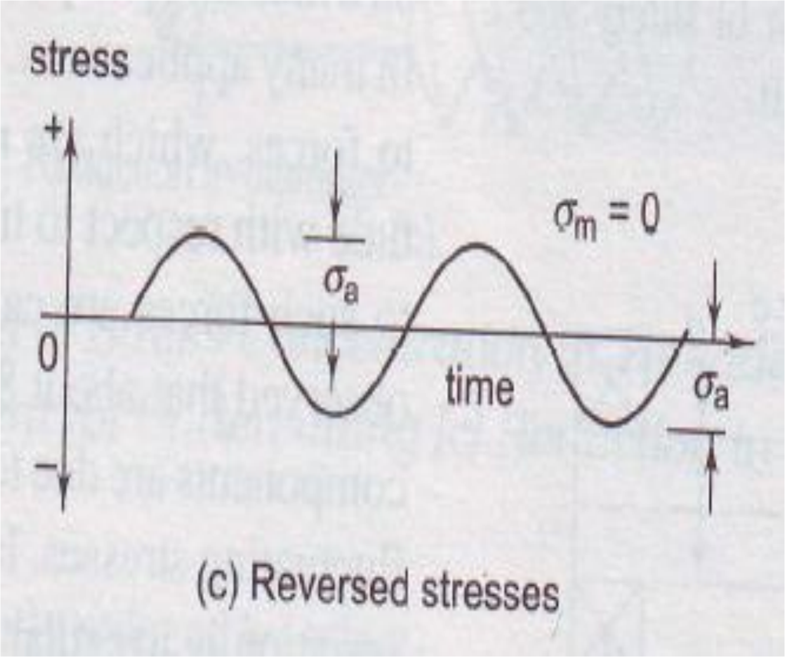
$$\sigma_u = \frac{1}{2}(\sigma_{max} - \sigma_{min})$$

2. Repeated stress



(b) Repeated stresses

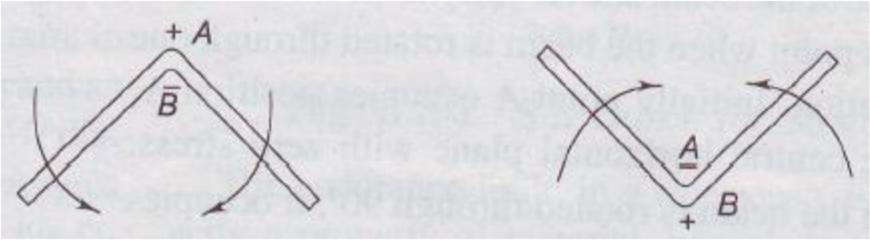
3. Reversed stress



(c) Reversed stresses

Fatigue failure

- Fatigue failure is defined as time delayed fracture under cyclic loading.
- Best example is bending and unbending of a wire.



- Other examples are transmission shafts, connecting rods, gears, vehicle suspension springs and ball bearings.

- Cracks occurs generally in the

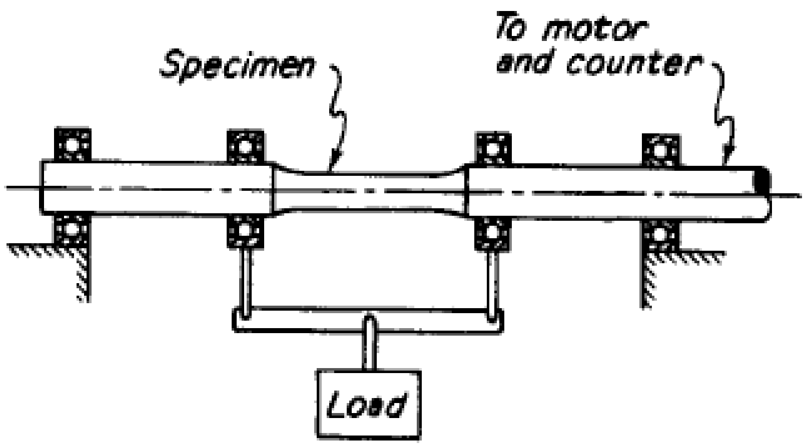
➢ Regions of discontinuity (keyways, oil holes etc)
➢ Regions of irregularities in machining operations (scratches, stamp mark, inspection mark etc)
➢ Internal cracks due to defect in materials like blow holes.

Endurance limit

- The fatigue or endurance limit of a material is defined as the maximum amplitude of completely reversed stress that the standard specimen can sustain for an unlimited number of cycles without fatigue failure. (10^6 cycles)

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

Rotating beam machine subjected to bending moment developed by R.R. Moore



(a) Machine for applying uniform bending moment to specimen

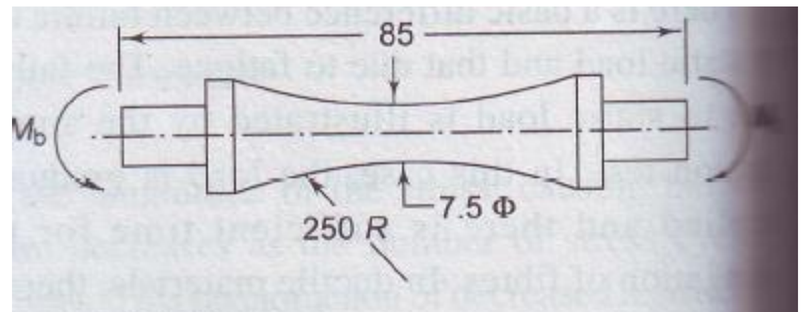


Fig. 5.17 *Specimen for Fatigue Test*

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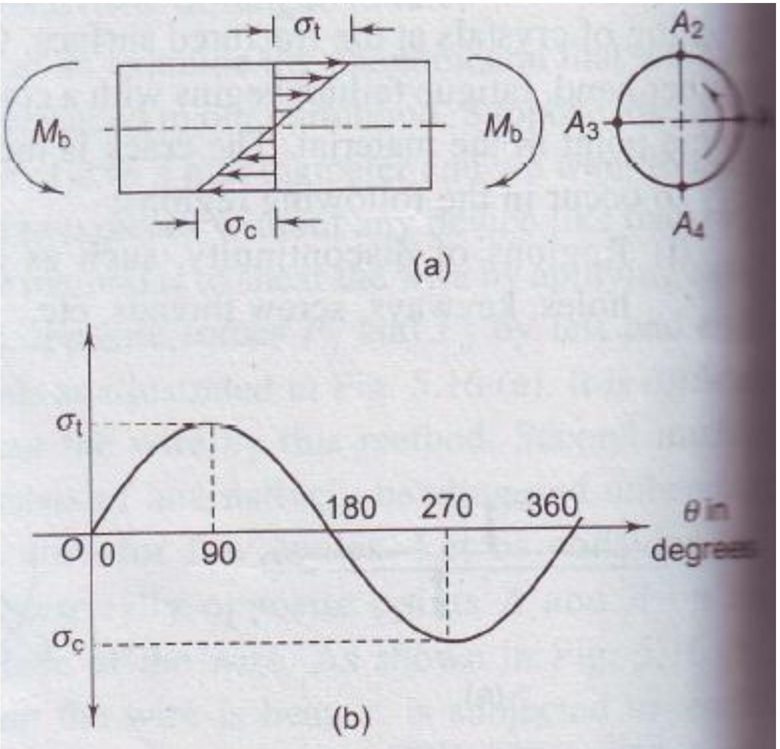
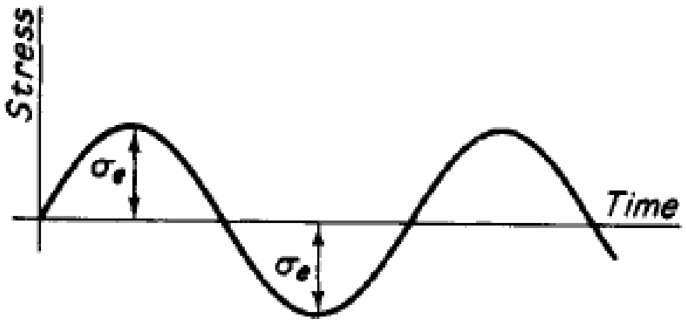


Fig. 5.18 *Rotating Beam Subjected to Bending Moment (a) Beam (b) Stress Cycle at Point A*



(b) Endurance limit stress is highest value of completely reversed bending stress for continuous operation

S-N curve

- The S-N curve is the graphical representation of stress amplitude (S_f) verses the number of stress cycles (N) before the fatigue failure on a log-log graph paper.

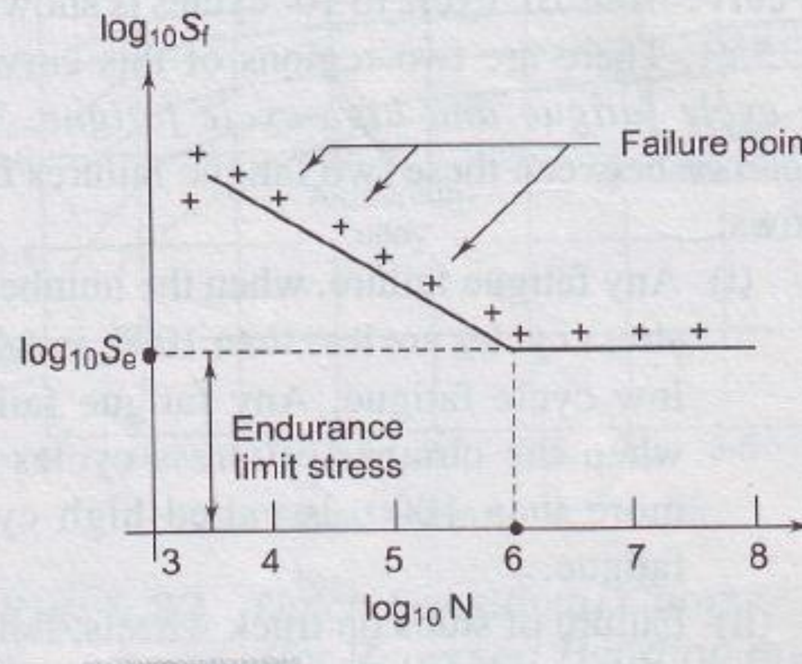


Fig. 5.20 *S-N Curve for Steels*

Notch sensitivity

- Fatigue stress concentration factor (K_f) is defined as the ratio of endurance limit of the notch free specimen to the endurance limit of the notched specimen.

- It is applicable to actual material and depends upon the grain size of the material.

- Notch sensitivity of a material is a measure of how much sensitive a material is to notches or geometric discontinuities.

- Notch sensitivity factor q is defined as the ratio of increase of actual stress over nominal stress to the increase of theoretical stress over nominal stress.

Relationship between K_f, K_t and q

$$\boxed{K_f = 1 + q(K_t - 1)}$$

I. When the material has no sensitivity to notches,

$$\boxed{q=0 \quad \text{and} \quad K_f = 1}$$

II. When the material is fully sensitive to notches,

$$\boxed{q=1 \quad \text{and} \quad K_f = K_t}$$

Estimation of endurance limit

- S_e' = endurance limit stress of a rotating beam specimen subjected to reversed bending stress (N/mm^2)

- S_e = endurance limit stress of a particular mechanical component subjected to reversed bending stress (N/mm^2)

- For steels $S_e' = 0.5\ S_{ut}$
- For cast iron $S_e' = 0.4\ S_{ut}$
- For wrought aluminum alloys $S_e' = 0.4\ S_{ut}$
- For cast aluminum alloys $S_e' = 0.3\ S_{ut}$

Factors affecting endurance limit

$$\boxed{S_e = K_a\, K_b\, K_c\, K_d\, S_e'}$$

where

K_a= surface finish factor

K_b= size factor

K_c= reliability factor

K_d= modifying factor to account for stress concentration factor.

Surface finish factor (K_a)

- Surface finish factor for steel only:-

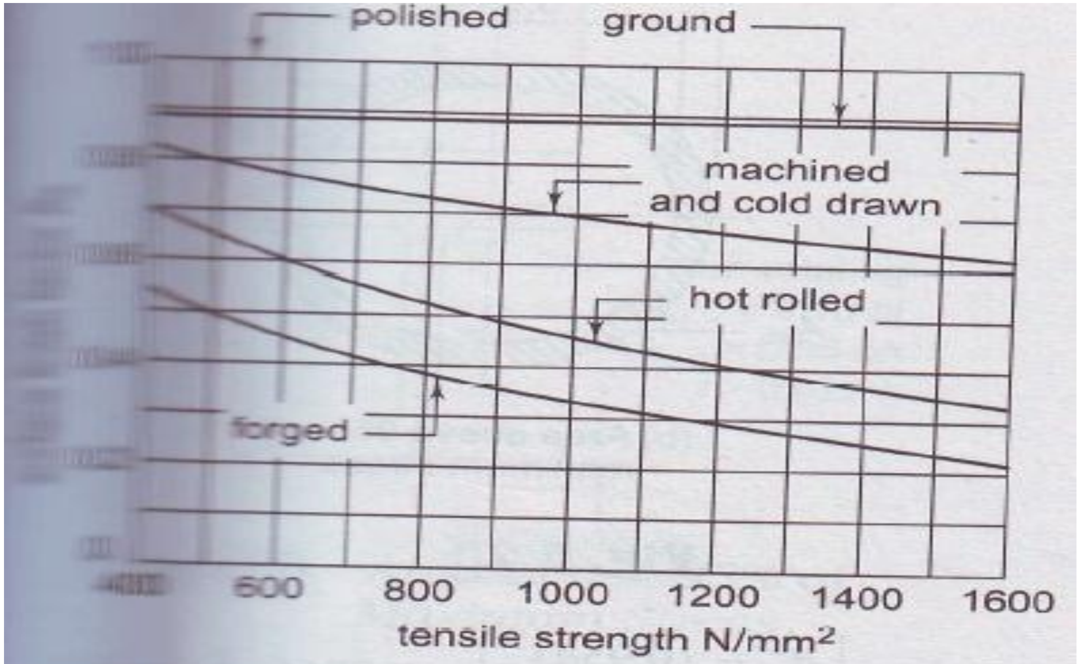


Fig. 5.24 *Surface Finish Factor*

- For cast iron parts K_a=1

Size factor (K_b)

Diameter (d) (mm)	K_b
$d \leq 7.5$	1.00
$7.5 \leq d \leq 50$	0.85
$d > 50$	0.75

Reliability factor (K_c)

Reliability R (%)	K_c
50	1.000
90	0.897
95	0.868
99	0.814

Modifying factor (Kd)

$$\boxed{K_d = 1/K_f}$$

Endurance limit of a component subjected to torsional shear stress is obtained from endurance limit in reversed bending by the following equation:-

1. According to maximum shear stress theory, $S_{se} = 0.5S_e$
2. According to distortion energy theory, $S_{se} = 0.577S_e$.

Endurance limit of a component subjected to axial loading is obtained by

$$(S_e)_a = 0.8\ S_e$$

Types of problem in fatigue design:-

1. Components subjected to completely reversed stresses, and
2. Components subjected to fluctuating stresses.

These design problems are further divided into two groups-

a. Design for infinite life, and
b. Design for finite life.

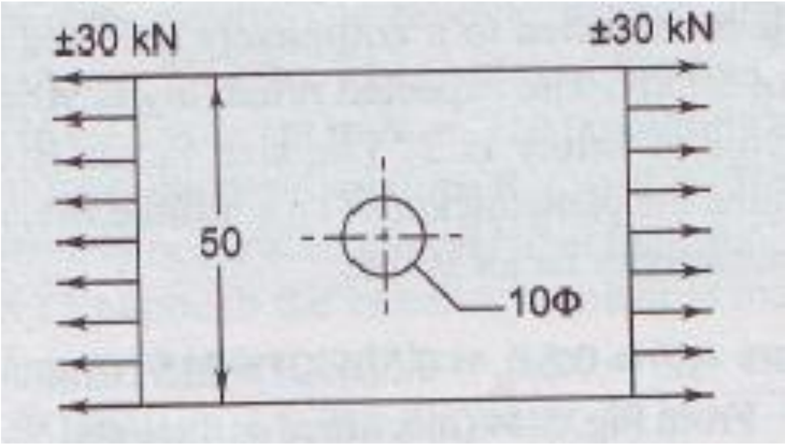
1. Reversed stress- design for infinite life

- Endurance limit becomes criteria of failure.
- Stress amplitude should be lower than the endurance limit in order to withstand the infinite number of cycles.
- Such components are designed with the help of the following equations:-

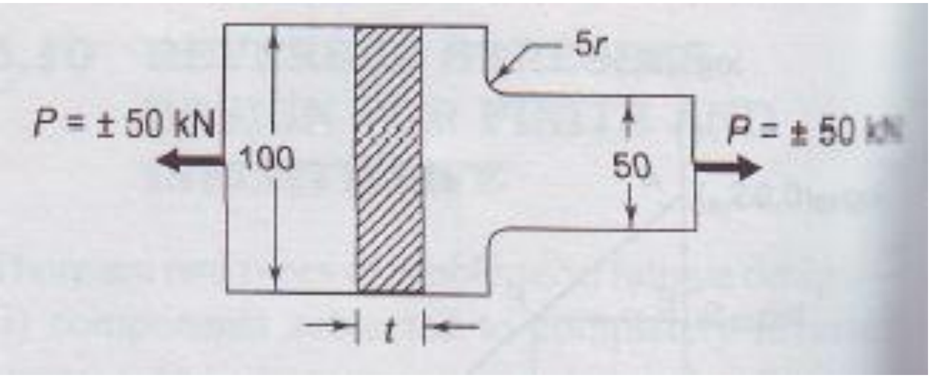
$$\sigma_a = \frac{S_e}{(fs)}$$

$$\tau_a = \frac{S_{se}}{(fs)}$$

Problem 1 A plate made of steel 20C8 (S_{ut}=440 N/mm^2) in hot rolled and normalized condition is shown in fig below. It is subjected to a completely reversed axial load of 30 kN. The notch sensitivity factor q can be taken as 0.8 and the expected reliability is 90%. The factor of safety is 2. The size factor can be taken as 0.85. determine the plate thickness for infinite life. (2007,2008)



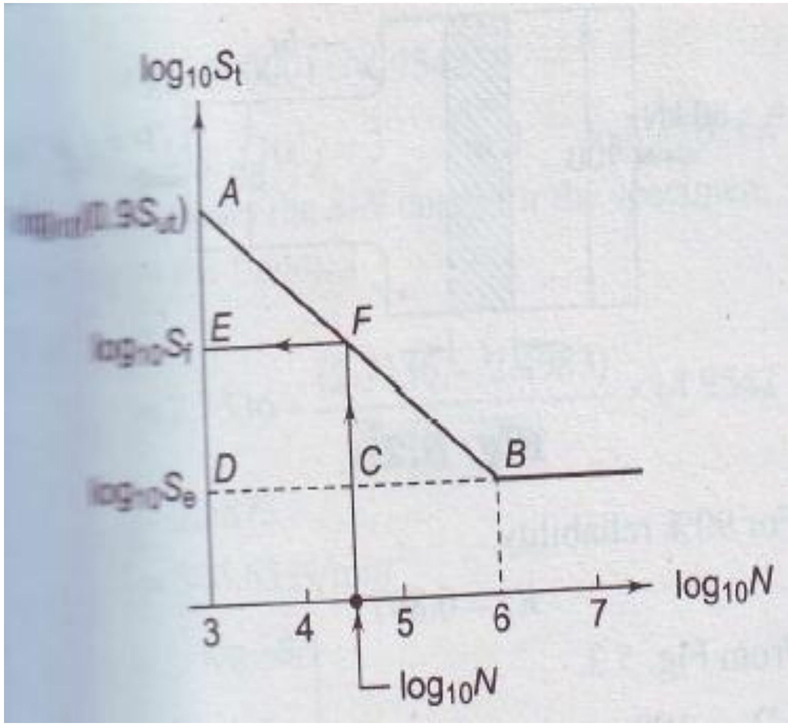
Problem 2 a component machined from a plate made of steel 45C8 (S_{ut}=630N/mm^2) is shown in fig below. It is subjected to a completely reversed axial force of 50 kN. The expected reliability is 90% and the factor of safety is 2. The size factor is 0.85. Determine the plate thickness t for infinite life, if notch sensitivity factor is 0.8. (2006)



2. Reversed stresses –design for finite life

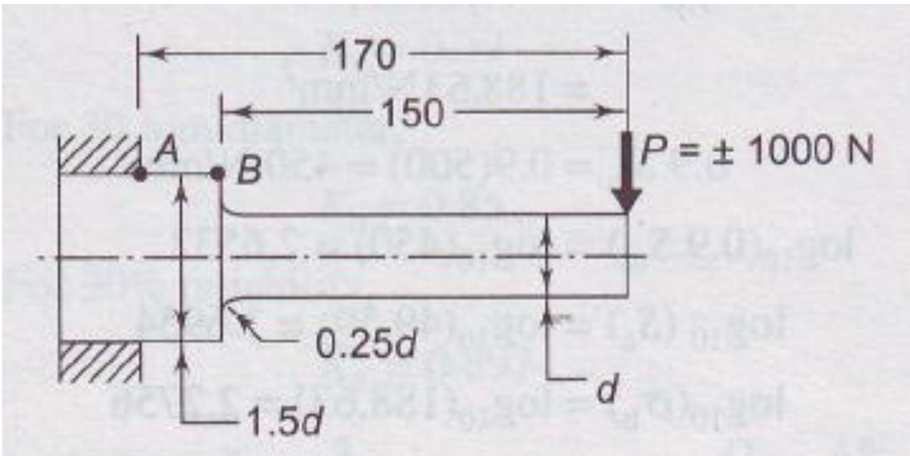
- S-N curve is used
- Design procedure for such problems are as follows:-

1. Locate point A with coordinates $[3, \log_{10}(0.9S_{ut})]$
2. Locate point B with coordinates $[6, \log_{10}(S_e)]$
3. Join AB which is used as a criterion of failure for finite life problems.
4. Depending upon life N draw a vertical line from $\log_{10}N$, will intersect AB at F.
5. Draw a line FE parallel to the abscissa. The ordinate at point E, i.e. $\log_{10}(S_f)$ gives the fatigue strength corresponding to N cycles.



Problem 1 A rotating bar made of steel 45C8 (S_{ut} =630 N/mm^2) is subjected to a completely reversed bending stress. The corrected endurance limit of the bar is 315 N/mm^2 . Calculate the fatigue strength of the bar for a life of 90,000 cyvles. (2011)

Problem 2 A cantilever beam made of cold drawn steel 20C8 (S_{ut}=540N/mm^2) is subjected to a completely reversed load of 1000 N as shown in fig. The notch sensitivity factor q at the fillet can be taken as 0.85 and the expected reliability is 90%. Determine the diameter d of the beam for a life of 10000 cycles. (2006,2010)



Cumulative damage in fatigue

- When the mechanical component is subjected to different stress levels for different parts of the work cycle e.g.

A component is subjected to completely reversed bending stress:-

1. σ_1 for n_1 cycles
2. σ_2 for n_2 cycles
3. σ_3 for n_3 cycles
4. σ_4 for n_4 cycles

And N_1, N_2, N_3, N_4 are the no. of stress cycles before fatigue failure for stresses σ_1, σ_2, σ_3, σ_4 individually applied to component

$$\boxed{\frac{n_1}{N_1}+\frac{n_2}{N_2}+\frac{n_3}{N_3}+\frac{n_4}{N_4}=1}$$

or

$$\boxed{\frac{\alpha_1}{N_1}+\frac{\alpha_2}{N_2}+\frac{\alpha_3}{N_3}+\frac{\alpha_4}{N_4}=\frac{1}{N}}$$

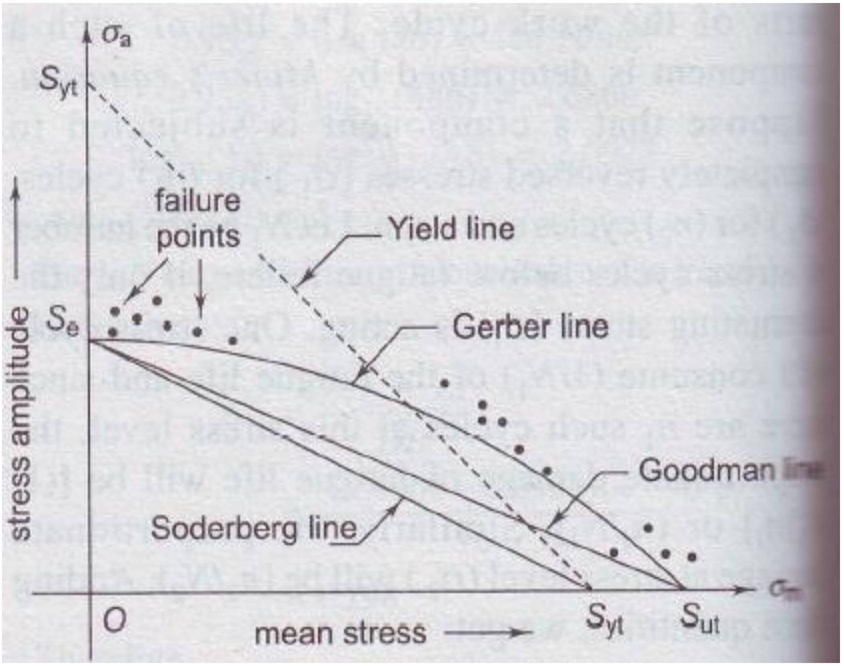
Where α_1, α_2, α_3, α_4 are proportionate of total life consumed by stresses σ_1, σ_2, σ_3, σ_4

Problem 1. The work cycle of a mechanical component subjected to completely reversed bending stress consists of the following three elements:

1. ± 350 N/mm^2 for 85 % of the time.
2. ± 400 N/mm^2 for 12 % of the time, and
3. ± 500 N/mm^2 for 3 % of the time.

The material for the component is 50C4 (S_{ut}=660 N/mm^2) and the corrected endurance limit of the component is 280 N/mm^2. Determine the life of the component. (2007)

3. fluctuating stress- design for infinite life



Problem is solved by using equation of-

1.Gerber line- a parabolic curve joining S_e to S_{ut}

2. Soderberg line- a straight line joining S_e to S_{yt}

3. Goodman line- a straight line joining S_e to S_{ut}

- Equation for Soderberg line-

$$\frac{S_m}{S_{yt}} + \frac{S_a}{S_e} = 1$$

- Equation for Goodman line-

$$\frac{S_m}{S_{ut}} + \frac{S_a}{S_e} = 1$$

- Equation for Gerber line-

$$\frac{S_a}{S_e} + \left(\frac{S_m}{S_{ut}}\right)^2 = 1$$

Where $\sigma_a = \frac{S_a}{fs}$ *and* $\sigma_m = \frac{S_m}{fs}$

Modified Goodman diagram

Goodman diagram is modified by combining fatigue failure with Failure by yielding.

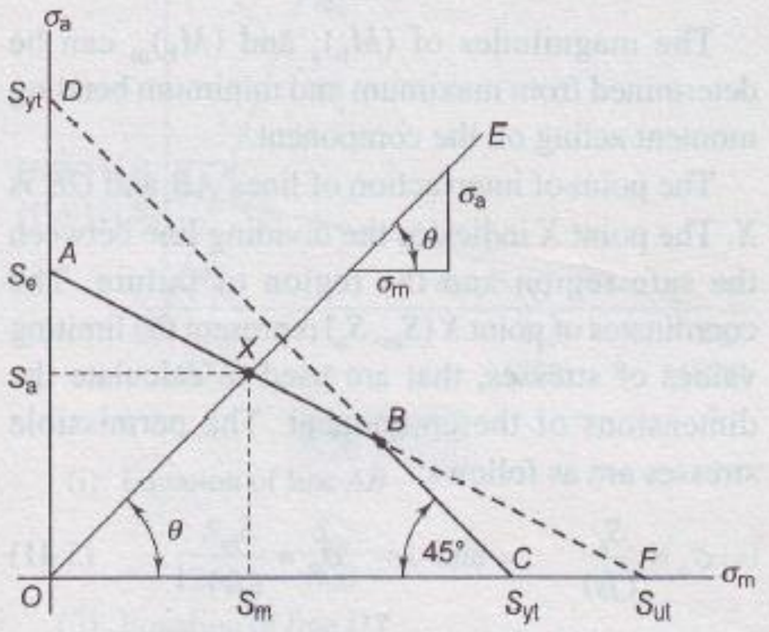


Fig. 5.40 *Modified Goodman Diagram for Axial and Bending Stresses*

While solving a problem, a line $\overline{OE}$ with slope $\tan\theta$ is constructed in such a way that,

$$\tan\theta = \frac{\sigma_a}{\sigma_m} \quad (5.38)$$

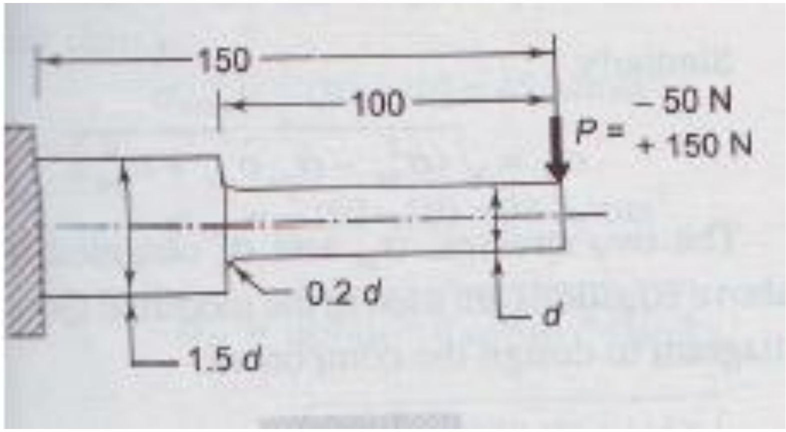
Since $$\frac{\sigma_a}{\sigma_m} = \frac{(P_a/A)}{(P_m/A)} = \frac{P_a}{P_m}$$

$$\therefore \quad \tan\theta = \frac{P_a}{P_m} \quad (5.39)$$

Problem 1. A machine component is subjected to fluctuating stress that varies from 40 to 100 N/mm^2. The corrected endurance limit stress for the machine component is 270 N/mm^2. The ultimate tensile strength and yield strength of the material are 600 and 450 N/mm^2 respectively. Find the factor of safety using:

i. Gerber theory
ii. Soderberg line
iii. Goodman line.

Problem 2. A cantilever beam made of cold drawn steel 40C8 (S_{ut}=600N/mm^2 and S_{yt}=380 N/mm^2) is shown in fig below. The force P acting at the free end varies from -50 N to +150 N. The expected reliability is 90% and the factor of safety is 2. The notch sensitivity factor at the fillet is 0.9. Determine the diameter of the beam at the fillet cross-section. (2009, 2008)



Fatigue design under combined stresses

- In case of two dimensional stresses

$$\sigma = \sqrt{\left(\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2\right)}$$

- The mean and alternating stress in case of 2-D stress-

$$\sigma_m = \sqrt{\left(\sigma_{xm}^2 - \sigma_{xm}\sigma_{ym} + \sigma_{ym}^2\right)}$$

$$\sigma_a = \sqrt{\left(\sigma_{xa}^2 - \sigma_{xa}\sigma_{ya} + \sigma_{ya}^2\right)}$$

- In case of bending and torsional moment

$$\sigma_a = \sqrt{\left(\sigma_{xa}^2 + 3\tau_{xya}^2\right)}$$

$$\sigma_m = \sqrt{\left(\sigma_{xm}^2 + 3\tau_{xym}^2\right)}$$

Problem 1 A machine component is subjected to two dimensional stresses. The tensile stress in the x-direction varies from 40 to 100 N/mm^2 while the tensile stress in the y-direction varies from 10 to 80 N/mm^2. The frequency of variation of these stresses is equal. The corrected endurance limit of the component is 270 N/mm^2. The ultimate tensile strength of the material of component is 660 N/mm^2. Determine the factor of safety used by the designer. (2006,2004)

Problem 2 A transmission shaft carries a pulley midway between the two bearings. The bending moment at the pulley varies from 200 N-m to 600 N-m, as the torsional moment in the shaft varies from 70 N-m to 200 N-m. The frequencies of variation of bending and torsional moments are equal to the shaft speed. The shaft is made of steel FeE400 (S_{ut}=540 N/mm^2 and S_{yt}=400 N/mm^2). The corrected endurance limit of the shaft is 200 N/mm^2. Determine the diameter of the shaft using a factor of safety of 2. (2006)

Chain Drives

- A chain drive consists of an endless chain wrapped around two sprockets
- The chain consists of a number of chain link connected by pin joints.
- Sprockets are toothed wheels with some special profile of teeth.

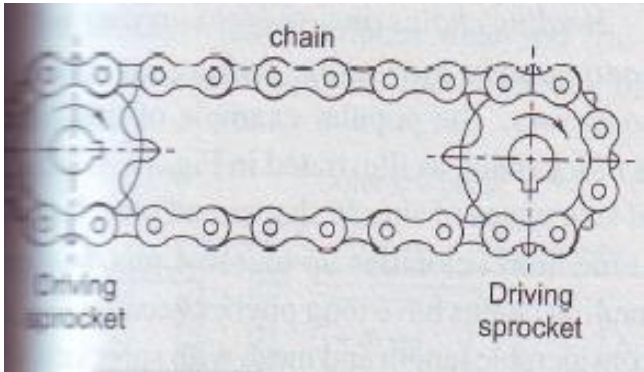


Fig. 14.1 *Chain Drive*

- Chain drives has some features of belt drive and some features of gear drive.
- Chain drive is a positive drive.
- Chain drives are used for V.R. 10:1, chain velocities up to 25m/s and power transmission up to 100 kW.

Roller chain

- It consists of alternate links of alternate links made of inner and outer link plates.
- Roller chain consists of five parts:-

i. Pin
ii. Bush
iii. Roller
iv. Inner link plate
v. Outer link plate

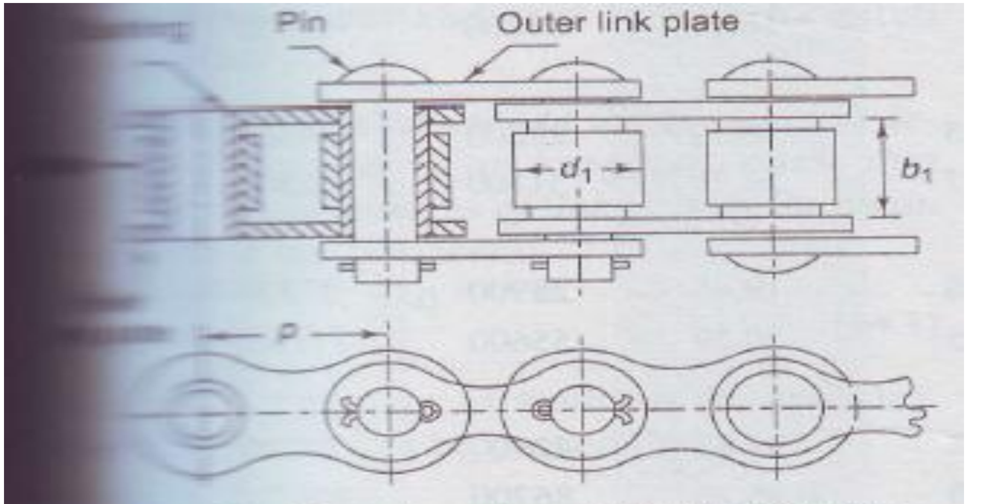


Fig. 14.5 Construction of Roller Chain

- Pitch (p) of the chain is the linear distance between the axes of adjacent rollers.

Classification of roller chains

- Roller chains are standardized and manufactured on the basis of the pitch.
- These chains are single strand or multi strand such as simplex, duplex or triplex chains.

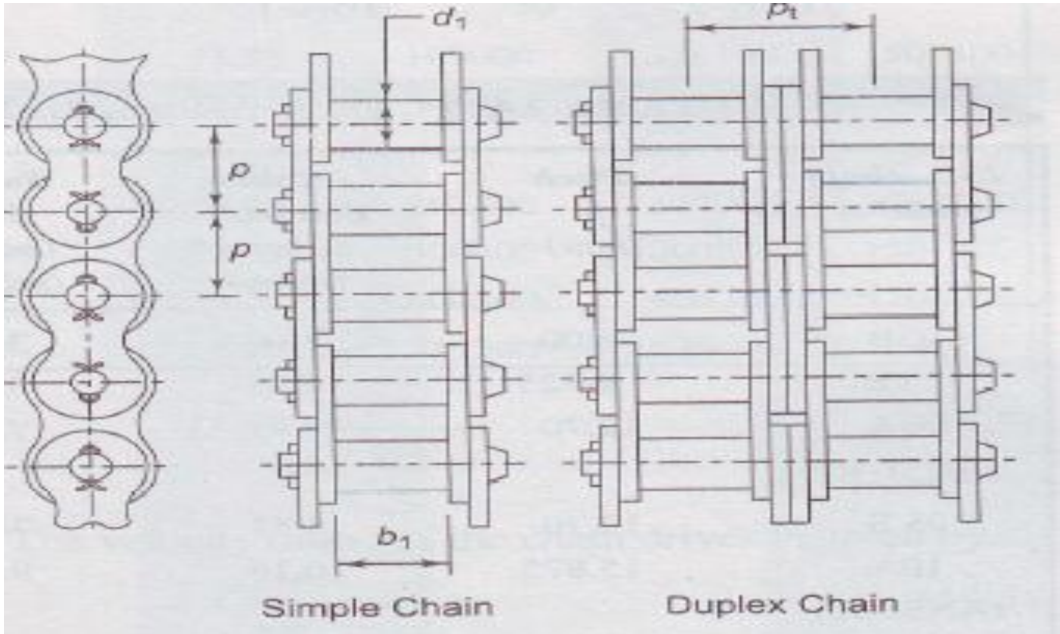


Fig. 14.6 *Simple and Duplex Chains*

Designation of roller chains

- Roller chains are designated on the basis of pitch.

1. Chain number consists of two parts –a number followed by a letter (e.g. 08B or 16A). The number in two digits express the pitch in sixteenth of an inch. The letter A means American standard ANSI series and letter B means British standard series.
2. The chain number is supplemented by a hyphenated suffix 1 for simplex chain, 2 for duplex and 3 for triplex chain. (e.g. 08B-2, 16A-1)
3. Breaking load is defined as the maximum tensile load at which failure of chain occurred.

A. Roller Chains

Chain No. ISO/DIN	Chain No. Roion	Pitch P mm	Roller dia max D_r mm	Width between Inner Plates min W mm	Pin Body dia max D_p mm	Plate Depth max G mm	Transverse pitch P_t mm	Overall over joint max A_1, A_2 A_3 mm	Bearing Area cm^2	Weight per Meter kgf	Breaking Load min kgf
04-1	R628	6.0	4.00	2.80	1.85	5.45	—	8.00	0.07	0.12	300
05B-1	R830	8.0	5.00	3.10	2.31	7.05	—	11.10	0.11	0.18	460
05B-2	DR830	8.0	5.00	3.10	2.31	7.05	5.64	15.80	0.22	0.33	800
05B-3	TR830	8.0	5.00	3.10	2.31	7.05	5.64	21.50	0.33	0.52	1140
	R940	9.525	6.35	4.00	3.28	8.15	—	14.40	0.22	0.37	900
06B-1	R957	9.525	6.35	5.90	3.28	8.15	—	16.40	0.28	0.41	910
06B-2	DR957	9.525	6.35	5.90	3.28	8.15	10.24	26.60	0.56	0.77	1730
06B-3	TR957	9.525	6.35	5.90	3.28	8.15	10.24	36.70	0.84	1.09	2540
082-1	R1224	12.7	7.75	2.40	3.68	9.90	—	8.2	0.17	0.28	1000
081-1	R1230	12.7	7.75	3.48	3.68	9.90	—	11.00	0.21	0.30	820
	R1230H	12.7	7.75	3.48	3.68	9.90	—	12.10	0.22	0.35	1000
	R1248	12.7	7.75	4.90	3.68	9.90	—	12.80	0.26	0.34	820
084-1	R1248H	12.7	7.75	4.90	4.09	11.10	—	16.30	0.36	0.58	1600
083-1	R1249	12.7	7.75	4.90	4.09	10.25	—	14.40	0.32	0.49	1200
	R1253	12.7	7.75	5.25	3.68	10.35	—	13.80	0.29	0.38	1400
	R1264	12.7	7.77	6.40	3.97	11.70	—	15.90	0.39	0.47	1500
	R1252	12.7	8.51	5.30	4.45	11.70	—	18.10	0.39	0.58	1820
08A-1	R40	12.7	7.95	8.00	3.97	11.70	—	20.90	0.44	0.69	1410
08A-2	DR40	12.7	7.95	8.00	3.97	11.70	14.38	35.30	0.88	1.20	2820
08A-3	TR40	12.7	7.95	8.00	3.97	11.70	14.38	49.70	1.32	1.80	4230
08B-1	R1278	12.7	8.51	8.00	4.45	11.70	—	20.50	0.50	0.70	1820

Geometric Relationships

- The pitch circle diameter (D) of sprocket is defined as the diameter of an imaginary circle that

  passes through the centers of link

  pins as the chain is wrapped on the

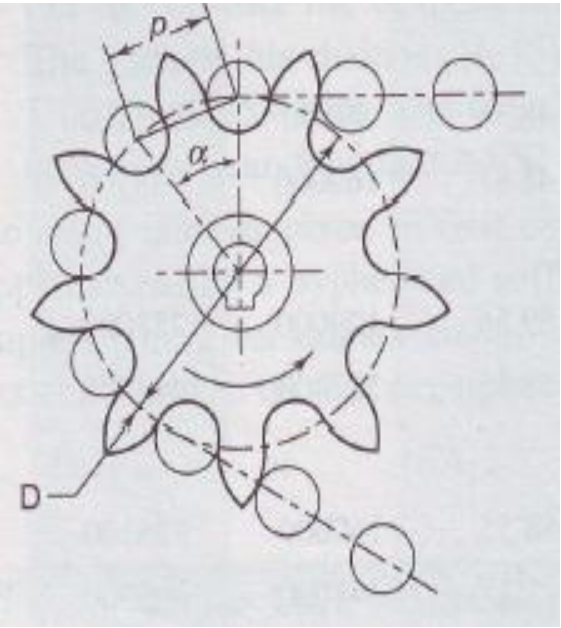
  sprocket.

- Pitch angle $\alpha = \frac{360}{z}$

  where z is no. of teeth on the sprocket.

- pitch circle diameter (D)

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



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- Velocity ratio (i):-

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

where n_1 and n_2 are speeds of rotation of the driving and driven sprockets. And z_1, z_2 are the number of teeth on driving and driven sprockets.

- Length of the chain

$$L = L_n \times p \quad (14$$

where,

L = length of the chain (mm)

L_n = number of links in the chain

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

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right) \quad (14.6)$$

where,

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

z_1 = number of teeth on thc smaller sprocket

z_2 = number of teeth on the larger sprocket

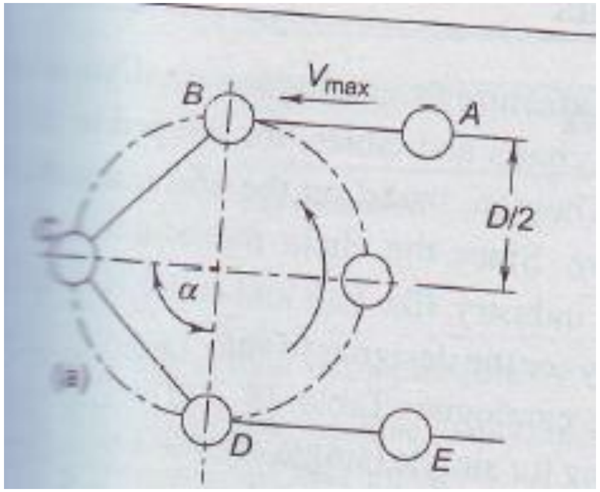
Polygonal effect in chain drive

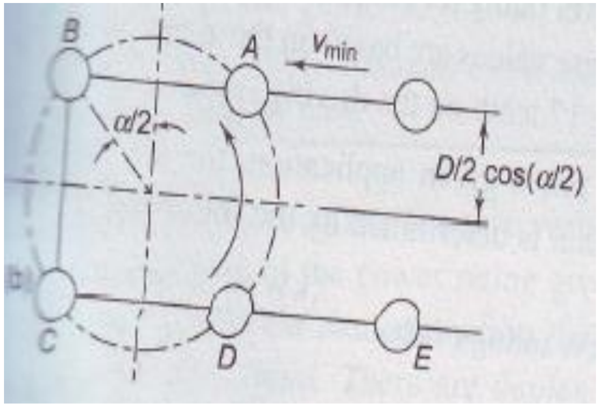
$$\text{linear velocity} \quad v_{max} = \frac{\pi D n}{60 \times 10^3} \, m/s$$

$$\text{linear velocity} \quad v_{min} = \frac{\pi D n \cos\left(\frac{\alpha}{2}\right)}{60 \times 10^3} \, m/s$$

$$\left(v_{max} - v_{min}\right) \propto \left[1 - \cos\left(\frac{\alpha}{2}\right)\right]$$

$$\left(v_{max} - v_{min}\right) \propto \left[1 - \cos\left(\frac{180}{z}\right)\right]$$





Power rating of roller chain

- Power transmitted by the roller chain

$$KW = \frac{P_1 v}{1000}$$

- The power rating of the roller chain is obtained on the basis of four failure criterion

1. Wear
2. Fatigue
3. Impact
4. Galling

- For a given application the kW rating of the chain is determined by

kW rating of chain = (kW to be transmitted) x K_s

where K_s = service factor

Problem 1 It is required to design a chain drive to connect a 15 kW, 1400 rpm electric motor to a transmission shaft running at 350 rpm. The operation involves moderate shocks.

1. specify the no. of teeth on the driving and driven sprockets
2. Select a proper roller chain for the drive.
3. Calculate the pitch circle diameters of driving and driven sprockets.
4. Determine the number of chain links.
5. Specify the correct centre distance between the axes of sprocket.

   During preliminary stages the centre distance can be assumed to be 40 times the pitch of the chain.

(2009)

Problem 2. Design a roller chain drive to connect a 15 kW 1500 rpm electric motor to a centrifugal pump running at 750 rpm. The service conditions involve moderate shocks.

(i) Select a proper roller chain and give a list of its dimensions.

(ii) Determine the pitch circle diameter of driving and driven sprockets.

(iii) Determine the number of chain links.

(iv) Recommend the correct centre distance between the axes of sprockets. **(CSVTU-2007)**

Problem 3. Design a roller chain drive to connect a 10 kW 1500 rpm electric motor to a centrifugal pump running at 750 rpm. The service conditions involve moderate shocks.

(i) Select a proper roller chain and give a list of its dimensions.

(ii) Determine the pitch circle diameter of driving and driven sprockets.

(iii) Determine the number of chain links.

(iv) Specify the correct centre distance. The conditions of operation are constant load, fixed centre distance, continuous lubrication, continuous running.

During preliminary stages the centre distance can be assumed to be 40 times the pitch of the chain. **(2011)**

Problem 4. It is required to design a chain drive to connect a 12 kW, 1400 rpm electric motor to a centrifugal pump running at 700 rpm. The service conditions involve moderate shocks.

1. Select a proper roller chain and give a list of dimensions.
2. Determine the pitch circle diameter of driving and driven sprockets.
3. Determine the number of chain links.
4. Specify the correct centre distance between the axes of sprockets.

Problem 5. It is required to design a chain drive to connect a 5 kW, 1400 rpm electric motor to a drilling machine. The speed reduction is 3:1. The centre distance should be approximately 500 mm.

(i) Select a proper roller chain for the drive.

(ii) Determine the number of chain links.

(iii) Specify the correct centre distance between the axes of sprockets. **(2008)**

Problem 6. A compressor is to be activated from 10 kW electric motor. The rpm of motor shaft is 1000 rpm and that of the compressor is 350 rpm. The minimum centre distance is 500 mm. The compressor operates 16 hrs day. Design a suitable chain drive. (2009)