UNIT-1 - STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS

PART –A

1)**How the machine design may be classified? (Nov 2012)**

The machine design maybe classified as follows.

a) Adaptive design b)Developed design c)New design d)Rational design e)Empirical design f)Industrial design

2) **What are the various phases of design process? (May 2015)**

i. Recognition of need. ii. Definition of problem iii. Synthesis
iv. Analysis and optimization v. Evaluation vi. Presentation

3) **Define creep**.

When a part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called creep.

4) **Differentiate between resilience and toughness. (May 2011)**

Resilience is the property of the material to absorb energy and to resist shock and impact loads. This property is essential for spring materials.

5) **What are the factors affecting selection of material for machine element? (Nov 2010)**

The factors affecting selection of material for machine element are,

a) Load applied b)Purpose and operating conditions of the part c)Suitability for manufacture d)Minimum weight and optimal size e)Availability and cost.

6) **What are the different types of loads that can act on machine components?**

The types of loads act on machine components are,

a) Steady load b)Variable load c)Shock load d)Impact load.

7) **Define bearing stress.**

A localized compressive stress at the surface of contact between two members of machine part that are relatively at rest is known as bearing stress.

8) **What is principle stress and principle plane?**

A plane which has no shear stress is called principle plane the corresponding stress is called principle stress.

9)**Differentiate between direct stress and bending stress.**

Direct stress : Load is applied axially. The stress distribution is uniform throughout the cross Section.
Bending stress: Load is applied laterally i.e., Perpendicular to the axis.

10)**What is torsion shear stress?**

When a machine member is subjected to the action of two equal and opposite couples acting in parallel planes (or torque or twisting moment) then the machine member is said to be subjected to torsion. The stress set-up by torsion is known as torsion is known as torsion shear stress.

11)**Define: Factor of safety (Nov 2012, April 2011)**

Factor of safety (FOS) is defined as the ratio between the maximum stress and working stress.
FOS for brittle material is 4.

12)**List out the factors involved in arriving at factor of safety. (Nov 2011)**

i. Material properties ii. Nature of loads iii. Presence of localized stresses
iv. Mode of failures v. Presence of initial stresses

13)**What is curved beam? Give some example for curved beam. (Nov 2012)**

If the neutral axis of the cross section is shifted towards the centre of curvature of the beam causing a non – linear distribution of stress, then the given beam is curved beam.

14) **What is eccentric load and eccentricity?**

An external load ,whose line of action is parallel but does not coincide with the centrodial axis of the machine component ,is known as an eccentric load .the distance between the centrodial axis of the component and the eccentric load is called eccentricity.

15**) If the section is unsymmetrical, where the maximum and minimum bending stress will occur for curved beam?**

If the section is unsymmetrical, then the maximum bending stress may occur at either the inside fibre or the outside fibre.

16)**Give some methods of reducing stress concentration. (Nov 2012)**

i. Avoiding sharp corners.
ii. Providing fillets.
iii. Use of multiple holes instead of single hole
iv. Undercutting the shoulder parts.

17)**What is an S-N Curve?(Dec 2015)**

An S- N curve has fatigue stress on Y axis and number of loading cycles in *X* axis. It is used to find the fatigue stress value corresponding to a given number of cycles.

18)**Distinguish between brittle fracture and ductile fracture.**

In brittle fracture, crack growth is up to a small depth of the material. In ductile fracture large amount of plastic deformation is present to a higher depth

19)**Define stress concentration and stress concentration factor.(May 2014)**

Whenever a machine component changes the shape of its cross-section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called ***stress concentration***. It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness or scratches etc.

The theoretical or form stress concentration factor is defined as the ratio of the maximum stress in a member (at a notch or a fillet) to the nominal stress at the same section based upon net area.

Mathematically, theoretical or form stress concentration factor, Kt =Maximum stress /Nominal stress

The value of Kt depends upon the material and geometry of the part.

20)**What are various theories of failure? (Nov 2012)**

i. Maximum principal stress theory. ii. Maximum shear stress theory.

iii. Maximum principal strain theory

21)**Define endurance limit. (May 2015)**

Endurance limit is the maximum value of completely reversed stress that the standard specimen can sustain an infinite number (10^6) of cycles without failure

22)**What are the factors affecting endurance strength.**

Factors affecting endurance strength are

i. Load ii. Surface finish iii. Size iv. Temperature v. Impact vi. Reliability

23)**What are the types of variable stresses?**

a. Completely reversed or cyclic stresses b. Fluctuating stresses c. Repeated stresses

24)**Differentiate between repeated stress and reversed stress.**

Repeated stress refers to a stress varying from zero to a maximum value of same nature.

Reversed stress of cyclic stress varies from one value of tension to the same value of compression.

25)**Explain notch sensitivity. State the relation between stress concentration factor, fatigue stress concentration factor and notch sensitivity**.

Notch sensitivity (*q*) is the degree to which the theoretical effect of stress concentration is actually reached. The relation is, $K_f = 1 + q\ (K_t-1)$

26) Describe the material properties of hardness, stiffness and resilience. (Nov 2013, May 2014)

Hardness is the ability of material to resist scratching and indentation.

Stiffness is the ability of material to resist deformation under loading.

Resilience is the ability of material to resist absorb energy and to resist shock and impact load.

27) What are the methods used to improve fatigue strength? (Nov 2013, Nov 2014)

Cold working like shot peening, burnishing, **Heat Treatment** like induction hardening, case hardening, nitrating and **Pre Stressing Or Auto Fretting**.

28)What is "Adaptive Design"? where is it used? Give Examples.(Nov 2012)

This is used where a new product is developed just by making small changes to the existing product. This is best suited to occasions where no or limited scope is available to go for an entirely new design.

29) What do you mean by Optimum design?(Nov 2011)

Optimization is a process of maximizing a decided quantity or minimizing undecided one.

30). Define Poisson's Ratio? (April 2011)

It is a ratio between lateral strains to the longitudinal strain.

31). What are the common materials used in engineering design?(Nov 2015)

(i) alloy steel, (ii) low , medium and high carbon steel (iii) cast iron (iv) Composites -fiber and metal matrix composites

PART- B

1) a) Give the general procedure in machine design.

b) What are the general considerations in machine design?

c)Give some important mechanical properties of the metals.

a) Give the general procedure in machine design.

1. Need or Aim
2. Synthesis (Mechanisms)
3. Analysis of Forces
4. Material Selection
5. Design of Elements
6. Modification
7. Detailed Drawing
8. Production (Explain in detail)

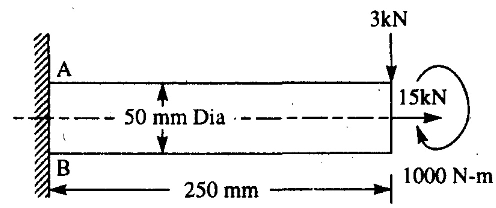
b) What are the general considerations in machine design?

1. Strength & Stiffness
2. Surface finish and tolerances
3. Manufacturability
4. Ergonomics and aesthetics
5. Working Atmosphere
6. Wear & Hardness Requirement
7. Cooling and Lubrication
8. Safety & Reliability
9. Noise Requirement
10. Cost. (Explain in detail)

c)Give some important mechanical properties of the metals.

The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load. These mechanical properties of the metal include **strength, stiffness, elasticity, plasticity, ductility, brittleness, malleability, toughness, resilience, creep and hardness**.
(Explain in detail)

2) A shaft, as shown in Fig, is subjected to a bending load of 3 kN, pure torque of 1000 N-m and an axial pulling force of 15 kN. Calculate the stresses at A and B. All dimensions in mm.



Solution. Given : W = 3 kN = 3000 N ; T = 1000 N-m = 1×10^6 N-mm ; P = 15 kN= 15×10^3 N ;
d = 50 mm; x = 250 mm

STEPS: 1. Calculate the cross-sectional area of the shaft,
2. Calculate Tensile stress due to axial pulling at points A and B,
3. Calculate Bending stress at points A and B,
4. This bending stress is tensile at point A and compressive at point B.

∴ Resultant tensile stress at point A,

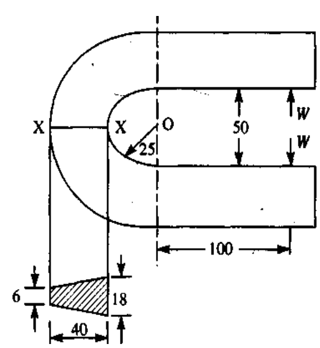
$$\sigma_A = \sigma_b + \sigma_o = 61.1 + 7.64 = 68.74 \text{ MPa}$$

and Resultant compressive stress at point B,

$$\sigma_B = \sigma_b - \sigma_o = 61.1 - 7.64 = 53.46 \text{ MPa}$$

5. Calculate the shear stress at points A and B due to the torque transmitted
6. Calculate the maximum and minimum principal (or normal) stress at point A, and maximum shear stress
7. Calculate the maximum and minimum principal (or normal) stress at point A, and maximum shear stress

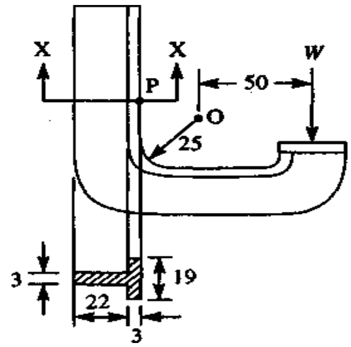
3) The frame of a punch press is shown in Fig. Find the stresses at the inner and outer surface at section X-X of the frame, if W= 5000N. Section at X-X; All dimensions in mm.



Given : W = 5000 N ; b_i = 18 mm ; b_o = 6 mm ; h = 40 mm ; R_i = 25 mm ;
R_o = 25 + 40 = 65 mm. Calculate the following,

1. area of section at X-X,
2. radius of curvature of the neutral

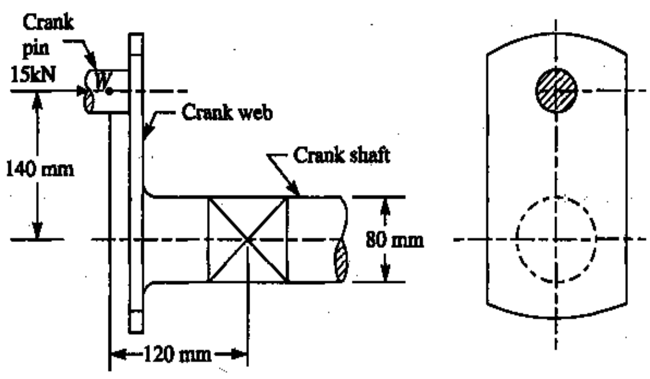
axis, (use PSG DB)
3. radius of curvature of the centroidal axis,
4. Distance between the centroidal axis and neutral axis,
5. distance between the load and centroidal axis,
6. Bending moment about the centroidal axis
7. direct tensile stress at section X-X,
8. Distance from the neutral axis to the inner and outer surface,
9. Maximum bending stress and resultant stress at the inner surface and outer surface.



4) A C-clamp is subjected to a maximum load of W, as shown in Fig. If the maximum tensile stress in the clamp is limited to 140 MPa, find the value of load W. Section of X-X ; All dimensions in mm. (Nov 2012)

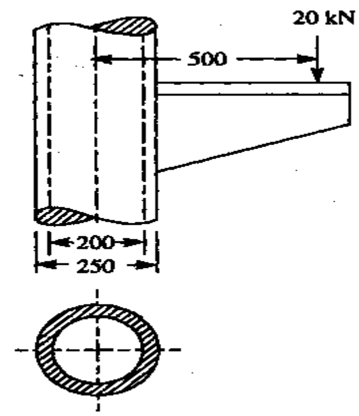
Solution. Given : $\sigma_{t(max)}$ = 140 MPa = 140 N/mm^2 ; R_i = 25 mm ; R_o = 25 + 25 = 50 mm ;b_i = 19 mm ; t_i = 3 mm ; t = 3 mm ; h = 25 mm
calculate the following,
1. area of section at *X-X*,
2. radius of curvature of the neutral axis, (use PSG DB)
3. radius of curvature of the centroidal axis,
4. Distance between the centroidal axis and neutral axis,
5. distance between the load and centroidal axis,
6. Bending moment about the centroidal axis
7. direct tensile stress at section X-X,
8. Distance from the neutral axis to the inner and outer surface,
9. Maximum bending stress and resultant stress at the inner surface and outer surface.

5) An overhang crank with pin and shaft is shown in Fig. A tangential load of 15 kN acts on the crank pin. Determine the maximum principal stress and the maximum shear stress at the centre of the crankshaft bearing.



Solution. Given : W = 15 kN = 15 × 10^3 N ; d = 80 mm ;
y = 140 mm ; x = 120 mm
1. Calculate the bending moment at the centre of the crankshaft bearing,
2. Calculate torque transmitted at the axis of the shaft,
3. Calculate bending stress due to the bending moment
4. Calculate shear stress due to the torque transmitted,
5. Calculate Maximum principal stress
6. Calculate Maximum shear stress

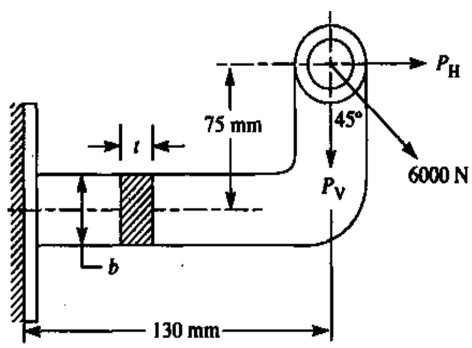
6) A hollow circular column of external diameter 250 mm and internal diameter 200 mm carries a projecting bracket on which a load of 20 kN rests, as shown in Fig. The centre of the load from the centre of the column is 500 mm. Find the stresses at the sides of the column. All dimensions in mm.



Solution. Given : D = 250 mm ; d = 200 mm ;P = 20 kN = 20 × 10^3 N ; e = 500 mm, calculate

1. cross-sectional area of column,
2. Direct compressive stress,
3. Section modulus for the column,
4. Bending moment and bending stress
5. Maximum Compressive stress and tensile stress.

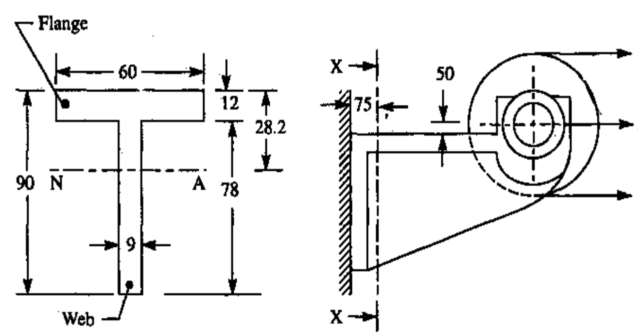
7) A mild steel bracket as shown in Fig. is subjected to a pull of 6000N acting at 45° to its horizontal axis. The bracket has a rectangular section whose depth is twice the thickness. Find the cross-sectional dimensions of the bracket, if the permissible stress in the material of the bracket is limited to 60 MPa. (Nov 2012)



Solution. Given : P = 6000 N ; θ = 45° ; σ = 60 MPa = 60 N/mm^2
find,

1.Cross-sectional area and Section Modulus
2.load, bending moment and maximum bending stress of horizontal component.(P_H)
3. load, bending moment , Direct stress and maximum bending stress of vertical component.(P_V)
4.dimension of the bracket by equating permissible stress to resultant stress.
5. the final answers is t=28.4 mm, b=56.8mm.

8) A horizontal pull P = 5 kN is exerted by the belting on one of the cast iron wall brackets which carry a factory shafting. At a point 75 mm from the wall, the bracket has a T-section as shown in Fig. Calculate the maximum stresses in the flange and web of the bracket due to the pull.



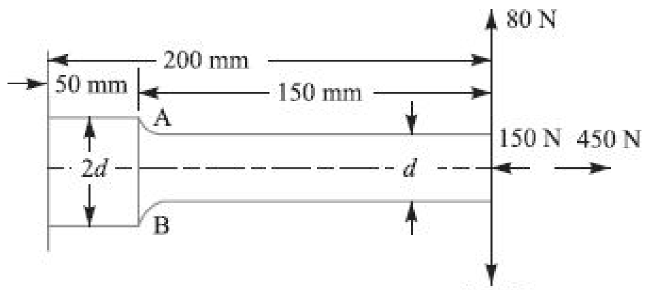
All dimensions in mm.

Given : Horizontal pull, P = 5 kN = 5000 N
find,

1. cross-sectional area of the section, and direct tensile stress.
2. find the position of neutral axis in order to determine the bending stresses. The neutral axis passes through the centre of gravity of the section.
$\bar{Y}$ = Distance of centre of gravity (i.e. neutral axis) from top of the flange.
3. moment of inertia of given section about N.A.
4. distance of N.A. from the top of flange , bottom of the web and point of application of load.
5. bending moment exerted in section
6. tensile stress on flange and compressive stress on web
7. maximum tensile stress and compressive stress on flange .

9) A steel cantilever is 200 mm long. It is subjected to an axial load which varies from 150 N (compression) to 450 N (tension) and also a transverse load at its free end which varies from 80 N up to 120 N down. The cantilever is of circular cross-section. It is of diameter 2d for the first 50mm and of diameter 'd' for the remaining length. Determine its diameter taking a factor of safety of 2. Assume the following values:

Yield stress = 330Mpa
Endurance limit in reversed loading = 300Mpa
Correction factors =0.7 in reversed axial Loading = 1.0 in reversed Bending
Stress concentration factor = 1.44 for bending
= 1.64 for axial loading
Size effect factor = 0.85
Surface effect factor = 0.90
Notch sensitivity index = 0.90 (Nov 2015)



Solution. Given : l = 200 mm; $Wa(max)$ = 450 N; $Wa(min)$ = – 150 N ; $Wt(max)$ = 120 N ; $Wt(min)$ = – 80 N; $F.S.$ =2 ; σy = 330 MPa = 330 N/mm^2 ; σe = 300 MPa = 300 N/mm^2 ; Ka = 0.7; Kb = 1; Ktb = 1.44 ; Kta = 1.64; Ksz = 0.85 ; $Ksur$ = 0.90 ; q = 0.90

find

1. Mean axial load and stress.
2. Variable Axial load and stress.
3. fatigue stress concentration factor for reversed axial loading,

$$K_{fa} = 1 + q\,(K_{ta} - 1) = 1 + 0.9\,(1.64 - 1) = 1.576$$

4. endurance limit stress for reversed axial loading,

$$\sigma_{ea} = \sigma_e \times Ka = 300 \times 0.7 = 210 \text{ N/mm}^2$$

5. equivalent normal stress at point A due to axial loading,
6. Mean Bending load , Bending moment and bending stress
7. Variable Bending load , Bending moment and bending stress
8. fatigue stress concentration factor for reversed bending

$$K_{fb} = 1 + q\,(K_{tb} - 1)$$

9. endurance limit stress for reversed bending, $\sigma_{eb} = \sigma_e \times K_b$
10. equivalent normal stress at point A due to bending
11. total equivalent normal stress at point A,
12. find the diameter by equating permissible stress to total equivalent normal stress(d = 12.9mm)

10) A pulley is keyed to a shaft midway between two bearings. The shaft is made of cold drawn steel for which the ultimate strength is 550 MPa and the yield strength is 400 MPa. The bending moment at the pulley varies from – 150 N-m to + 400 N-m as the torque on the shaft varies from – 50 N-m to +150 N-m. Obtain the diameter of the shaft for an indefinite life. The stress concentration factors for the keyway at the pulley in bending and in torsion are 1.6 and 1.3 respectively. Take the following values:

Factor of safety = 1.5
Load correction factors = 1.0 in bending,
and 0.6 in torsion
Size effect factor = 0.85
Surface effect factor = 0.88 (Nov 2012, Nov 2015)

Solution. Given : σ_u = 550 MPa = 550 N/mm^2 ; σ_y = 400 MPa = 400 N/mm^2 ;Mmin = – 150 N-m; Mmax = 400 N-m ; Tmin = – 50 N-m ; Tmax = 150 N-m ; Kfb = 1.6 ; Kfs = 1.3 ; F.S. = 1.5 ; Kb = 1 ; Ks = 0.6 ; Ksz = 0.85 ; Ksur = 0.88

Let d = Diameter of the shaft in mm.

find

1. mean or average bending moment and stress
2. variable bending moment and stress
3. Assuming the endurance limit in reversed bending as one-half the ultimate strength and since the load correction factor for reversed bending is 1 (i.e. Kb = 1), therefore endurance limit in reversed bending,
4. Since there is no reversed axial loading, therefore equivalent normal stress due to bending,
5. mean and variable torque
6. mean and variable shear stress
7. Endurance limit stress for reversed torsional or shear loading assuming yield strength in shear
8.diameter by equating equivalent shear stress and permissible shear stress

11) Determine the diameter of a circular rod made of ductile material with a fatigue strength (complete stress reversal), σ_e = 265 MPa and a tensile yield strength of 350 MPa. The member is subjected to a varying axial load from Wmin = -300 x 10^3 N to Wmax = 700 x 10^3 N and has a stress concentration factor = 1.8. Use factor of safety as 2.0.

Solution. Given : σ_e = 265 MPa = 265 N/mm^2 ; σy = 350 MPa = 350 N/mm^2 ; Wmin = – 300 × 10^3 N ; Wmax = 700 × 10^3 N ; Kf = 1.8 ; F.S. = 2
calculate
1. mean load and mean stress
2. variable load and stress
3. find diameter using Soderberg's formula

12) A circular bar of 500 mm length is supported freely at its two ends. It is acted upon by a central concentrated cyclic load having a minimum value of 20 kN and a maximum value of 50 kN. Determine the diameter of bar by taking a factor of safety 1.5, size effect of 0.85, surface finish factor of 0.9. The material properties of bar at given by: ultimate strength of 650 MPa, yield strength of 500 MPa and endurance strength of 350 MPa. (Nov 2012)

Given : l = 500 mm ; W_{min} = 20 kN = 20 × 10^3 N ; W_{max} = 50 kN = 50 × 10^3 N ;F.S. = 1.5 ; K_{sz} = 0.85 ; K_{sur} = 0.9 ; σu = 650 MPa = 650 N/mm^2 ; σ_y = 500 MPa = 500 N/mm^2 ;σ_e = 350 MPa = 350 N/mm^2
Calculate
1. maximum and minimum bending moment
2. mean bending moment and stress
3. variable bending moment and stress
4. diameter of bar using Goodman's formula (d=59.3mm) and Soderberg's formula(d=62.1mm)

UNIT -II - DESIGN OF SHAFTS AND COUPLINGS
PART –A

1)**Define shaft.**

A shaft is a rotating machine element which is used to transmit power from one place to another. It is used for the transmission of torque and bending moment.

2)**Differentiate between shaft and axle (Nov 2012)**

An axle, though similar in shape to the shaft, is a stationary machine element and is used for transmission of bending moment only. It simply acts as a support for some rotating body.

3)**What is spindle?**

A spindle is a short shaft that imparts motion either to a cutting tool or to a work piece.

4)**What are the types of shaft?**

The various types of stresses induced in the shafts are,
a) Shear stresses due to transmission of torque
b) Bending stresses (c) Stresses due to combined Torsional and bending loads.

5)**What are the standard sizes of transmission shafts?(May 2015)**

The standard sizes of transmission shafts are,
a) 25 mm to 60 mm with 5 mm steps b)60 mm to 110 mm with 10 mm steps
b) 110 mm to 140mm with 15 mm steps d)140 mm to 500 mm with 20 mm steps

Standard length – 5 m, 6 m and 7m.

6)**Define the term critical speed (May 2015)**

The speed, at which the shaft runs so that the additional deflection of the shaft from the axis of rotation becomes infinite, is known as critical or whirling speed.

7)**On what basis the shafts are designed?**

a) Based on rigidity and stiffness. B)Based on strength. C)Based on critical speed.

8)**What are the ways of improving lateral rigidity of shafts? (Nov 2015)**

The ways of improving lateral rigidity of shafts are,

a) Maintaining proper bearing clearances b) Correct gear teeth alignment.

9)**Why rotating shaft are generally made with circular cross section?**

Stress distribution pattern will be uniform throughout the circular cross section.

10)**State any two reasons for preferring hollow shaft over solid shaft. (Nov 2012)**

The two reasons for preferring hollow shaft over solid shaft are,

a) For same weight of shaft, hollow shaft can transmit 1.5 times the torque transmitted by solid shaft.
b) For a particular power transmission, hollow shaft requires minimum weight.

11)**What is key? What is a keyway? (Nov 2012)**

Key is an element which is used to connect two machine parts for preventing relative motion of rotation with respect to each other. Key way is a slot or recess in a shaft and hub of the pulley to accommodate a key.

12)**What are the types of keys? (Nov 2012)**

i. Saddle key ii. Tangent key iii. Sunk key iv. Round key and taper pins

13)**What is woodruff key? State its main application. (Nov 2013)**

Wood ruff key is piece from a cylindrical disc having segmental cross section. A woodruff key is capable of tilting in a recess milled out in the shaft by a cutter having the same curvature as the disc from which the key is made. They are largely used in machine tools and automobile constructions.

14)**What are the various forces acting on a sunk key?(Nov2015)**

The various forces acting on a sunk key are, a)Forces due to fit of the key in its keyway b)Forces due to torque transmitted by the shafts.

15)**What are splines?**

The keys are made integral with the shaft which fits in the keyways broached in the hub. Such shafts are known as splined shafts. These shafts usually have four, six, ten or sixteen splines. The splined shafts are relatively stronger than shafts having a single key way.

16)**What is the effect of keyways cut into the shaft?(Nov 2015)**

The key way cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the keyway and reduction in the cress sectional area of the shaft. In other words the Torsional strength of the shafts is reduced.

17)**What is the function of a coupling between two shafts?**

Couplings are used to connect sections of long transmission shafts and to connect the shaft of a driving machine to the shaft of a driven machine

18) Differentiate between rigid coupling and flexible coupling.

Rigid coupling	Flexible coupling
It is used to connect two shafts having both lateral aligned.	It is used to connect two Shafts which are perfectly and angular misalignment.

19)What are the purposes in machinery for which couplings are used?

1. To provide the connection of shafts of units those are manufactured separately such as motor and generator and to provide for disconnection for repairs or alterations.
2. To provide misalignment of the shafts or to introduce mechanical flexibility.
3. To reduce the transmission of shock from one shaft to another.
4. To introduce protection against over load

20)What is a flange coupling? (May 2015)

Flange coupling is a coupling having two separate cast iron flanges. Each flange is mounted on the shaft end and keyed to it. The faces are turned u at right angle to the axis of the shaft. One of the flange has a projected portion and the other flange has a corresponding recess. This helps to bring the shafts into line and maintain alignment.

21) What are the main functions of the knuckle joints?

a)It is used to transmit axial load from one machine element to other.
b) Small angular movement is possible between the rods.

22)What are the types of flexible coupling and rigid couplings? (Nov 2012)

Flexible coupling is a type of coupling used to connect two shafts having both lateral and angular misalignment. The types are,

a) Bushed pin type coupling b)Universal coupling c)Oldham's coupling

Rigid coupling is used to connect two shafts which are perfectly aligned. The types are

a) Sleeve or muff coupling b)Clamp or split muff or compression coupling c)Flange coupling.

23)What is difference between coupling and a clutch?

A coupling is a device used to make permanent or semi permanent connection whereas a clutch permits rapid connection or disconnection at the will of the operator.

24)List out the requirements of a shaft coupling?

The requirements of a shaft coupling are,

(a)It should be easy to connect or disconnect. b).It should transmit the full power of the shaft.

(c). It should hold the shafts in perfect alignment. d).It should have no projecting parts.

25)When a solid flange coupling is preferred?

For very large shafts or when large Torsional moments and forces are to be transmitted such as those used for propeller shafts, solid flange couplings are preferred.

26) Discuss the various forces acting on the keys.(Nov 2014)

(i) Forces due to torque transmitted giving rise to the shear and compressive strength

(ii)Forces due to tight fit of the key cause rise in compressive stress at shaft and hub

27) What are the possible modes of failure at pin in flange coupling? (Nov 2015)

(i) shear failure (ii) crushing failure

PART- B

1) A shaft is supported by two bearings placed 1 m apart. A 600 mm diameter pulley is mounted at a distance of 300 mm to the right of left hand bearing and this drives a pulley directly below it with the help of belt having maximum tension of 2.25 kN. Another pulley 400 mm diameter is placed 200 mm to the left of right hand bearing and is driven with the help of electric motor and belt, which is placed horizontally to the right. The angle of contact for both the pulleys is 180° and μ = 0.24. Determine the suitable diameter for a solid shaft, allowing working stress of 63 MPa in tension and 42 MPa in shear for the material of shaft. Assume that the torque on one pulley is equal to that on the other pulley. (Nov 2012)

Solution. Given : AB = 1 m ; D_C = 600 mm or R_C = 300 mm = 0.3 m ; AC = 300 mm = 0.3 m ;T_1 = 2.25 kN = 2250 N ; D_D = 400 mm or R_D = 200 mm = 0.2 m ; BD = 200 mm = 0.2 m ; ϴ= 180° = π rad ; μ= 0.24 ;

σ_b = 63 MPa = 63 N/mm^2 ; τ= 42 MPa = 42 N/mm^2

solution

1. Draw space diagram
2. Calculate tension by using formula $t1/t2 = e^{\mu\alpha}$
3. Find vertical load acting at C & D in the shaft
4. Find T_3 and T_4 using formulas
   Torque,$T=(T_3-T_4)R_D$ and $T_3/T_4=T_1/T_2$
5. Find horizontal load acting on C and D
6. Find maximum bending moment for vertical loading and horizontal loading by taking moments at A and B.
7. Find resultant B.M at C(M_C) and D(M_D).

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2}$$

$$M_D = \sqrt{(M_{DV})^2 + (M_{DH})^2}$$

8. Select max value and let it be M

9. Find equivalent twisting moment using formula $T_e=\sqrt{(M^2+T^2)}$ and find diameter of shaft using formula

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

10. Find equivalent bending moment using formula $M_e=1/2\,(M+\sqrt{(M^2+T^2)})$ and find diameter of shaft using formula

$$M_e = \frac{\pi}{32} \times \sigma_b \times d^3$$

11. From both the diameter select the diameter whichever is maximum.

2) A steel solid shaft transmitting 15 kW at 200 r.p.m. is supported on two bearings 750 mm apart and has two gears keyed to it. The pinion having 30 teeth of 5 mm module is located 100 mm to the left of the right hand bearing and delivers power horizontally to the right. The gear having 100 teeth of 5 mm module is located 150 mm to the right of the left hand bearing and receives power in a vertical direction from below. Using an allowable stress of 54 MPa in shear, determine the diameter of the shaft. (Nov 2015)

Solution. Given : P = 15 kW = 15 × 10^3 W ; N = 200 r.p.m. ; AB = 750 mm ; T_D = 30 ; m_D = 5 mm ; B_D = 100 mm ;

TC = 100 ; m_C = 5 mm ; AC = 150 mm ; τ = 54 MPa = 54 N/mm^2

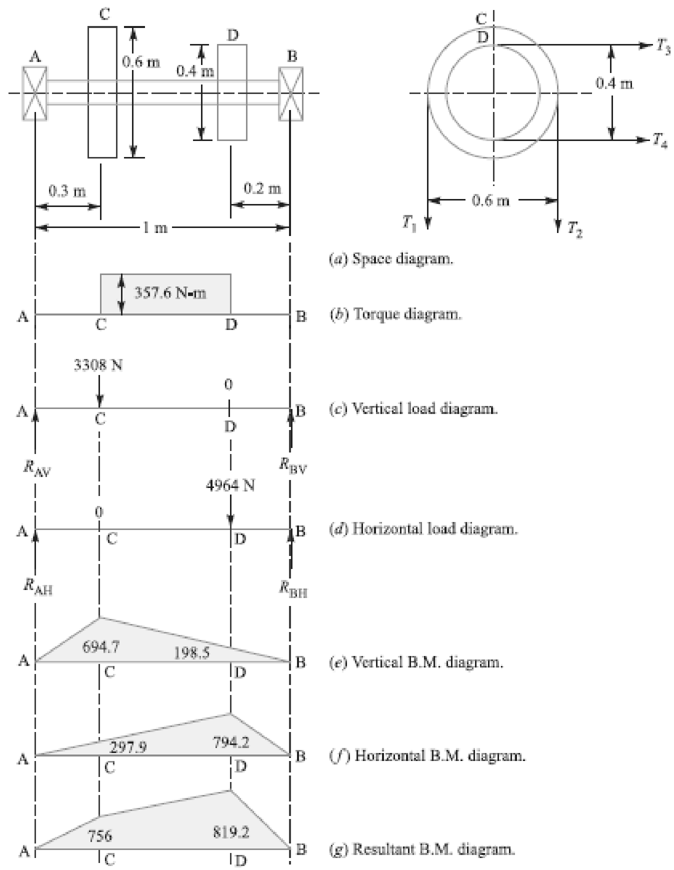
1. Find torque transmitted
2. Radius of gear C and pinion D can be calculated using formula (diameter of gear=no. of teeth x module)
3. Tangential force at C and D
4. Find vertical load acting at C & D in the shaft
5. Find horizontal load acting on C and D
6. Find maximum bending moment for vertical loading and horizontal loading by taking moments at A and B.
7. Find resultant B.M at C(M_C) and D(M_D).

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2}$$

$$M_D = \sqrt{(M_{DV})^2 + (M_{DH})^2}$$

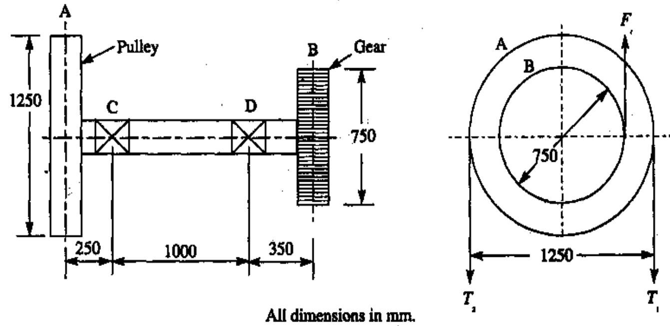
8. select max value and let it be M
9. Find equivalent twisting moment using formula $T_e=\sqrt{(M^2+T^2)}$ and find diameter of shaft using formula

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$



3) A hollow shaft of 0.5 m outside diameter and 0.3 m inside diameter is used to drive a propeller of a marine vessel. The shaft is mounted on bearings 6 metre apart and it transmits 5600 kW at 150 r.p.m. The maximum axial propeller thrust is 500 kN and the shaft weighs 70 kN. Determine(1)The maximum shear stress developed in the shaft, and 2.The angular twist between the bearings. (May 2015)

4)The Figure shows a shaft carrying a pulley A and a gear B and supported in two bearings C and D. The shaft transmits 20 kW at 150 r.p.m. The tangential force Ft on the gear B acts vertically upwards as shown. The pulley delivers the power through a belt to another pulley of equal diameter vertically below the pulley A. The ratio of tensions T1/ T2 is equal to 2.5. The gear and the pulley, weigh 900 N and 2700 N respectively. The permissible shear stress for the material of the shaft may be taken as 63 MPa. Assuming the weight of the shaft to be negligible in comparison with the other loads, determine its diameter. Take shock and fatigue factors for bending and torsion as 2 and 1.5 respectively. (Nov 2012,Nov 2015)



Given : P = 20 kW = 20 × 103 W ; N = 150 r.p.m. ; $T1/T2$ = 2.5 ; W_B = 900 N ;W_A = 2700 N ; τ = 63 MPa = 63 N/mm^2 ; Km = 2 ; Kt = 1.5 ; D_B = 750 mm or R_B = 375 mm ;D_A = 1250 mm or R_A = 625 mm.

1). Torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 150} = 1273 \text{ N-m} = 1273 \times 10^3 \text{ N-mm}$$

∴ Total vertical load acting downward on the shaft at A

$$= T_1 + T_2 + W_A = 3395 + 1358 + 2700 = 7453 \text{ N}$$

Assuming that the torque on the gear B is same as that of the shaft, therefore the tangential force acting vertically upward on the gear B,

$$F_t = \frac{T}{R_B} = \frac{1273 \times 10^3}{375} = 3395 \text{ N}$$

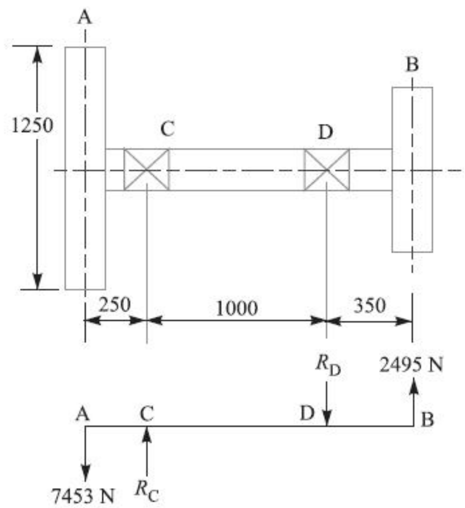
Since the weight of gear B (W_B = 900 N) acts vertically downward, therefore the total vertical load acting upward on the shaft at B

$$= F_t - W_B = 3395 - 900 = 2495 \text{ N}$$

Taking moments about D, we get

$$R_C \times 1000 = 7453 \times 1250 + 2495 \times 350 = 10.2 \times 10^6$$

$$\therefore \quad R_C = 10.2 \times 10^6 / 1000 = 10\,200 \text{ N}$$



2).find the maximum bending moment and maximum torque and substitute in the formula and find the diameter.

$$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2}$$

$$= \sqrt{(2 \times 1863 \times 10^3)^2 + (1.5 \times 1273 \times 10^3)^2}$$

$$= 4187 \times 10^3 \text{ N-mm}$$

$$4187 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 63 \times d^3 = 12.37\, d^3$$

$$d^3 = 4187 \times 10^3 / 12.37 = 338 \times 10^3$$

$$d = 69.6 \text{ say } 70 \text{ mm Ans.}$$

5) Compare the weight, strength and stiffness of a hollow shaft of the same external diameter as that of solid shaft. The inside diameter of the hollow shaft being half the external diameter. Both the shafts have the same material and length. (NOV 2013)

Solution. Given : $do = d$; $di = do / 2$ or $k = di / do = 1 / 2 = 0.5$

find the following.

1. comparison of weight : weight of the hollow shaft,

$$W_H = \text{Cross-sectional area} \times \text{Length} \times \text{Density}$$

$$= \frac{\pi}{4}\left[(d_o)^2 - (d_i)^2\right] \times \text{Length} \times \text{Density}$$

and weight of the solid shaft,

$$W_S = \frac{\pi}{4} \times d^2 \times \text{Length} \times \text{Density}$$

$$\frac{W_H}{W_S} = \frac{(d_o)^2 - (d_i)^2}{d^2} = \frac{(d_o)^2 - (d_i)^2}{(d_o)^2}$$

$$= 1 - \frac{(d_i)^2}{(d_o)_2} = 1 - k^2 = 1 - (0.5)^2 = 0.75 \text{ Ans.}$$

2. Comparison of strength : the strength of the hollow shaft,

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

ınd strength of the solid shaft,

$$T_S = \frac{\pi}{16} \times \tau \times d^3$$

Dividing equation (*iii*) by equation (*iv*), we get

$$\frac{T_H}{T_S} = \frac{(d_o)^3\,(1 - k^4)}{d^3} = \frac{(d_o)^3\,(1 - k^4)}{(d_o)^3} = 1 - k^4$$

$$= 1 - (0.5)^4 = 0.9375 \text{ Ans.}$$

3. Comparison of stiffness : the stiffness

∴ Stiffness of a hollow shaft,

$$S_H = \frac{G}{L} \times \frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right]$$

and stiffness of a solid shaft,

$$S_S = \frac{G}{L} \times \frac{\pi}{32} \times d^4$$

Dividing equation (*v*) by equation (*vi*), we get

$$\frac{S_H}{S_S} = \frac{(d_o)^4 - (d_i)^4}{d^4} = \frac{(d_o)^4 - (d_i)^4}{(d_o)^4} = 1 - \frac{(d_i)^4}{(d_o)^4}$$

$$= 1 - k^4 = 1 - (0.5)^4 = 0.9375 \text{ Ans.}$$

6) Design and make a neat dimensioned sketch of a muff coupling which is used to connect two steel shafts transmitting 40 kW at 350 r.p.m. The material for the shafts and key is plain carbon steel for which allowable shear and crushing stresses may be taken as 40 MPa and 80 MPa respectively. The material for the muff is cast iron for which the allowable shear stress may be assumed as 15 MPa. (Nov 2012)

Solution. Given : P = 40 kW = 40 × 10^3 W;N = 350 r.p.m.; τs = 40 MPa = 40 N/mm^2; σ_{cs} = 80 MPa =80 N/mm^2; τ_c = 15 MPa = 15 N/mm^2

1. find diameter of the shaft. use T = (P * 60) / (2*N*π), $T = \frac{\pi}{16} * \tau_s * d^3$, the diameter is 52mm (approx 55 mm)

2. Design for sleeve : Outer diameter of the muff,

D= 2*d* + 13 mm = 2 × 55 + 13 = 123 say 125 mm **Ans.**

length of the muff,

L = 3.5 ***d*** = 3.5 × 55 = 192.5 say 195 mm **Ans.**

$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_c\left(\frac{D^4 - d^4}{D}\right) = \frac{\pi}{16} \times \tau_c\left[\frac{(125)^4 - (55)^4}{125}\right]$$

$$= 370 \times 10^3\,\tau_c$$

$$\therefore \quad \tau_c = 1100 \times 10^3/370 \times 10^3 = 2.97 \text{ N/mm}^2$$

the induced shear stress in the muff (cast iron) is less than the permissible shear stress of 15 N/mm^2, therefore the design of muff is safe.

3. Design for Key: from PSG DB for shaft diameter d= 55 mm

therefore width of key is w=18 mm= t

Length of the shaft is l = L/2 = 97.5 mm

check the induced shear and crushing stresses in the key, consider shearing of the key, the torque transmitted (T),

$$1100 \times 10^3 = l \times w \times \tau_s \times \frac{d}{2} = 97.5 \times 18 \times \tau_s \times \frac{55}{2} = 48.2 \times 10^3\, \tau_s$$

$$\therefore \quad \tau_s = 1100 \times 10^3 / 48.2 \times 10^3 = 22.8 \text{ N/mm}^2$$

Now considering crushing of the key. We know that torque transmitted (T),

$$1100 \times 10^3 = l \times \frac{t}{2} \times \sigma_{cs} \times \frac{d}{2} = 97.5 \times \frac{18}{2} \times \sigma_{cs} \times \frac{55}{2} = 24.1 \times 10^3\, \sigma_{cs}$$

$$\therefore \quad \sigma_{cs} = 1100 \times 10^3 / 24.1 \times 10^3 = 45.6 \text{ N/mm}^2$$

Since the induced shear and crushing stresses are less than the permissible stresses, therefore the design of key is safe.

7) Design a cast iron protective type flange coupling to transmit 15 kW at 900 r.p.m. from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used : Shear stress for shaft, bolt and key material =40 MPa , Crushing stress for bolt and key= 80 MPa ; Shear stress for cast iron= 8 MPa , Draw a neat sketch of the coupling. (May 2015)

Solution. Given : P = 15 kW = 15 × 10^3 W ; N = 900 r.p.m. ; Service factor = 1.35 ; $\tau_s = \tau_b = \tau_k = 40$ MPa = 40 N/mm^2 ; $\sigma_{cb} = \sigma_{ck}$ = 80 MPa = 80 N/mm^2 ; τ_c = 8 MPa = 8 N/mm^2

The protective type flange coupling is designed as discussed below :

1. Design for hub

find the diameter of the shaft (d). the torque transmitted by the shaft,

use T = (P * 60) / (2*N*π), $T = \frac{\pi}{16} * \tau_s * d^3$; d = 30.1 say 35 mm.

outer diameter of the hub,D = 2d = 2 × 35 = 70 mm Ans.

and length of hub, L = 1.5 d = 1.5 × 35 = 52.5 mm Ans.

Now check the induced shear stress for the hub material which is cast iron. Considering the hub as a hollow shaft. The maximum torque transmitted (Tmax)

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right] = \frac{\pi}{16} \times \tau_c \left[\frac{(70)^4 - (35)^4}{70}\right] = 63\,147\, \tau_c$$

$$\tau_c = 215 \times 10^3 / 63\,147 = 3.4 \text{ N/mm}^2 = 3.4 \text{ MPa}$$

Since the induced shear stress for the hub material (*i.e.* cast iron) is less than the permissible value of 8 MPa, therefore the design of hub is safe.

2. Design for key

Since the crushing stress for the key material is twice its shear stress (i.e. σck = 2τk), therefore a square key may be used. from PSG DB, for a shaft of 35 mm diameter, Width of key, w = 12 mm Ans. and thickness of key, t = w = 12 mm Ans.

The length of key (l) is taken equal to the length of hub.

∴ l = L = 52.5 mm Ans.

check the induced stresses in the key by considering it in shearing and crushing. Considering the key in shearing. We know that the maximum torque transmitted (Tmax),

$$215 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 52.5 \times 12 \times \tau_k \times \frac{35}{2} = 11\,025\, \tau_k$$

$$\tau_k = 215 \times 10^3 / 11\,025 = 19.5 \text{ N/mm}^2 = 19.5 \text{ MPa}$$

Considering the key in crushing. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 52.5 \times \frac{12}{2} \times \sigma_{ck} \times \frac{35}{2} = 5512.5\, \sigma_{ck}$$

$$\sigma_{ck} = 215 \times 10^3 / 5512.5 = 39 \text{ N/mm}^2 = 39 \text{ MPa}$$

Since the induced shear and crushing stresses in the key are less than the permissible stresses, therefore the design for key is safe.

3. Design for flange

The thickness of flange (tf) is taken as 0.5 d. ∴ tf = 0.5 d = 0.5 × 35 = 17.5 mm Ans.

Check the induced shearing stress in the flange by considering the flange at the junction of the hub in shear. the maximum torque transmitted (Tmax),

$$215 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (70)^2}{2} \times \tau_c \times 17.5 = 134\ 713\ \tau_c$$

$$\therefore \qquad \tau_c = 215 \times 10^3 / 134\ 713 = 1.6 \text{ N/mm}^2 = 1.6 \text{ MPa}$$

Since the induced shear stress in the flange is less than 8 MPa, therefore the design of flange is safe.

4. Design for bolts

Let d1 = Nominal diameter of bolts. the diameter of the shaft is 35 mm, therefore let us take the number of bolts, n = 3 and pitch circle diameter of bolts, D_1 = 3d = 3 × 35 = 105 mm

The bolts are subjected to shear stress due to the torque transmitted. the maximum torque transmitted (Tmax),

$$215 \times 10^3 = \frac{\pi}{4} (d_1)^2\ \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4} (d_1)^2\ 40 \times 3 \times \frac{105}{2} = 4950\ (d_1)^2$$

$$\therefore \qquad (d_1)^2 = 215 \times 10^3 / 4950 = 43.43 \text{ or } d_1 = 6.6 \text{ mm}$$

Assuming coarse threads, the nearest standard size of bolt is M 8. Ans.

Other proportions of the flange are taken as follows :

Outer diameter of the flange,

D_2 = 4 d = 4 × 35 = 140 mm Ans.

Thickness of the protective circumferential flange,

t_p = 0.25 d = 0.25 × 35 = 8.75 say 10 mm Ans.

8) Design and draw a protective type of cast iron flange coupling for a steel shaft transmitting 15 kW at 200 r.p.m. and having an allowable shear stress of 40 MPa. The working stress in the bolts should not exceed 30 MPa. Assume that the same material is used for shaft and key and that the crushing stress is twice the value of its shear stress. The maximum torque is 25% greater than the full load torque. The shear stress for cast iron is 14 MPa.

Solution. Given : P = 15 kW = 15 × 10^3 W ; N = 200 r.p.m. ; τs = 40 MPa = 40 N/mm^2 ;

τ_b = 30 MPa = 30 N/mm^2 ; $\sigma_{ck} = 2\tau_k$; Tmax = 1.25 Tmean ; τc = 14 MPa = 14 N/mm^2

The protective type of cast iron flange coupling is designed as discussed below :

1. Design for hub

find the diameter of shaft (d). the full load or mean torque transmitted by the shaft,

use T = (P * 60) / (2*N*π), $T = \frac{\pi}{16} * \tau_s * d^3$; d =48.4 mm say 50 mm

We know that the outer diameter of the hub,

D = 2 d = 2 × 50 = 100 mm Ans.

and length of the hub, L = 1.5 d = 1.5 × 50 = 75 mm Ans.

check the induced shear stress for the hub material which is cast iron, by considering it as a hollow shaft. the maximum torque transmitted (Tmax),

$$895 \times 10^3 = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D}\right) = \frac{\pi}{16} \times \tau_c \left(\frac{(100)^4 - (50)^2}{100}\right) = 184\ 100\ \tau_c$$

$$\tau_c = 895 \times 10^3 / 184\ 100 = 4.86 \text{ N/mm}^2 = 4.86 \text{ MPa}$$

Since the induced shear stress in the hub is less than the permissible value of 14 MPa, therefore the design for hub is safe.

2. Design for key

the crushing stress for the key material is twice its shear stress, therefore a square key may be used.

we find that for a 50 mm diameter shaft, Width of key, w = 16 mm Ans.

and thickness of key, t = w = 16 mm Ans.

The length of key (l) is taken equal to the length of hub. ∴l = L = 75 mm Ans.

check the induced stresses in the key by considering it in shearing and crushing. Considering the key in shearing. the maximum torque transmitted (*Tmax*),

$$895 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 75 \times 16 \times \tau_k \times \frac{50}{2} = 30 \times 10^3 \tau_k$$

$$\therefore \quad \tau_k = 895 \times 10^3 / 30 \times 10^3 = 29.8 \text{ N/mm}^2 = 29.8 \text{ MPa}$$

Considering the key in crushing. We know that the maximum torque transmitted (T_{max}),

$$895 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 75 \times \frac{16}{2} \times \sigma_{ck} \times \frac{50}{2} = 15 \times 10^3 \sigma_{ck}$$

$$\therefore \quad \sigma_{ck} = 895 \times 10^3 / 15 \times 10^3 = 59.6 \text{ N/mm}^2 = 59.6 \text{ MPa}$$

Since the induced shear and crushing stresses in key are less than the permissible stresses, therefore the design for key is safe.

3. Design for flange

The thickness of the flange (tf) is taken as 0.5 d. ; ∴tf = 0.5 × 50 = 25 mm Ans.
check the induced shear stress in the flange, by considering the flange at the junction of the hub in shear. the maximum torque transmitted (Tmax),

$$895 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (100)^2}{2} \times \tau_c \times 25 = 392\,750\ \tau_c$$

$$\therefore \quad \tau_c = 895 \times 10^3 / 392\,750 = 2.5 \text{ N/mm}^2 = 2.5 \text{ MPa}$$

Since the induced shear stress in the flange is less than the permissible value of 14 MPa, therefore the design for flange is safe.

4. Design for bolts

Let d1 = Nominal diameter of bolts. the diameter of shaft is 50 mm, therefore the number of bolts,
n = 4
and pitch circle diameter of bolts,D1 = 3 d = 3 × 50 = 150 mm
The bolts are subjected to shear stress due to the torque transmitted, the
maximum torque transmitted (Tmax),

$$895 \times 10^3 = \frac{\pi}{4} (d_1)^2 \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4} (d_1)^2\, 30 \times 4 \times \frac{150}{4} = 7070\ (d_1)^2$$

$$\therefore \quad (d_1)^2 = 895 \times 10^3 / 7070 = 126.6 \qquad \text{or} \qquad d_1 = 11.25 \text{ mm}$$

Assuming coarse threads, the nearest standard diameter of the bolt is 12 mm (M 12). Ans.
Other proportions of the flange are taken as follows :
Outer diameter of the flange,
D2 = 4 d = 4 × 50 = 200 mm Ans.
Thickness of the protective circumferential flange,
tp = 0.25 d = 0.25 × 50 = 12.5 mm Ans.

9)Design and draw a cast iron flange coupling for a mild steel shaft transmitting 90 kW at 250 r.p.m. The allowable shear stress in the shaft is 40 MPa and the angle of twist is not to exceed 1° in a length of 20 diameters. The allowable shear stress in the coupling bolts is 30 MPa. (Nov 2012)

Solution. Given : P = 90 kW = 90 × 10^3 W ; N = 250 r.p.m. ; τs = 40 MPa = 40 N/mm^2 ;
θ = 1° = π / 180 = 0.0175 rad ; τ_b = 30 MPa = 30 N/mm^2
find the diameter of the shaft (d). the torque transmitted by the shaft,
use T = (P * 60) / (2*N*π), $T = \frac{\pi}{16} * \tau_s * d^3$; d =76 mm
considering rigidity of the shaft, T/ J = C* Ɵ/ l, d= 78 mm say 80 mm.
1). design for hub
2). design of key
3). design for flange
4). design for bolts.

10) Design a bushed pin type of flexible coupling for connecting a motor and a pump shaft. The following data are provided. Power transmitted = 20 kW, speed = 1000 rpm, diameter of the motor and pump shaft = 50 mm , allowable bearing pressure in the rubber bush =0.3MPa.(Nov 2015)

Solution. Given : D = 400 mm ; L = 600 mm or r = 300 mm ; pm = 0.5 N/mm^2 ; p = 2.5 N/mm^2 ;
W = 50 kN ; T1 + T2 = 6.5 kN ; θ = 35° ; p′ = 1N/mm^2 ; l / r = 5
1. Design of the crankshaft when the crank is at the dead centre

(a) Design of crankpin
(b) Design of left hand crank web
(c) Design of right hand crank web
From the balancing point of view, the dimensions of the right hand crank web (i.e. thickness and width) are made equal to the dimensions of the left hand crank web.
(d) Design of shaft under the flywheel
2. Design of the crankshaft when the crank is at an angle of maximum twisting moment
(a) Design of crankpin
(b) Design of shaft under the flywheel
(c) Design of shaft at the juncture of right hand crank arm
(d) Design of right hand crank web
(e) Design of left hand crank web
The dimensions for the left hand crank web may be made same as for right hand crank web.
(f) Design of crankshaft bearings

UNIT - III- DESIGN OF TEMPORARY & PERMANENT JOINTS

PART –A

1 **How is a bolt designated?**

A bolt is designated by a letter M followed by nominal diameter and pitch in mm.

2**. What factors influence the amount of initial tension?**

i. External load ii. Material used iii. Bolt diameter

3. **What is bolt of uniform strength?**

A bolt of uniform strength has equal strength at the thread and shank portion.

4. **What are the ways to produce bolts of uniform strength?**

i. Reducing shank diameter equal to root diameter. ii. Drilling axial hole

5. **What stresses act on screw fastenings?**

i. Initial stresses due to screwing up ii. Stresses due to external forces
iii. Combined stresses.

6**. What are the different applications of screwed fasteners?**

The different applications of screwed fasteners are
a. For readily connecting & disconnecting machine parts without damage
b. The parts can be rigidly connected c. Used for transmitting power

7. **What are the advantages of screwed fasteners? (Nov 2012)**

The advantages of screwed fasteners are (a). They are highly reliable in operation
b. They are convenient to assemble & disassemble c. A wide range of screws can be used for various operating conditions d. They are relatively cheap to produce.

8. **Define pitch.**

Pitch is defined as the distance from appoint on one thread to the corresponding on the adjacent thread in the same axis plane.

9. **Define lead.**

Lead is defined as the distance, which a screw thread advances axially in one rotation of the nut.

10. **What are the different types of metric thread?**

1. BSW (British standard Whit worth) 2. BSE (British standard End

11. **Define welding.**

Welding can be defined as a process of joining two similar or dissimilar metals with or without application of pressure along with or without addition of filler material.

12. **What are the types of welded joints?**

i. Butt joint ii. Lap joint iii. T – joint iv. Corner joint v. Edge joint.

13. **What are the two types of stresses are induced in eccentric loading of loaded joint? (Nov 2012)**

1. Direct shear stress. 2. Bending or Torsional shear stress.

14. **Define butt and lap joint**

Butt joint: The joint is made by welding the ends or edges of two plates.
Lap joint: The two plates are overlapping each other for a certain distance.
Then welded. Such welding is called fillet weld.

15 .**When will the edge preparation need?**

If the two plates to be welded have more than 6mm thickness, the edge preparation should be carried out.

16. **What are the two types of fillet weld?**

i. Longitudinal or parallel fillet weld ii. Transverse fillet weld

17 .**State the two types of eccentric welded connections.**

i. Welded connections subjected to moment in a plane of the weld.

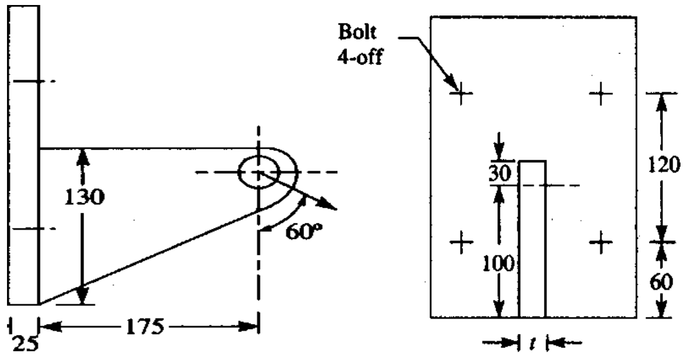
ii. Welded connections subjected to moment in a plane normal to the plane of the weld.

18. **What are the practical applications of welded joints?**

It has employed in manufacturing of machine frames, automobile bodies, aircraft, and structural works.

PART- B

1) Determine the size of the bolts for the bracket as shown in Fig. , if it carries a load of 40kN at an angle of 60°to the vertical. The material of the bracket and the bolts is same for which the safe stress can be assumed as 70MPa in tension. All dimensions in mm



1. Solution

$W = 40kN = 40\times10^3 N$, $\sigma_t = 70MPa = 70N/mm^2$, $\tau = 50MPa = 50N/mm^2$

$\sigma_c = 105MPa = 105N/mm^2$

Note:

1 Mpa = 1 N/mm^2

i, A direct tensile load equally shared by all the four bolts, and

ii, A turning moment about the centre of gravity of the bolts, in the anticlockwise direction

∴ Direct tensile load on each bolt,

$W_{t1} = \frac{W_H}{4} = 8660N$, where $(W_H = W\cos\theta)$

Distance of horizontal component from the centre of gravity (G) of the bolts, $= 60 + 60 - 100 = 20$ mm

∴ Turning moment due to W_H about G,

$T_H = W_H \times 20 = 692.8 \times 10^3$ Nmm (Anticlockwise)

Due to the vertical component W_V the following two effects are produced.

i, A direct shear load equally shared by all the four bolts, and,

ii, A turning moment about the edge of the bracket in the clockwise direction.

∴ Direct shear load on each bolt,

$W_s = \frac{W_V}{4} = 5000N$

Distance of vertical component from the edge E of the bracket, $= 175mm$

∴ Turning moment due to W_V about the edge of the bracket,

$T_V = W_V \times 175 = 3500\times10^3$ Nmm (Clockwise)

From above, we see that the clockwise moment is greater than the anticlockwise moment, therefore

Net turning moment $= 3500\times10^3 - 692.8\times10^3 = 2807.2\times10^3$ Nmm (Clockwise) → ①

Due to this moment, the bracket tends to tilt about the lower edge E,

w - Load on each bolt per mm distance from the edge E due to the turning effect of the bracket.

L_1 = Distance of bolts 1 and 2 from the tilting edge E = 60mm, and

L_2 = Distance of bolts 3 and 4 from the tilting edge E $= 60 + 120 = 180mm$

∴ Total moment of the load on the bolts about the tilting edge E,

$= 2(wL_1)(L_1) + 2(wL_2)(L_2) = 72000\,w$ Nmm → ②

From ① & ②,

$w = 39\,N/mm^2$

Since the heavily loaded bolts are those which lie at a greater distance from the tilting edge, therefore the upper bolts 3 and 4 will be heavily loaded.

∴ the diameter of the bolts should be based on load on the upper bolts.

$W_{t2} = wL_2 = 7020N$

∴ Total tensile load on each of the upper bolt,

$W_t = W_{t1} + W_{t2} = 15680N$

Since each upper bolt is subjected to a tensile load ($W_t = 15680N$) and a shear load ($W_s = 5000N$), therefore equivalent tensile load,

$W_{te} = \frac{1}{2}\left[W_t + \sqrt{W_t^2 + 4W_s^2}\right] = 17140N$ → ③

Size of the bolts

W.K.T, Tensile load on ~~the~~ each bolt, $= \frac{\pi}{4}(d_c^2)\,\sigma_t$ → ④

From ③ & ④,

$d_c = 17.65mm$

We find that from the standard core diameter is 18.933mm and the corresponding $\boxed{\text{size of the bolt is M22}}$.

Thickness of the arm of the bracket:

W.K.T, cross sectional area of the arm,

$A = b \times t = 130t \ mm^2$

Section modulus of the arm,

$Z = \frac{1}{6} t (b^2) = 2817t \ mm^3$

i, Direct tensile stress,

$\sigma_{t1} = \frac{W_H}{A} = \frac{266.5}{t} \ N/mm^2$

ii, Bending stress causing tensile in the upper most fibres of the arm and compressive in the lower most fibres of the arm.

$M_H = W_H \left(100 - \frac{130}{2}\right) = 1212.4 \times 10^3 \ Nmm$.

Bending stress, $\sigma_{t2} = \frac{M_H}{Z} = \frac{430.4}{t} \ N/mm^2$

Due to vertical component W_V, the following two stresses are induced in the arm.

i, Direct shear stress,

$\tau = \frac{W_V}{A} = \frac{154}{t} \ N/mm^2$.

ii, Bending moment $= M_V = W_V (175+25) = 4 \times 10^6 \ Nmm$

Bending stress, $\sigma_{t3} = \frac{M_V}{Z} = \frac{1420}{t} \ N/mm^2$.

Net tensile stress induced in the upper most fibres of the arm of the bracket.

$\sigma_t = \sigma_{t1} + \sigma_{t2} + \sigma_{t3} = \frac{266.5}{t} + \frac{430.4}{t} + \frac{1420}{t} = \frac{2116.9}{t} \ N/mm^2 \rightarrow (5)$

W.K.T,

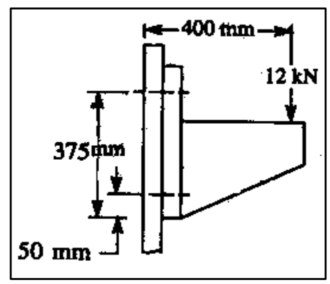
Maximum tensile stress, $\sigma_{t\,max}$,

$70 = \frac{1}{2}\sigma_t + \frac{1}{2}\sqrt{\sigma_t^2 + 4\tau^2} \rightarrow (6)$

sub (5) in (6), we get.

$\boxed{t = 31 \ mm}$

2) For supporting the travelling crane in a workshop, the brackets are fixed on steel columns as shown in Fig. The maximum load that comes on the bracket is 12 kN acting vertically at a distance of 400 mm from the face of the column. The vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at a distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible value of the tensile stress for the bolt material is 84 MPa.



2. Solution:

$W = 12 kN = 12 \times 10^3 N$, $L = 400 \ mm$, $L_1 = 50 mm$, $L_2 = 375 mm$,

$\sigma_t = 84 Mpa = 84 \ N/mm^2$, $n = 4$

W.K.T, Direct shear load on each bolt, $W_s = \frac{W}{n}$

Since the load W will try to tilt the bracket in the clockwise direction about the lower edge, therefore the bolts will be subjected to tensile load due to turning moment.

Maximum tensile load carried by bolts 3 and 4,

$W_t = \frac{W L L_2}{2[(L_1^2) + (L_2^2)]} = 6.29 \ kN$

Since the bolts are subjected to shear load as well as tensile load therefore equivalent tensile load,

$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right] = 7490 N$

Size of the bolt:

W.K.T, the equivalent tensile load (W_{te})

$7490 = \cancel{66} \frac{\pi}{4}(d_c)^2 \sigma_t$

$d_c = 10.65 \ mm$. The corresponding bolt size is M14.

Cross section of the arm of the bracket

Assume that the arm of the bracket extends upto the face of the steel column.

Section modulus, $Z = \frac{1}{6} t b^2$

$\therefore$ Maximum bending moment on the bracket,

$\cancel{M = 12}$ $M = W \times 400 = 4.8 \times 10^6 \ N mm$.

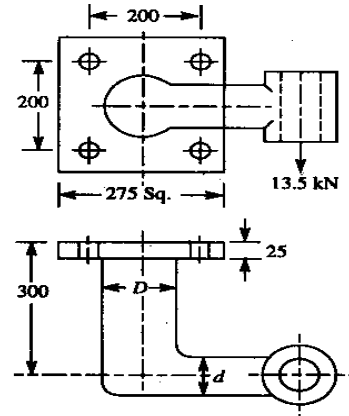
W.K.T, the bending tensile stress,

$84 = \frac{M}{Z}$

$b = 250 mm$

$t = 5.5 mm$

3) Figure shows a solid forged bracket to carry a vertical load of 13.5 kN applied through the centre of hole. The square flange is secured to the flat side of a vertical stanchion through four bolts. Estimate the tensile load on each top bolt and the maximum shearing force on each bolt. Find the bolt size, if the permissible stress is 65 MPa in shear. All dimensions in mm



5. Solution:

$W = 13.5\,kN = 13500\,N$, $\sigma_t = 110\,Mpa = 110\,N/mm^2$, $\tau = 65\,Mpa = 65\,N/mm^2$

Diameter D for the arm of the bracket

The arm D is subjected to bending moment as well as twisting moment,

~~$M = 13.5 \times 10^3 (\quad)$~~

$M = W(300 - 25) = 3712.5 \times 10^3\,Nmm$

Equivalent twisting moment,

$T_e = \sqrt{M^2 + T^2} = 5017 \times 10^3\,Nmm.$

Also, $T_e = \frac{\pi}{16} \tau D^3$

$\boxed{D = 75\,mm}$

Diameter (d) for the arm of the bracket

Bending moment, $M = W\left(250 - \frac{75}{2}\right) = 2868.8 \times 10^3\,Nmm$.

Section modulus, $Z = \frac{\pi}{32} d^3 = 0.0982 d^3$.

Bending tensile stress, σ_t :

$110 = \frac{M}{Z}$

$\boxed{d = 65\,mm}$

Tensile load on each top bolt

Due to the eccentric load, w the bracket has a tendency to tilt about edge E–E.

∴ The total moment of the load of the bolts

$= 2(wL_1^2)(L_2^2) = 115625\,w\,Nmm \longrightarrow (1)$

Turning moment of the load about the tilting edge,

$= WL = 4050 \times 10^3\,Nmm \longrightarrow (2)$

From (1) & (2), $w = 35.03\,N/mm$.

∴ Tensile load on each top bolt $= wL_2 = \boxed{8320\,N}$

Maximum shearing force on each bolt:

W.K.T, shear load on each bolt acting vertically downwards,

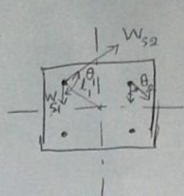
$W_{s1} = \frac{W}{n} = 3375\,N.$

Distance of each bolt from the centre of gravity (G) of the bolts,

$l_1 = l_2 = l_3 = l_4 = \sqrt{l_1^2 + l_2^2} = 141.4\,mm$

Secondary shear load on each bolt,

$W_{s2} = \frac{We l_1}{l_1^2 + l_2^2 + l_3^2 + l_4^2} = 5967\,N$



From the geometry,

$\theta_1 = \theta_4 = 135°$ and $\theta_2 = \theta_3 = 45°$

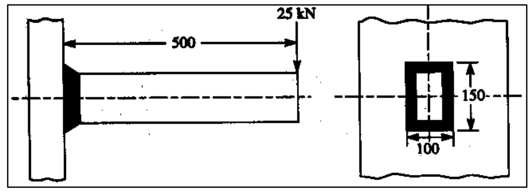
∴ Maximum shearing force on bolts 1 and 4,

$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2W_{s1}W_{s2}\cos 135} = 4303\,N.$

and maximum shearing force on the bolts 2 and 3.

$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2W_{s1}W_{s2}\cos 45°} = 8687\,N$

4). A rectangular cross-section bar is welded to a support by means of fillet welds as shown in Fig. Determine the size of the welds, if the permissible shear stress in the weld is limited to 75 MPa. All dimensions in mm. **(Nov 2012)**



6. Solution:

$P = 25 kN = 25 \times 10^3 N$, $\tau_{max} = 75 Mpa = 75 N/mm^2$, $l = 100 mm$

$b = 150 mm$, $e = 500 mm$.

The joint is subjected to direct shear stress and the bending stress. W.K.T, the throat area for a rectangular fillet weld.

$A = t(2b + 2l)$

$= 0.707 s \ (2b + 2l) = 353.55 \ mm^2$

∴ Direct shear stress, $\tau = \frac{P}{A} = \frac{70.72}{s} \ N/mm^2$.

Bending moment, $M = P \times e = 12.5 \times 10^6 N mm$

For rectangular section, section modulus

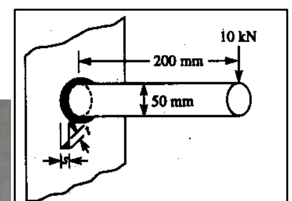
$Z = t\left(bl + \frac{b^2}{3}\right) = 15907.5 s \ mm^3$

Bending stress, $\sigma_b = \frac{M}{Z} = \frac{785.8}{s} \ N/mm^2$.

Maximum shear stress, τ_{max}.

$75 = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2} = \frac{391.2}{s}$ $\boxed{S = 5.32 mm}$

5). A 50 mm diameter solid shaft is welded to a flat plate as shown in Fig. If the size of the weld is 15 mm, find the maximum normal and shear stress in the weld.



Solution:

$D = 50mm$, $s = 15mm$, $P = 10 KN = 10,000N$ $e = 200 mm$.

The joint is subjected to direct stress and bending stress.

∴ The throat area of the fillet, $A = t \times \pi D = 1666 \ mm^2$

Direct shear stress, $\tau = \frac{P}{A} = 6 N/mm^2$

Bending moment, $M = P \times e = 2 \times 10^6 \ N mm$

For circular section, section modulus,

$Z = \frac{\pi t D^2}{4} = 20825 \ mm^3$.

∴ Bending stress, $\sigma_b = \frac{M}{Z} = 96 Mpa$

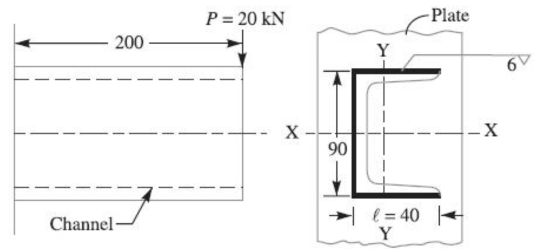
Maximum normal stress,

$\sigma_{t(max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} = 96.4 Mpa$

Maximum shear stress,

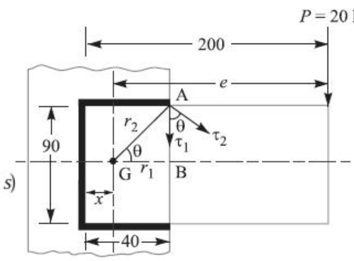
$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} = 48.4 Mpa$.

6). Find the maximum shear stress induced in the weld' of 6 mm size when a channel, as shown in figure, is welded to a plate, and loaded with 20 kN force at a distance of 200 mm.



Solution. Given : s = 6 mm; P = 20 kN = 20 × 103N ; l = 40 mm ; b = 90 mm, t = Throat thickness.
find the centre of gravity (G) of the weld system $x$be the distance of centre of gravity from the left hand edge of the weld system.

$$x = \frac{l^2}{2l+b} = \frac{(40)^2}{2\times 40+90} = 9.4 \text{ mm}$$



and polar moment of inertia of the throat area of the weld system about G,

$$J = t\left[\frac{(b+2l)^3}{12} - \frac{l^2\,(b+l)^2}{b+2l}\right]$$

$$= 0.707\,s\left[\frac{(90+2\times 40)^3}{12} - \frac{(40)^2\,(90+40)^2}{90+2\times 40}\right] \quad \ldots (\because\ t = 0.707\,s)$$

$$= 0.707 \times 6\,[409.4\times 10^3 - 159\times 10^3] = 1062.2\times 10^3 \text{ mm}^4$$

Distance of load from the centre of gravity (G), *i.e.* eccentricity,

$$e = 200 - x = 200 - 9.4 = 190.6 \text{ mm}$$
$$r_1 = BG = 40 - x = 40 - 9.4 = 30.6 \text{ mm}$$
$$AB = 90/2 = 45 \text{ mm}$$

maximum radius of the weld,

$$r_2 = \sqrt{(AB)^2 + (BG)^2} = \sqrt{(45)^2 + (30.6)^2} = 54.4 \text{ mm}$$

$$\therefore \quad \cos\theta = \frac{r_1}{r_2} = \frac{30.6}{54.4} = 0.5625$$

throat area of the weld system,

$$A = 2\times 0.707s\times l + 0.707s\times b = 0.707\,s\,(2l+b)$$
$$= 0.707\times 6\,(2\times 40+90) = 721.14 \text{ mm}^2$$

∴ Direct or primary shear stress,

$$\tau_1 = \frac{P}{A} = \frac{20\times 10^3}{721.14} = 27.7 \text{ N/mm}^2$$

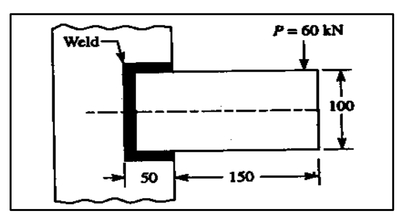
and shear stress due to the turning moment or secondary shear stress,

$$\tau_2 = \frac{P\times e\times r_2}{J} = \frac{20\times 10^3\times 190.6\times 54.4}{1062.2\times 10^3} = 195.2 \text{ N/mm}^2$$

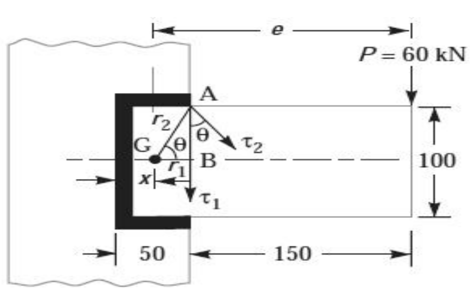
resultant or maximum shear stress,

$$\tau = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\tau_1\times\tau_2\times\cos\theta}$$
$$= \sqrt{(27.7)^2 + (195.2)^2 + 2\times 27.7\times 195.2\times 0.5625}$$
$$= 212 \text{ N/mm}^2 = 212 \text{ MPa Ans.}$$

7) A rectangular steel plate is welded as a cantilever to a vertical column and supports a single concentrated load P, as shown in Fig. Determine the weld size if shear stress in the same is not to exceed 140 MPa. All dimensions in mm **. (Nov 2012)**



Solution. Given : P = 60 kN = 60 × 103N ; b = 100 mm ; l = 50 mm ; τ = 140 MPa = 140 N/mm^2 , s = Weld size, and t = Throat thickness.



$$x = \frac{l^2}{2l + b} = \frac{(50)^2}{2 \times 50 + 100} = 12.5 \text{ mm}$$

and polar moment of inertia of the throat area of the weld system about G,

$$J = t\left[\frac{(b+2l)^3}{12} - \frac{l^2\ (b+l)^2}{b+2l}\right]$$

$$= 0.707\,s\left[\frac{(100+2\times 50)^3}{12} - \frac{(50)^2\ (100+50)^2}{100+2\times 50}\right] \ldots (\because\ t = 0.707\ s)$$

$$= 0.707s\ [670 \times 10^3 - 281 \times 10^3] = 275 \times 10^3\ s \text{ mm}^4$$

Distance of load from the centre of gravity (G) *i.e.* eccentricity,

$$e = 150 + 50 - 12.5 = 187.5 \text{ mm}$$

$$r_1 = BG = 50 - x = 50 - 12.5 = 37.5 \text{ mm}$$

$$AB = 100 / 2 = 50 \text{ mm}$$

We know that maximum radius of the weld,

$$r_2 = \sqrt{(AB)^2 + (BG)^2} = \sqrt{(50)^2 + (37.5)^2} = 62.5 \text{ mm}$$

$$\therefore \qquad \cos\theta = \frac{r_1}{r_2} = \frac{37.5}{62.5} = 0.6$$

We know that throat area of the weld system,

$$A = 2 \times 0.707s \times l + 0.707s \times b = 0.707\ s\ (2l + b)$$

$$= 0.707s\ (2 \times 50 + 100) = 141.4\ s \text{ mm}^2$$

$\therefore$ Direct or primary shear stress,

$$\tau_1 = \frac{P}{A} = \frac{60 \times 10^3}{141.4\,s} = \frac{424}{s} \text{ N/mm}^2$$

and shear stress due to the turning moment or secondary shear stress,

$$\tau_2 = \frac{P \times e \times r_2}{J} = \frac{60 \times 10^3 \times 187.5 \times 62.5}{275 \times 10^3\ s} = \frac{2557}{s} \text{ N/mm}^2$$

the resultant shear stress,

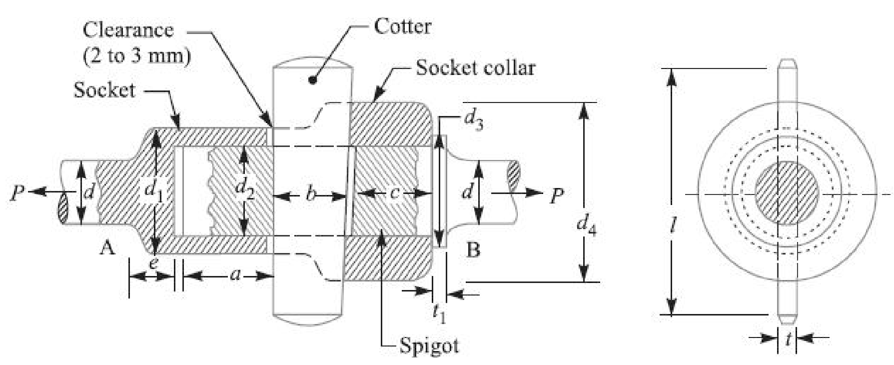
$$\tau = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\,\tau_1 \times \tau_2 \times \cos\theta}$$

$$140 = \sqrt{\left(\frac{424}{s}\right)^2 + \left(\frac{2557}{s}\right)^2 + 2 \times \frac{424}{s} \times \frac{2557}{s} \times 0.6} = \frac{2832}{s}$$

$$s = 2832 / 140 = 20.23 \text{ mm} \quad \text{Ans.}$$

8). Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied statically. Tensile stress = compressive stress = 50 MPa ; shear stress = 35 MPa and crushing stress= 90 MPa.

Solution. Given : P = 30 kN = 30 × 10^3 N ; σt = 50 MPa = 50 N / mm^2 ; τ = 35 MPa = 35 N / mm^2 ; σc = 90 MPa = 90 N/mm^2



Solution:

$P = 30kN = 30\times10^5 N$, $\sigma_t = 50Mpa = 50N/mm^2$

$\tau = 35Mpa = 35N/mm^2$, $\sigma_c = 90Mpa = 90N/mm^2$.

1. Diameter of the rods

Let d = Diameter of the rods.

consider failure of the rod in tension.

$$30\times10^3 = \frac{\pi}{4} d^2 \sigma_t$$

$$d = 28 mm$$

2. Diameter of spigot and thickness of cotter:

consider failure of spigot in tension across the weakest section,

$$30\times10^3 = \left[\frac{\pi}{4} d_2^2 - d_2 t\right]\sigma_t$$

$$d_2 = 40mm, \quad t = \frac{d_2}{4} = 10mm$$

3. Outer diameter of socket.

Consider the failure of the socket in tension across the slot.

$$30\times10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)t\right]\sigma_t$$

$$d_1 = 50mm$$

4. Width of cotter:

$$30\times10^3 = 2b \times t \times \tau$$

$$b = 43 mm$$

5. Diameter of socket collar:

Failure of the socket collar and cotter in crushing.

$$30\times10^3 = (d_4 - d_2)\, t\, \sigma_c$$

$$d_4 = 75mm$$

6. Thickness of socket collar

Consider failure of the socket end in shearing

$$30\times10^3 = 2(d_4 - d_2)\,c \times \tau$$

$$c = 12\ \text{mm}$$

7. Distance from the end of the slot to the end of the r

Consider failure of the rod end in shear.

$$30\times10^3 = 2a\,d_2\,\tau$$

$$a = 11\ \text{mm}$$

8. Diameter of spigot collar

considering the failure of spigot collar in crushing.

$$30\times10^3 = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

$$d_3 = 45\ \text{mm}$$

9. Thickness of spigot collar:

Considering the failure of spigot collar in shearing.

$$30\times10^3 = \pi d_2 t_1 \tau$$

$$t_1 = 8\ \text{mm}$$

10. The length of cotter (l) is taken as $4d$.

$$l = 4d = 112\ \text{mm}.$$

11. The dimension e is taken as $1.2d$

$$e = 1.2d = 34\ \text{mm}$$

9). Design a knuckle joint to transmit 150 kN. The design stresses may be taken as75 MPa in tension, 60 MPa in shear and 150 MPa in compression.

Solution. Given : $P = 150\ \text{kN} = 150 \times 10^3\ \text{N}$; $\sigma t = 75\ \text{MPa} = 75\ \text{N/mm}^2$; $\tau = 60\ \text{MPa} = 60\ \text{N/mm}^2$; $\sigma c = 150\ \text{MPa} = 150\ \text{N/mm}^2$

Solution:

$P = 150\,kN = 150\times10^3 N$, $\sigma_t = 75\,Mpa = 75 N/mm^2$,

$\tau = 60\,MPa = 60 N/mm^2$, $\sigma_c = 150\,Mpa = 150 N/mm^2$

1. Failure of the solid rod in tension

$$P = \frac{\pi}{4} d^2 \sigma_t$$
$$d = 52\,mm$$
$$d_1 = d = 52\,mm$$
$$d_2 = 2d = 104\,mm$$
$$d_3 = 1.5d = 78\,mm$$
$$t = 1.25d = 65\,mm$$
$$t_1 = 0.75d = 40\,mm$$
$$t_2 = 0.5d = 26\,mm$$

2. Failure of the knuckle pin in shear

$$P = 2\times\frac{\pi}{4}\times d_1^2\times\tau \qquad (\because \text{Double shear})$$
$$\tau = 35.3\,Mpa$$

3. Failure of the single eye (or) rod end in tension

$$P = (d_2 - d_1)\,t\times\sigma_t$$
$$\sigma_t = 44.4\,Mpa$$

4. Failure of the single eye (or) rod end in shear

$$P = (d_2 - d_1)\,t\times\tau$$
$$\tau = 44.4\,Mpa$$

5. Failure of the single eye or rod end in crushing

$$P = d_1\times t\times\sigma_c$$
$$\sigma_c = 44.4\,Mpa$$

6. Failure of the forked end in tension.

7. Failure of the forked end in shear.

$$P = (d_2 - d_1)\,2t_1\,\tau$$
$$\tau = 36\,Mpa$$

8. Failure of the forked end in crushing

$$P = d_1\times 2t_1\times\sigma_c$$
$$\sigma_c = 36\,Mpa$$

10).Design a double riveted butt joint with two cover plates for the longitudinal seam of a boiler shell 1.5 m in diameter subjected to a steam pressure of 0.95 N/mm^2. Assume joint efficiency as 75%, allowable tensile stress in the plate 90 MPa ; compressive stress 140 MPa ; and shear stress in the rivet 56 MPa.

Solution:

$D = 1.5m = 1500mm$, $p = 0.95 N/mm^2$, $\eta_l = 75\% = 0.75$,

$\sigma_t = 90 MPa = 90 N/mm^2$, $\sigma_c = 140 MPa = 140 N/mm^2$, $Z = 56 MPa = 56 N/mm^2$

1. Thickness of boiler shell plate

$$t = \frac{P \cdot D}{2\sigma_t \times \eta_l} + 1 = 12 mm$$

2. Diameter of rivet:

$$d = 6\sqrt{t} = 20.8 mm$$

Corresponding diameter of the rivet is 20 mm.

3. Pitch of rivets

W.K.T, Tearing resistance of the plate

$$P_t = (p - d) t \times \sigma_t = 1080(p - 21) \longrightarrow ①$$

Assuming that the rivets in double shear are 1.875 times stronger than in ~~shea~~ single shear.

Shearing strength of the rivets,

$$P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times Z$$

$$= 72745 N \longrightarrow ②$$

From ① & ②,

$$\boxed{p = 90 mm}$$

The maximum pitch of rivets for longitudinal joint of a boiler is given by,

$$p_{max} = C \times t + 41.28 \text{ mm.}$$ (From DDB. Value of C is 3.5)

$$= 84 mm$$

4. Distance between rows of rivets:

$$P_b = 0.33p + 0.67d = 42 mm.$$

5. Thickness of cover plates

$$t_1 = 0.625t = 7.5 mm$$

6. Margin:

$$m = 1.5d = 32 mm$$

Tearing resistance of the plate, $P_t = (p - d) t \times \sigma_t = 68040 N$

Shearing resistance of the rivets, $P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times Z = 72745 N$

Crushing resistance of the rivets, $P_c = n \times d \times t \times \sigma_c = 70560 N$.

The least value of P_t, P_s, or P_c is,

$$P_t = 68040 N.$$

Strength of unriveted plate,

$$P = p \times t \times \sigma_t = 90720 N.$$

$$\eta = \frac{P_t}{P} = 75\%.$$

Since the efficiency of designed joint is equal to the given efficiency of 75%, therefore the design is satisfactory.

11).Find the efficiency of the following riveted joints :

1. Single riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 50 mm.
2. Double riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 65 mm.

Assume, Permissible tensile stress in plate = 120 MPa

Permissible shearing stress in rivets = 90 MPa

Permissible crushing stress in rivets = 180 MPa

Solution. Given : t = 6 mm ; d = 20 mm ; σt = 120 MPa = 120 N/mm^2 ; τ = 90 MPa = 90 N/mm^2 ; σc = 180 MPa = 180 N/mm^2

Solution

$t = 6mm$, $d = 20mm$, $\sigma_t = 120 Mpa = 120 N/mm^2$, $\tau = 90 Mpa = 90 N/mm^2$

$\sigma_c = 180 Mpa = 180 N/mm^2$.

1. Efficiency of the first joint

pitch, $p = 50mm$

The rivet is subjected to tearing force, shearing force and crushing force. Therefore, it is required to find out tearing resistance, shearing and crushing resistance of the ~~pt~~ rivets.

i, Tearing resistance of the plate

$$P_t = (p-d)\, t \times \sigma_t = 21600 N.$$

ii, Shearing resistance of the rivet

$$P_s = \frac{\pi}{4} d^2 \tau = 28278 N.$$

iii, Crushing resistance of the rivet

$$P_c = d \times t \times \sigma_c = 21,600 N$$

Strength of the joint = Least of P_t, P_s and P_c = 21,600N

W.K.T, Strength of the unriveted or solid plate,

$$P = p \times t \times \sigma_t = 36,000 N$$

$$\text{Efficiency of the joint, } \eta = \frac{\text{Least of } P_t, P_s, \text{and } P_c}{P} = 60\%$$

2. Efficiency of the second joint

pitch, $p = 65mm$

i, Tearing resistance of the plate,

$$P_t = (p-d)\, t \times \sigma_t = 32400N$$

ii, Shearing resistance of the rivets,

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$$ (Here $n=2$ for double riveted lap joint)

$$= 56556 N$$

iii, Crushing resistance of the rivets,

$$P_c = n \times d \times t \times \sigma_c = 43,200 N$$

Strength of rivets = Least of P_t, P_s and P_c = 32400N

Strength of unriveted joint = $p \times t \times \sigma_t$ = 46800N

$$\eta = \frac{\text{Least of } P_t, P_s, \text{and } P_c}{P} = 69.2\%$$