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IES | GATE | PSU

MACHINE DESIGN

For

MECHANICAL ENGINEERING

MACHINE DESIGN

SYLLABUS

Design of static and dynamic loading; failure theories; Fatigue strength and the S-N diagram; Principles of the design of machine elements such as bolted, riveted and welded joints, shafts, spur gears, rolling and sliding contact bearings, breaks and clutches.

ANALYSIS OF GATE PAPERS

Exam Year	1 Mark Ques.	2 Mark Ques.	Total
2003	1	4	9
2004	3	2	7
2005	2	2	6
2006	-	4	8
2007	1	8	17
2008	-	7	14
2009	-	3	6
2010	-	3	6
2011	-	1	2
2012	-	2	4
2013	3	5	13
2014 Set-1	1	2	5
2014 Set-2	-	8	8
2014 Set-3	2	2	6
2014 Set-4	2	3	8
2015 Set-1	1	2	5
2015 Set-2	1	-	1
2015 Set-3	1	2	5
2016 Set-1	-	1	2
2016 Set-2	2	-	2
2016 Set-3	-	1	2

CONTENTS

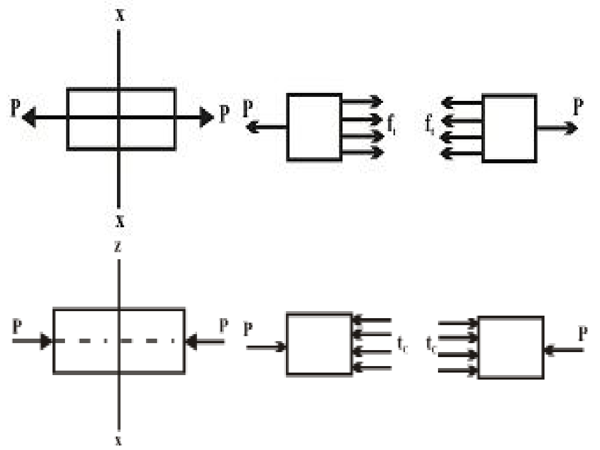
Topics **Page No**

1 STRESS IN MACHINE PARTS

1.1 SIMPLE STRESSES IN MACHINE PARTS

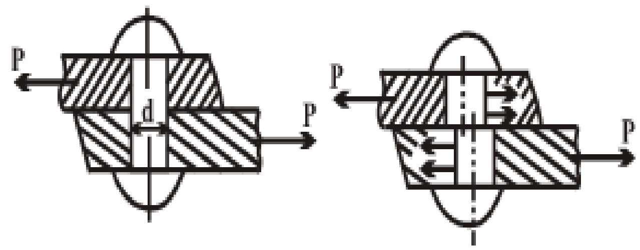
- The machine parts are subjected to various forces which may be due to either which may be due to either one or more of the following:-
  1) Energy transmitted
  2) Weight of Machine
  3) Frictional resistance
  4) Inertia of reciprocating parts
  5) Change of temperature
  6) Lack of balance of moving parts

- The following types of the load are important from the design point of view-
  1) Dead or steady load – A load is said to be dead or steady load, when it does not change in magnitude or direction.
  2) Live or variable load – A load is said to be a love or variable load, when it changes continually.
  3) Suddenly applied or shock loads – A load so said to be a suddenly applied or shock load, when it is suddenly applied or removed.
  4) Impact load – A load is said to be an impact load, when it is applied with some initial velocity.



- When a body is subjected to two equal and opposite axial pulls P (also called tensile load) then the stress induced at any section of the body is known as tensile stress.
  The ratio of the increases in length to the original length as tensile strain.
  Tensile stress f_t = P/A and tensile strain = e_c
  $= \frac{\delta l}{l}$
  and young's Modulus; $E = \frac{f}{c} = \frac{P.l}{A.\delta l}$
  P = Tensile or compressive force acting on the body.
  A = Cross – sectional area of the body
  l = original length
  δl = Change in length

- When a body is subjected to two equal and opposite forces, acting tangentially across the resisting section, as a result of which the body tends to shear off the section, then the stress produced is called shear stress corresponding strain is known as shear strain and it is measured by the angular deformation accompanying the shear stress.
  Shear stress, $f_s = \frac{\text{Tangential force}}{\text{Resisting area}}$

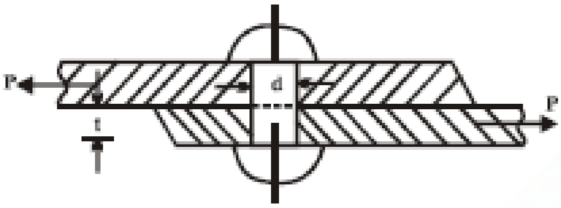


- All lap joints and single cover butt jo9ints are in single shear, while the butt joints with double cover plates are in double shear.
- In case of shear, the area involved is parallel to the external force applied.

- A localized compressive stress at the surface of contact between two members of a machine part, that are relatively at rest is known as bearing stress or crushing stress. The bearing stress is taken into account in the design of riveted joints, cotter joints, knuckle joints etc. Bearing stress

  $f_b = \frac{P}{d.t.n}$

  d = diameter of rivet
  t = thickness of the plate
  d. t = projected area of the rivet
  n = number of rivet per pitch length in bearing



- Barba established a law that in tension similar test pieces deforms similarly and two test pieces are said to e similarly if they have the same value of $l/\sqrt{A}$, where l is the gauge length and A is the cross – sectional area.
- When designing machine parts, it is desirable to keep the stress lower that the maximum or ultimate stress at which failure or the material takes place. This stress is known as the working stress or design stress.
- The ratio of the maximum stress to the working stress is known as factor of safety.

  Factor of safety

  $= \frac{\text{Maximum stress}}{\text{Working or design stress}}$

  For ductile materials –
  Factor of safety

  $= \frac{\text{Yield point stress}}{\text{Working or design stress}}$

  For brittle materials – Factor of safety

  $= \frac{\text{Ultimate stress}}{\text{Working or design stress}}$

- The selection of proper factor of safety to be used in designing any machine component depends upon a number of consideration such as the material, mode of manufacture, type of stress, general service conditions and shape of the parts.

1.2 THERMAL STRESSES

Whenever there is some increases or decreases in the temperature of a body, it causes the body to expand or contract. If the body is allowed to expand or contract freely, with the rise or fall of the temperature, no stresses are induced in the body. Such stresses are known as thermal stresses.

Let l = original length of the body
t = change in temperature (increases or decrease)
α = coefficient of thermal expansion
Increases in length = $\delta l = l. \alpha. T$
If ends of body are fixed to rigid supports, then compressive strain.

$e_c = \frac{\delta l}{l} = \alpha . t$

Thermal stress $f_t = e_c, E = \alpha . t. E$

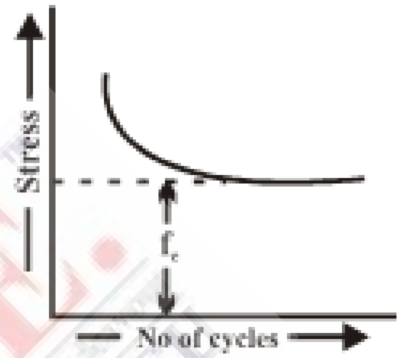
- When a body composed of two of different materials having different coefficient of thermal expansion, then due to the rise in temperature, the material with higher coefficient of thermal expansion will be subjected to compressive stress whereas material with low coefficient of expansion will be subjected to tensile stress.
- When a thin type is shrunk on to a wheel of diameter of is a little less than the wheel diameter. When tyre is heated, its circumference (π d) will increases to (π D). In this condition, it is slipped on to the wheel. When it cools it wants to return to its original circumference (π d), but the wheel, if it assumed to be rigid, prevents it form doing so.

Strain, e = $\frac{\pi D - \pi d}{\pi d} = \frac{D-d}{d}$

- This strain is known as circumferential or hoop strain.
  Circumferential or hope stress,
  $f = e.E = \frac{E(D-d)}{d}$
- When a tensile force act along the length of the bar, the length of the bar is increased by δl and the diameter decreases by an amount δd. The strain in the direction of force is known as linear strain and an opposite kind of strain and an opposite kind of strain in every direction at right angles to it, is known as lateral strain.
- Poisson's Ratio
  $(\mu) = \frac{\text{Lateral strain}}{\text{Linear strain}}$ = Constant
- When a body is subjected to three mutually perpendicular stresses, of equal intensity then the ratio of the direct stress of the corresponding volumetric strain is known as bulk modulus (K).
  K = $\frac{\text{Direct stress}}{\text{Volumetric strain}} = \frac{f}{\delta V / V}$
- Relation between Bulk modulus and young's modulus-
  K = $\frac{E}{3(1-2\mu)}$
- Relation between young's modulus and modulus of rigidity -
  G = $\frac{E}{2(1+\mu)}$

1.3 VARIABLE STRESSES IN MACHINE PARTS

- The stresses which vary from a minimum value to a maximum value of the same nature (i.e. tensile or compressive) are called fluctuating stresses.
- The stresses which vary from one value of compressive to the value of tensile or vice versa, are known as completely reversed or cyclic stresses.
- The stresses which vary from zero to a certain maximum value are called repeated stresses.
- The stresses which vary from a minimum value to a maximum value of the opposite nature (i.e, from a certain minimum compressive to a certain maximum compressive) are called alternating stresses.



- When a material is subjected to repeated stress, it fails at stresses below the yield point stresses. Such type of failure of a material is without any prior indications. The fatigue of material is effected by the size of the component, relative magnitude of static and fluctuating loads and the number of load reversals.
- If the stress is kept below a certain value as shown by dotted line, the material will not fail whatever may be the number of cycles. This stress, as represented by dotted line, is known as endurance or fatigue limit (f_c). If is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite number of cycles (usually 10^7 cycles).
- The term endurance limit is used for reversed bending only while for other types of loading the term endurance strength of the material.
- Mean or average stress-

$$f_m = \frac{f_{max} + f_{min}}{2}$$

- Reversed stress component or variable stress -

$$f_v = \frac{f_{max} - f_{min}}{2}$$

- For repeated loading (f_{min} = 0),

$$f_m = f_v = \frac{f_{max}}{2}$$

1.4 EFFECT OF LOADING ON ENDURANCE LIMIT:-

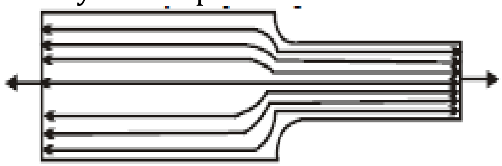
The endurance limit will be different for different types of loading. Let.
K_b = Loads correction factor for reversed bending load. Its value is usually taken as unity.
K_a = Load correction factor for the reversed axial load. Its value may be taken as 0.8.
K_s = Load correction factor for the reversed torsional or shear load. Its value may be taken as 0.55
K_{sur} = surface finish factor
K_{sz} = Size factor, this is due to the fact that a longer specimen will have more defects than a smaller one.
K_r = reliability factor
K_t = temperature factor
K_i = Impact factor.

- For reversed bending load, endurance limit, $f_e^1 = f_{eb}.K_{sur}.K_{sz}.K_r.K_t.K_i$
- For the reversed axial load, endurance limit $f_e^1 = f_{ea}.K_{sur}.K_{sz}.K_r.K_t.K_i$
- In solving problems, if the value of any of the above factors is not known; it may be taken as unity.
- The factor of safety for fatigue loading should be based on endurance limit.,

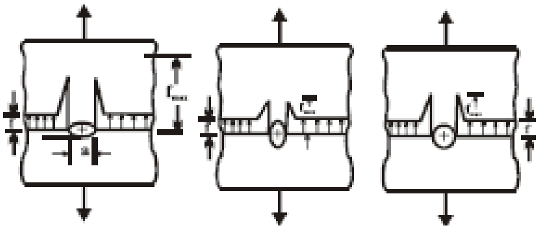
1.5 STRESS CONCENTRATION

When a machine component changes the shape of its shape of its cross – section the simple stress distribution no longer holds good & neighborhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration.

The material near the edges is stressed considerably higher than the average value. The maximum stress occurs at some point on the fillet and is directed parallel to the boundary at that point.



- Theoretical or form stress concentration factor is defined as the ratio of the maximum stress in a member (at a notch or a fillet)m to the nominal stresses at the same section. It is denoted by K_t.
- In static loading, stress concentration in ductile material is not so serious as in brittle materials, because in ductile materials local deformation or yielding takes place which reduces the concentration. In brittle materials, cracks may appear at these local concentration of stress which will increases the stress over the rest of the section. In order to avoid failure due to stress concentration, fillets at the changes of section must be provided.
- In cyclic loading, stress concentration is always serious because the ductility of the material is not effective in relieving the concentration of stress caused by cracks, flaws or any sharp discontinuity in the geometrical form of the member is above the endurance limit of the material, a crack may develop under the section of repeated load and the crack will lead to failure of the member.

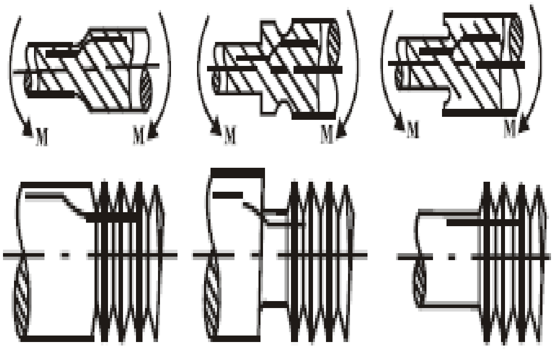


1.6 STRESS CONCENTRATION DUE TO HOLES-

The stress at the joints away from the hole is practically uniform and the maximum stress will be induced at the edge of the hole.

$$f_{max} = f\left(1+\frac{2a}{b}\right)$$

- The presence of stress concentration cannot be totally eliminated but it may be reduced to some extent.



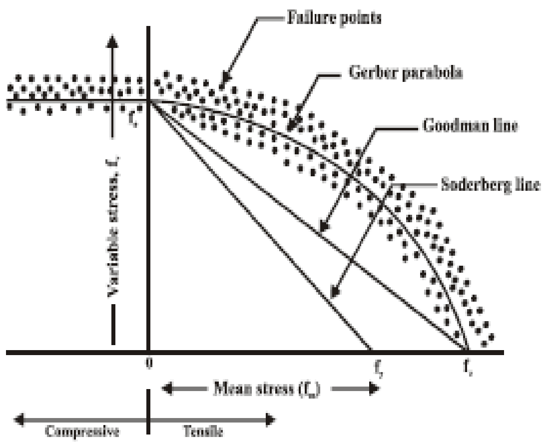
- **Notch Sensitivity:** It may be defined as the degree to which theoretical effect of stress concentration is actually reached. The stress gradient depends mainly on the radius of notch, hole or fillet and on the grain size of material. When notch sensitivity factor concentration factor may be obtained by the relations-
  $k_f = 1 + q\,(k - 1)$ (for tensile stress)
  $k_{fs} = 1 + q\,(k_{ts} - 1)$(for shear stress)
- There are several ways in which problems involving combination of stresses may be solved.

(1) Gerber Method

f_e = fatigue corresponding to the case of complete reversal ($f_m = 0$)
f_u = static ultimate strength corresponding to $f_v = 0$
Generally, the test data for ductile material fall closer to Gerber Parabola, but because of scatter in the test points, a straight line relationship (i.e, Goodman line and soderberg line) is usually. According to Gerber,

$$\frac{1}{F.S.} = \left(\frac{f_m}{f_u}\right)^2 F.S. + \frac{f_v}{f_e}$$

where,
F.S. = Factor of safety
Considering fatigue stress concentration factor (k_p)



(2) Goodman Method for Combination of Stresses :-

A Goodman line is used when the design is based on ultimate strength and may be used for ductile or brittle materials Line AB concocting f_e and f_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 stresses line CD may be drawn parallel to the line AB.

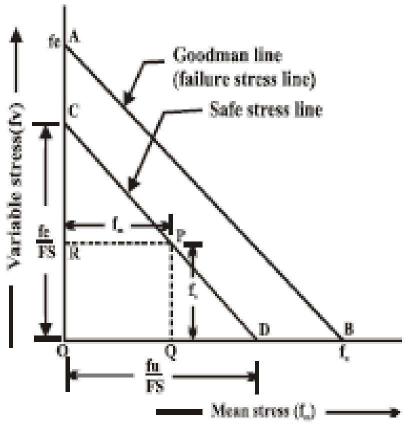
$$\frac{1}{F.S.} = \frac{f_m}{f_u} + \frac{f_v}{fe}$$

Consider the load, surface finish and size factor.

$$\frac{1}{F.S.} = \frac{f_m}{f_u} + \frac{f_v k_f}{fe.K_{sur}.K_{sz}}$$

- Hence we have assumed the same factor of safety (F.S.) for the ultimate tensile strength (f_u) and endurance limit (f_e). In case the factor of safety relating to both these stresses is different then.

$$\frac{f_v}{f_e/(F.S.)_e} = 1 - \frac{f_m}{f_u/(F.S.)_u}$$



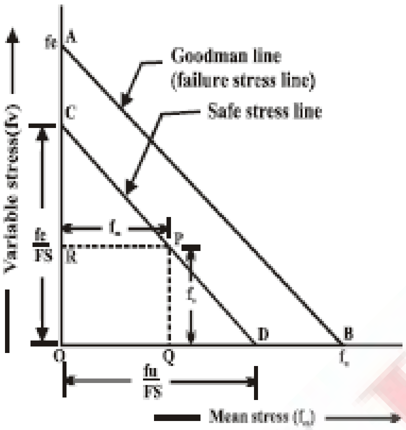
(3) Soderberg Method:-

A straight line connecting the endurance limit (f_e) and the yield strength (f_y) is a soderberg line. This line is used when the design is based on yield strength.
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.

$$\frac{1}{F.S.} = \frac{f_m}{f_y} + \frac{f_v.k_f}{f_e}$$

Considering the load factor, surface finish factor and size the relation is

$$\frac{1}{F.S.} = \frac{f_s}{f_y} + \frac{f_v.k_f}{f_{eb}.k_{sur}.ksz}$$



- The soderberg method is particularly used for ductile materials. For a reversed shear loading

$$\frac{1}{F.S.} = \frac{f_{ms}}{f_{ys}} + \frac{f_{vs}.k_{fs}}{f_{eb}.k_{sur}.k_{sz}}$$

1.7 APPLICATION OF SODERBERG'S EQUATION

(a) Axial loading – mean stress $f_m = \frac{W_v}{A}$

Variable stress $f_v = \frac{W_v}{A}$

Where, W_m = mean or average load
W_v = variable load
A = cross – sectional area

$$\therefore F.S. = \frac{f_y.A}{W_m + \left(\frac{f_y}{f_e}\right)k_f.W_v}$$

(b) Simple Bending-

Bending stress $f_b = \frac{M.y}{I} = \frac{M}{Z}$

Variable bending stress $f_v = \frac{M_v}{Z}$

Design bending stress,

$$f_b = \frac{32}{\pi d^3}\left[M_m + \left(\frac{f_y}{f_e}\right)k_f.M_v\right] \text{and } F.S. = \frac{f_y}{f_b}$$

(c) Simple Torsion of circular shaft-

$$T = \frac{\pi}{16}.f_s.d^3$$

Mean or average shear stress, $f_{ms} = \frac{16T_m}{\pi d^3}$

Variables shear stress , $f_{vs} = \frac{16T_m}{\pi d^3}$

$$f_s = \frac{16}{\pi d^3}\left[T_m + \left(\frac{f_{ys}}{f_{es}}\right)k_{fs}.T_v\right]$$

$$\therefore F.S. = \frac{f_{ys}}{\frac{16}{\pi d^3}\left[T_m + \left(\frac{f_{ys}}{f_{es}}\right)k_{fs}.T_v\right]}$$

(d) Combined bending and torsion of circular shaft maximum shear stress,

$$(f_s)_{max} = \frac{f_{ys}}{F.S.} = \frac{1}{2}\sqrt{f_b^2 + 4_s^f}$$

$$\frac{f_{ys}}{F.S.} = \frac{1}{\pi d^3}\sqrt{\left[\left(\frac{f_y}{f_e}\right)k_f.M\right]^2 + T^2}$$

GATE QUESTIONS

Topics		Page No

1 STATIC, DYNAMIC LOADING & FAILURETEORIES

Q.1 The piston rod of diameter 20 mm and length 700 mm in a hydraulic cylinder is subjected to a compressive force of 10 kN due to the internal pressure. The end conditions for the rod may be assumed as guided at the piston end and hinged at the other end. The Young's modulus is 200 GPa. The factor of safety for the piston rod is

a) 0.68 b) 2.75
c) 5.62 d) 11.0

[GATE-2007]

Q.2 A thin spherical pressure vessel of 200 mm diameter and 1 mm thickness is subjected to an internal pressure varying from 4 to 8 MPa. Assume that the yield, ultimate and endurance strength of material are 600, 800 and 400 MPa respectively. The factor of safety as per Goodman's relation is

a) 2.0 b) 1.6
C) 1.4 d) 1.2

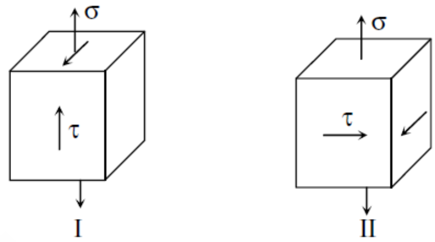
[GATE-2007]

Q.3 A forged steel link with uniform diameter of 30 mm at the centre is subjected to an axial force that varies from 40 kN in compression to 160 kN in tension. The tensile (S_u), yield (S_y) and corrected endurance (S_e) strengths of the steel material are 600 MPa, 420 MPa and 240 MPa respectively. The factor of safety against fatigue endurance as per Soderberg's criterion is

a) 1.26 b) 1.37
c) 1.45 d) 2.00

[GATE-2009]

Q.4 Consider the two states of stress as shown in configurations I and II in the figure below. From the standpoint of distortion energy (von-Mises) criterion, which one of the following statements is true?



a) I yields after II
b) II yields after I
c) Both yield simultaneously
d) Nothing can be said about their relative yielding

[GATE-2014]

Q.5 Which one of following is **NOT** correct?

a) Intermediate principal stress is ignored when applying the maximum principal stress theory
b) The maximum shear stress theory gives the most accurate results amongst all the failure theories
c) As per the maximum strain energy theory, failure occurs when the strain energy per unit volume exceeds a critical value
d) As per the maximum distortion energy theory, failure occurs when the distortion energy per unit volume exceeds a critical value

[GATE-2014]

Q.6 A rotating steel shaft is supported at the ends. It is subjected to a point load at the centre. The maximum bending stress developed is 100 MPa. If the yield, ultimate and corrected endurance strength of the shaft material is

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13	14
(c)	(b)	(a)	(c)	(b)	2	1.8	(b)	173.28	(c)	(c)	*	(b)	(b)

EXPLANATIONS

Q.1 (c)

Given: d = 20 mm, l = 700 mm,

E = 200 GPa = 200×10^9 N/m^2 = 200×10^3 N/mm^2

Compressive or working Load = 10 kN

According to Euler's theory, the crippling or buckling load (W^{cr}) under various end conditions is given by the general equation,

$$Wcr = \frac{c\pi^2 El}{l^2}$$

Give that one end of guided at the piton end and hinged at the other end So.

C = 2

From equation (i)

$$W_{cr} = \frac{2\pi^2 EI}{l^2} = \frac{2\pi^2 E}{l^2}\times\frac{\pi}{64}d^4,$$

$$l = \frac{\pi}{64}d^4$$

$$= \frac{2\times9.81\times200\times10^3}{(700)^2}\times\frac{3.14}{64}\times(20)^4$$

= 62864.08 N = 62.864 kN

We known that, factor of safety (FO)

$$FOS = \frac{\text{Crippling Load}}{\text{Working Load}} = \frac{62.864}{10} = 6.28$$

The most appropriate is (c).

Q.2 (b)

Given : d = 200 mm, t = 1mm, σ_u=800Mpa, σ_e =400 MPa

Circumferential stress induced in spherical pressure vessel is

$$\sigma = \frac{P\times r}{2t} = \frac{p\times100}{2\times1} = 50\ p\ MPa$$

Given that, pressure vessel is subject to an internal pressure varying from 4 to 8 MPa.

So σ_{min} = 50×4 =200Mpa

σ_{min} = 50×8 =400Mpa

Mean stress.

$$\sigma_m = \frac{\sigma_{min}+\sigma_{max}}{2} = \frac{200+400}{2} = 300MPa$$

Variable stress.

$$\sigma_v = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{400-200}{2} = 100MPa$$

From the Goodman method

$$\frac{1}{F.S} = \frac{\sigma_m}{\sigma_u}+\frac{\sigma_v}{\sigma_e}$$

$$= \frac{300}{800}+\frac{100}{400} = \frac{3}{8}+\frac{1}{4} = \frac{5}{8} \Rightarrow F.S - \frac{8}{5} = 1.6$$

Q.3 (a)

Given

Su or su = 600MPa, Sy

or sy = 420MPa, Se or se = 240MPa,

D= 30 mm

F_{mm} = 160 kN (Tension).

F_{min} = -40 kN (Compression)

Maximum stress

$$\sigma_{max} = \frac{F_{max}}{A} = \frac{160\times10^3}{\frac{\pi}{4}(30)^2} = 226.47MPa$$

Minimum stress

$$\sigma_{min} = \frac{F_{min}}{A} = \frac{40\times10^3}{\frac{\pi}{4}(30)^2} = -56.62MPa$$

Mean stress.

$$\sigma n = \frac{\sigma_{max}+\sigma_{min}}{2} = \frac{226.47 - 56.62}{2}$$

= 84.925 MPa

ASSIGNMENT QUESTIONS

Q.1 If the size of a standard specification for fatigue testing machine is increased the endurance limit for the material will

a) have same value as that of standard specimen
b) increases
c) decreases
d) none of these

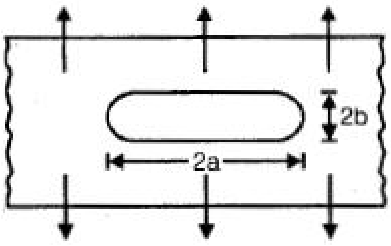
Q.2 Stress concentration factor is ratio of

a) endurance limit without stress concentration to endurance limit with stress concentration
b) endurance limit with stress concentration to endurance limit without stress concentration
c) endurance limit with stress concentration to factor of safety
d) none of these

Q.3 Resistance to fatigue of a material is measured by

a) Young's modulus
b) Endurance limit
c) Ultimate tensile strength
d) Coefficient of elasticity

Q.4 In a semi-infinite flat plate shown in the figure, the theoretical stress concentration factor K_t for on elliptical hole of major axis 2a and minor axis 2b is given by

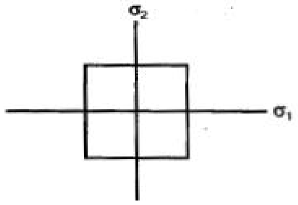


a) $K_t = \frac{a}{b}$ b) $K_t = 1 + \frac{a}{b}$

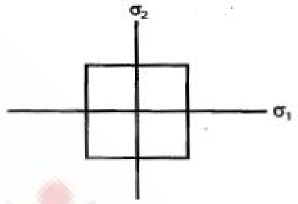
c) $K_t = 1 + \frac{2b}{a}$ d) $K_t = 1 + \frac{2a}{b}$

Q.5 Which one of the following graph represents von-miss yield criterion

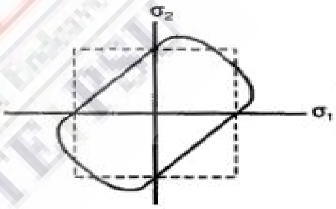
a)



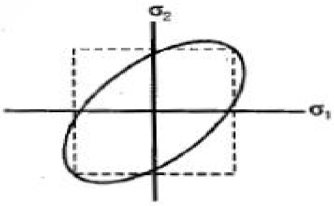
b)



c)



d)



Q.6 The designation M 33 x 2 of a bolt means

a) Metric threads M 33, of 2 in numbers
b) Metric threads with cross section of $33 mm^2$
c) Metric threads of 33 mm outside diameter and 2 mm pitch
d) Bolt of 33 mm nominal diameter having 2 thread per Cm

Q.7 For bolts of uniform strength, the shank diameter is made equal to

a) major diameter of threads
b) pitch diameter of threads
c) minor diameter of threads
d) nominal diameter of threads

Q.8 According to IBR, the thickness of boiler shell should not be less than
a) 4 mm b) 5 mm
c) 6 mm d) 7 mm

Q.9 According to IBR, the factor of safety of riveted joint should not be less than
a) 1 b) 2
c) 3 d) 4

Q.10 The rivet head used for boiler plate riveting is usually
a) Snap head
b) Pan head
c) Counter sunk head
d) conical head

Q.11 Which of the following is a flexible coupling?
a) Muff coupling
b) Marine coupling
c) Protected type flange coupling
d) Oldham coupling

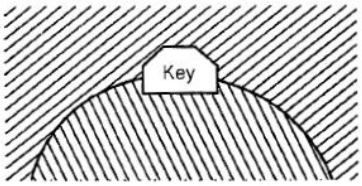
Q.12 In a fillet welded joint, the weakest area of weld is
a) Toe b) Root
c) Throat d) Face

Q.13 For overhauling condition
a) friction angle < helix angle
b) friction angle> helix angle
c) friction angle = helix angle
d) none of these

Q.14 The maximum efficiency of self locking screw is
a) 50% b) 70%
c) 75% d) 80%

Q.15 In the assembly design of shaft, pulley and key the weakest member is
a) Pulley b) Key
c) Shaft d) None

Q.16 The key shown in the above figure is a



a) Barth key b) Kennedy key
c) Lewis key d) Wood ruff key

Q.17 Which key transmits power through frictional resistance only
a) Woodruff b) Kennedy
c) Sunk d) Saddle

Q.18 In a multiple disc clutch n_1 and n_2 are the number of discs on the driving and driven shafts respectively. The number of pair of contact surfaces will be
a) $n_1 + n_2$ b) $n_1 + n_2 - 1$
c) $n_1 + n_1 + 1$ d) $n_1+n_2/2$

Q.19 The frictional torque transmitted in a flat pivot bearing assuming uniform wear
a) μWR b) $\frac{3}{4}\mu WR$
c) $\frac{2}{3}\mu WR$ d) $\frac{1}{2}\mu WR$
Where μ = Coefficient of friction
W = Load over the bearing
R = Radius of bearing

Q.20 Tapered roller bearing can take
a) radial load only
b) axial load only
c) both radial & axial load & the ratio of these being less than unity
d) both radial and axial load and the ratio of these being greater than unity

Q.21 Which of the following are antifriction bearing
a) sleeve bearings
b) hydrostatic bearings
c) ball and roller bearing
d) none of these

Q.22 In a journal bearing, the radius of the friction circle increases with Increase in
a) load
b) radius of the journal
c) speed of the journal
d) viscosity of the journal

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13	14
(c)	(a)	(b)	(d)	(c)	(c)	(c)	(d)	(d)	(a)	(d)	(c)	(a)	(a)
15	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**
(a)	(d)	(d)	(b)	(d)	(d)	(c)	(b)	(c)	(d)	(b)	(a)	(b)	(b)
29	**30**	**31**	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**	**41**	**42**
(b)	(a)	(c)	(c)	(c)	(d)	(b)	(d)	(c)	(b)	(b)	(b)	(d)	(b)
43	**44**	**45**	**46**	**47**	**48**	**49**	**50**	**51**	**52**	**53**	**54**	**55**	**56**
(d)	(a)	(c)	(c)	(a)	(a)	(c)	(c)	(c)	(c)	(b)	(a)	(b)	(d)
57	**58**	**59**	**60**	**61**	**62**	**63**	**64**						
(a)	(b)	(a)	(d)	(b)	(d)	(c)	(b)						

EXPLANATIONS

Q.1 (c)
In the diameter or size of the components is more the surface area will have greater number of surface defect. Hence endurance strength of component reduced with increase in size.

Q.2 (a)
Fatigue stress concentration factor can be defined as the ratio of endurance limit without stress concentration to the endurance limit with stress concentration.

Q.3 (b)
Endurance limit is term defining the fatigue resistance of material.

Q.4 (d)
$$K_t = 1 + \frac{4a}{2b} = 1 + \frac{2a}{b}$$

Q.5 (c)

Q.6 (c)

Q.7 (c)
Bolt of uniform strength are made by
1 . Reducing the diameter of shank of bolt corresponding to that of minor Diameter
2. Making a hole and making the area of shank equal to root area

Q.8 (d)

Q.9 (d)

Q.10 (c)

Q.11 (b)
Oldham coupling is the flexible coupling used to connect two shaft have lateral misalignment.

Q.12 (c)
Because throat area is the minimum area and $A_{throat} = h \times L$

Q.13 (a)
If $\phi > a$
i.e., friction angle is less then Helix angle, then torque require to lower the load will be negative i.e. , no force is required to lower the load

the load itself will begin to turn the screw and descend down, unless a restraining torque is applied this condition is known as overhauling.

Q.14 **(b)**.
For self-locking, the maximum efficiency of screw should be 50%.
If efficiency is more than 50% the condition is known as overhauling.

Q.15 **(a)**
Key is the weakest member in the assembly of shaft, pulley and key. Key acts as a safely device, whenever there is excess load appears on the pulley key fails first & it keeps safer to shaft and pulley.

Q.16 **(d)**

Q.17 **(d)**
Saddle key is a key that fits in the key way of the hub only. In this case there is no keyway provided on the shaft and friction between shaft, key and hub prevents relative motion between the shaft and the hub and power is transmitted by means of friction only.

Q.18 **(b)**
Number of pair in contact= n_1+n_2-1

Q.19 **(d)**
Frictional torque transmitted

$$T_f = \frac{1}{2}\mu WR \text{ (Uniform wear)}$$

$$T_f = \frac{2}{3}\mu WR \text{ (Uniform wear)}$$

Q.20 **(d)**

Q.21 **(c)**
Since coefficient of friction (.t) for Ball and Roller bearings are very low as compared to sliding contact bearing hence they are known as antifriction bearing.

Q.22 **(b)**
Radius of friction circle = $\mu \times r$
Where μ = Coefficient of friction
r = Radius of journal

Q.23 **(c)**

Q.24 **(c)**
Bearing characteristic number = $\frac{ZN}{P}$
Hence, it is unction of viscosity, speed and bearing pressure.

Q.25 **(b)**
The load carrying capacity of sliding contact bearing (Hydrodynamic bearing) is linearly proportional to speed while rolling contact bearing have finite life for a given combination of load and speed.
Sliding contact bearing is suitable for high load high speed condition particularly from the consideration of a long life.

Q.26 **(a)**

Q.27 **(b)**

Q.28 **(b)**
Steel-screw, Phosphor Bronze-nut, Cast iron also used for nut in case of low speed.

Q.29 **(b)**

Q.30 **(a)**
Axle is used to support the wheel hence it is subjected to only bending moment. Since it has no role to play in transmission of power hence it is not subjected to twisting moment.

Q.31 **(c)**

Q.32 **(c)**

Q.33 **(c)**