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DEPARTMENT OF MECHANICAL ENGINEERING

ME8593 – DESIGN OF MACHINE ELEMENTS

COURSE HANDOUTS

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

1. **Factor of safety**

The ratio between maximum stresses to working stress is known as factor of safety.

2. **Endurance limit**.

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.

3. **Impact load**

If the time of load application is less than one third of the lowest natural period of vibration of the part, it is called an impact load.

4. **Stress concentration and stress concentration factor.**

Stress concentration is the increase in local stresses at points of rapid change in cross section or discontinuities.

Stress concentration factor is the ratio of maximum stress at critical section to the nominal stress

5. **Griffith theory. (Or) State the condition for crack growth.**

A crack can propagate if the energy release rate of crack is greater than crack resistance

6. **Notch sensitivity (q)**

Notch sensitivity (q) is the degree to which the theoretical effect of stress Concentration is actually reached.

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7. **Machinability**

It is the property of the material, which refers to a relative case with which a material can be cut. It is measured in a number of ways such as comparing the tool life for cutting different material.

8. Ductility

It is the property of the material enabling it to be drawn into wire, with the application of tensile force. It must be both strong and plastic. It is usually measured in terms of percentage elongation and reduction in area. (eg) Ni, Al, Cu

9. Fatigue

When a material is subjected to repeated stress, it fails at stresses below the yield point stress; such type of failure of the material is called fatigue.

1. 1 INTRODUCTION TO MACHINE DESIGN

Machine design is the creation of new and better machines and improving the existing one. A new or better machine is one which is more economical in the overall cost of production and operation.

1.1.1 CLASSIFICATION OF MACHINE DESIGN

1. Adaptive design: The designers work is concerned with adaptation of existing design.

2. Development design: This type of design needs considerable scientific training and design ability in order to modify the existing designs into a new idea.

3. New design: This type of design needs a lot of research technical ability and designers and creative thinking.

1.1.2 GENERAL CONSIDERATION IN MACHINE DESIGN

(i) Type of load and stresses caused by the load.

(ii) Motion o parts

(iii) Selection of materials

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(iv) Frictional resistance and lubrication

(v) Convenient and economical features

(vi) Safety of operation

1.1.3 FACTORS INFLUENCING MACHINE DESIGN

(i) Strength and stiffness

(ii) Surface finish and tolerances

(iii) Manufacturability

(iv) Ease of handling

(v) Working atmosphere

(vi) Cooling and lubrication

(vii) Safety

(viii) Noise requirement

(ix) Cost

1.2 BENDING STRESS IN STRAIGHT BEAMS

Consider a straight beam subjected to a bending moment M as shown in figure. The following assumption are usually made delivering the bending formula

(i) The material of the beam is perfectly homogeneous and isotropic.

(ii) The material of the beam obeys Hooks la

(iii) The Young‟s modulus is same in tension and compression.

(iv) The loads are applied in the plane of bending.

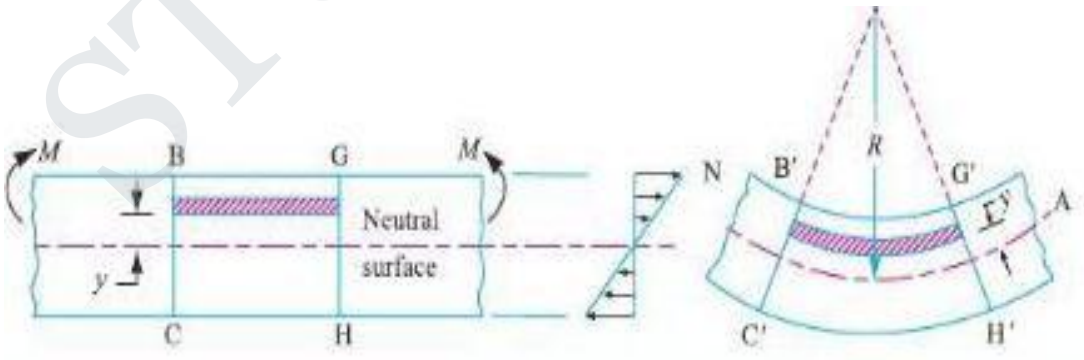

Figure1.1 Bending Stress in Straight Beams

When a beam is subjected to the bending moment the fibers on the upper side of the beam will be compress and lower side elongate due to tension. The surface between the top and bottom fibbers are neither neither shortens nor lengthened. Such a surface is called neutral surface. The

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intersection of the neutral surface with any normal cross section of the beam is known is neutral axis. The bending equation is given by

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$$

M- Bending moment acting at the given section

σ- bending stress

I-moment of inertia of the cross section about the neutral axis

y- Distance from the neutral axis to the extreme fibber

E- Young‟s modulus of the material of the beam

R- Radius of curvature of the beam

1.3BENDING STRESS IN CURVED BEAMS

In straight beams the neutral axis of the section coincides with its centroidal axis and the stress distribution is linear. But in curved beams the neutral axis of the cross section is shifted towards the centre of curvature of the beam causing a nonlinear distribution of stress. Application of curved beam principle is used in crane hooks, chain links planers etc

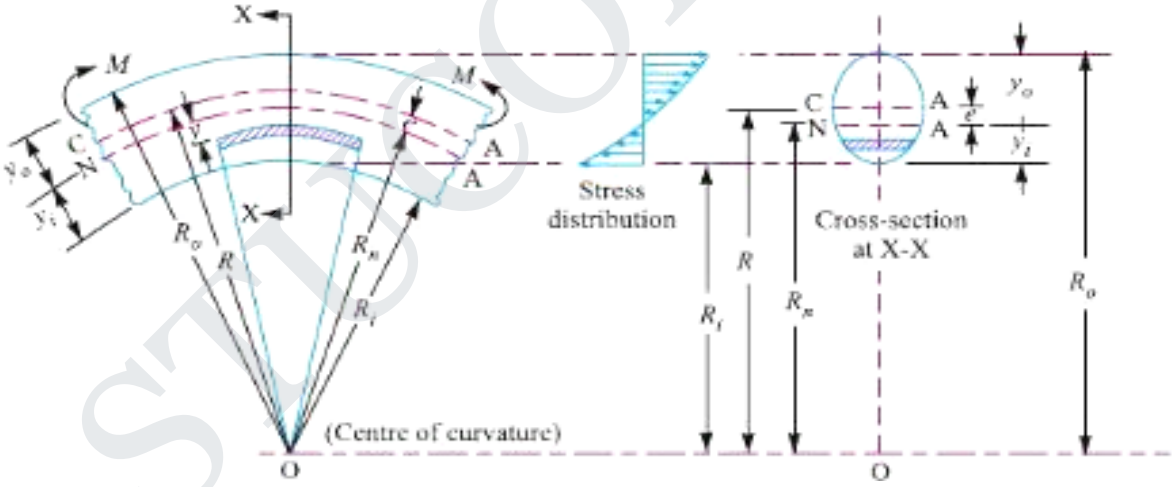

Figure1.2 Bending Stress in Curved Beams

Consider a curved beam subjected to a bending moment M as shown in figure. The general expression for bending stress σ_b) in a curved beam at any fiber at a distance y from the neutral axis is

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$$\sigma_b = \frac{M}{Ae}\left(\frac{y}{R_n - y}\right)$$

M- Bending moment acting at the given section about the centroidal axis

A- Area of cross-section e- Radius of curvature of the neutral axis

R- Radius of curvature of the centroidal axis

Rn- radius of curvature of the neutral axis

y- Distance from the neutral axis to the fiber under consideration.

It is positive for the distances towards the center of curvature and negative for the distance away from the center of curvature.

1.4 STRESS CONCENTRATION

When every a machine component changes the shape of cross section the simple stress distribution no longer holds good and the neighborhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. Consider a plate with transverse elliptical hole and subjected to a tensile load as shown in figure. From the figure the stress at the point away from the hole is practically uniform and the maximum stress will be induced at the edge of the hole.

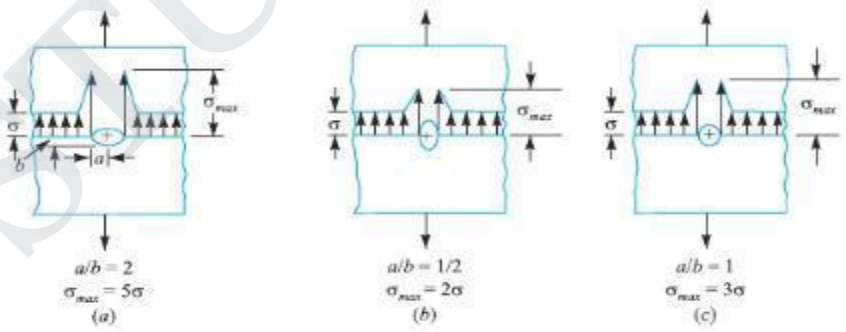

Fig 1.3. k_t = max stress/nominal stress

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Methods of Reducing Stress Concentration

- Avoiding sharp corners
- Providing fillets
- Use of multiple holes instead of single hole.
- Undercutting the shoulder part

1.5 THEORIES OF FAILURE

Stress produce in a body due to the application of the load is beyond the elastic limit the permanent deformation occurs in the body. If the load is removed the body will not retain its original shape. There are some permanent deformations in the body. Whenever permanent deformation occurs in the body the body is said to be failed. The failure of a component due to increase of tensile stress or due to other quantities such as shear stress and strain energy also attain definite values and any one of these may be deciding factor of the failure have advanced to explain the cause of failure. According to the important theories the failure takes place when a certain limiting value is reached by one of the following

1.5.1 Maximum principal stress (or) maximum normal stress (or) Ranking theory

Failure occurs when the maximum normal stress is equal to the tensile yield strength.

$$\sigma_1(\text{or})\ \sigma_2(\text{or})\ \sigma_3(\text{which is maximum}) = \sigma_y/n \text{ (for ductile material)}$$
$$\sigma_1(\text{or})\ \sigma_2(\text{or})\ \sigma_3(\text{which is maximum}) = \sigma_u/n \text{ (for brittle material)}$$

This theory is based on failure in tensile or compression and ignores the possibility of failure due to shearing stress, ductile material mostly fail by shearing. So this theory is used for brittle material.

1.5.2 Maximum shear theory (or) Guest's theory (or) Coloumb theory

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Failure occurs when the maximum shear stress developed in the machine member become equal to the maximum shear stress at yielding in a tensile test.

$$(\sigma_1-\sigma_2) \text{ or } (\sigma_2-\sigma_3) \text{ or } (\sigma_3-\sigma_1) = \sigma_y/n$$

This theory is mostly used for ductile materials.

1.5.3 Maximum strain theory (or) Venant's theory

Failure occurs when the maximum strain in the member equal in the tensile yield strain.

$$\sigma_1-\nu(\sigma_2+\sigma_3) \text{ (or) } \sigma_2-\nu(\sigma_3+\sigma_1) \text{ (or) } \sigma_3-\nu(\sigma_1+\sigma_2) =\sigma_y/n$$

1.5.4 Maximum strain energy theory

Failure is induced in the member when the strain energy stored per unit volume of the member becomes equal to the strain energy per unit volume at the yield point.

$$\sigma_1^2+\sigma_2^2+\sigma_3^2-2\nu(\sigma_1\sigma_2+\sigma_2\sigma_3+\sigma_3\sigma_1) = (\sigma_y/n)^2$$

1.5.5 Distortion energy theory (Vonmiseshenky theory)

$$\sigma_1^2+\sigma_2^2+\sigma_3^2-\sigma_1\sigma_2-\sigma_2\sigma_3-\sigma_3\sigma= (\sigma_y/n)^2$$

1.6. DESIGN OF VARIABLE LOADING

Consider a rotating beam of circular cross section and carrying a load of W, this load induces stresses in the beam which are cyclic in nature.

Upper fibres of beam(a) under compression and lower fibre (B) tensile after half revolution the point B occupies the position of point A and point A occupies the point of B. thus point B is now compression and point A is tensile.

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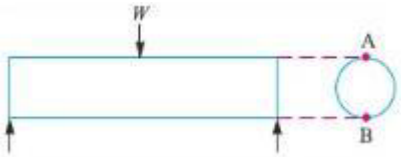

Fig 1.4 simply supported beam with center point load

The stresses which vary from one value of compressive to same value of tensile or vice versa are known as completely reversed or cyclic stresses.

The stresses which vary from a minimum value to a maximum value of same nature is called fluctuating 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 is called alternative stresses (from a certain minimum compressive to a maximum tensile or minimum tensile to a certain maximum compressive)

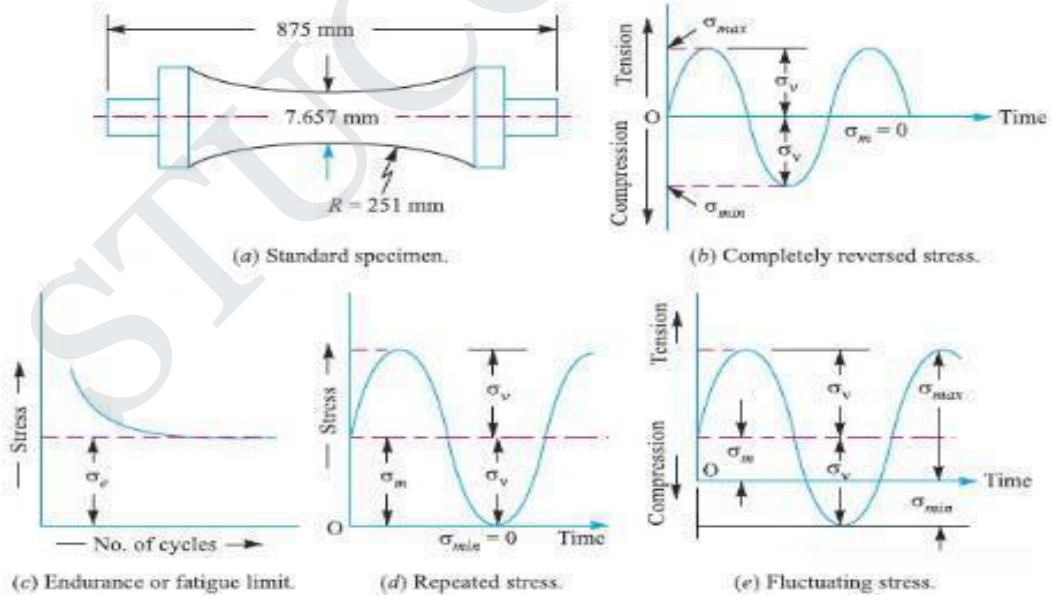

fig 1.5 variable stresses

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This is defined as the degree to which the actual stress concentration effect compares with theoretical stress concentration effect.

1.6.2 ENDURANCE LIMIT

It is defined as maximum value of completely reversed bending stress which a polished specimen can withstand without failure for infinite number of cycles.

1.6.3 FACTORS AFFECTING ENDURANCE STRENGTH

1. Surface finish factor (KSF)
2. Size factor (KSZ)
3. Reliability factor (KR)
4. Miscellaneous factors (K)
5. Load factor (KL)

1.7 SOLVED PROBLEMS

1. A 20KN tensile load acts on the following members. Considering stress concentration, calculate the maximum stress induced in each member.

(a)A stepped shaft of diameter 50mm and 25mm with fillet radius 5mm. (b) a rectangle plate 60mm wide and 10mm thick with a transverse hole of 12mm diameter.

Solution:

Case1: stepped shaft**:**

Tensile load , p=20 KN =20,000N

Maximum diameter, D=50mm

Minimum diameter, d=25mm

Fillet radius=5mm.

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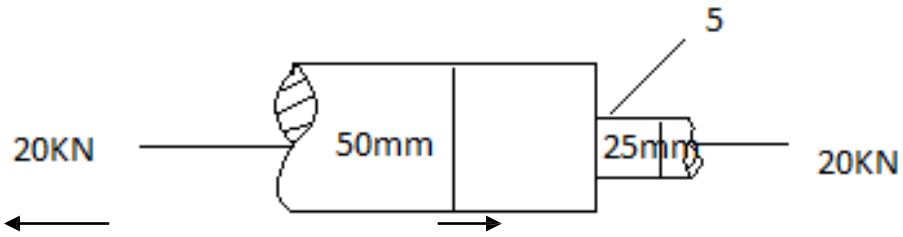


In stepped shaft, due to minimum cross-section more stress is induced in small shaft than the bigger shaft.

Hence,

Nominal stress induced in small shaft, $\sigma o=P/A=P/(\pi/4x^{d^2})$

$= (4X20, 000)/\pi x^{25^2}$

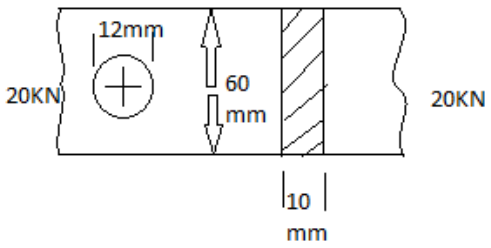
$=40.7N/mm^2$

Also, due to change of diameters, stress concentration factor should be considered for obtaining maximum induced stress.

Now, r/d=5/25=0.2 and D/d=50/25=2

Hence, the stress concentration factor for the above ratio conditions is, Kt=1.5Maximum induced stress $=K_tX\sigma_o=1.5X40.7=61N/mm^2$

Case2: Rectangular plate



Tensile load, P=20KN=20,000N

Width of the plate, b=60mm

Thickness of plate, t=10mm

Diameter of hole, d=12mm

Here also, more stress is induced in the minimum cross section of the plate ,where hole is present.

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Now, nominal stress induced, σ_o =P/A=P/(b-d)t

=20,000/(60-12)10 =41.7N/mm^2

Also, ratio of diameter of hole to width of plate,

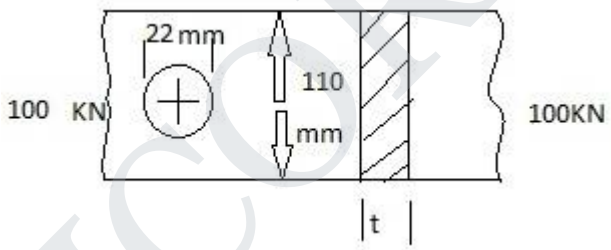
D/b=12/60 =0.2

Stress concentration factor, Kt=2.5 for d/b=0.2(from graph 4.99)

Maximum induced stress =$K_t X\sigma_o$ =2.5X41.7 =104N/mm2

2. **A tie-bar has to carry a load of 100KN. What must be the thickness of bar of 110mm Width, if there is the rivet hole of 22mm diameter on its centre line? Working stress for the tie bar is 75Mpa**.

Solution:



Load =100KN =100X10^{3}N

Width of plate, b =110mm

Diameter of hole, d=22mm

Thickness 0f plate =t mm

Working stress, σt=75Mpa=75x10^{6}N/m^2=75N/mm^2

Nominal stress induced in the cross-section passing across hole

σ_o=P/A=P/(b-d)t=(100x10^3)/(110-22)t=1136/t

Now, d/b=22/110=0.2, for which K_t=2.5

Hence, maximum stress=$K_t x\sigma_o$=2.5x(1136/t)=(2840/t)

This maximum stress should be less than working stress for safe design.

i.e., (2840/t) ≤75

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(or) t >(2840/75) > 37.8mm= 40mm

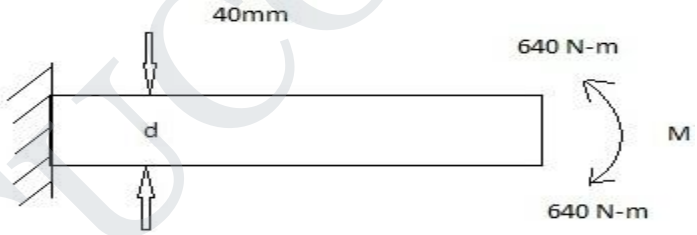
Thickness of plate=40mm.

3. **A round bar of dia40mm has a smooth ground surface and is made of steel for which ultimate strength is 500N/mm^2 yield strength is 400N/mm^2 and endurance limit is 250N/mm^2 .**

Determine the factor of safety of the bar when it is subjected to alternating bending stress in a symmetric cycle if the maximum bending moment in the cycle is 640 N-m.

Solution.

Given



Diameter of the shaft d=40mm

σ_u=500N/mm ^{2}

σ_y=400N/mm^2

σ_e=250N/mm^2

Maximum bending moment M_{ama}=640N-m=640x10^{3}N-mm

Maximum bending stress,

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$\sigma_{bmax}=32\ M_{max}/\pi d^3$

$= (32x640x10^3)/\pi x40^3$

$=102N/mm^2$

Since this is a symmetric cycle,

$\sigma_{bmin}=-102N/mm^2$

then, mean stress $\sigma_m=\sigma_{bmax}+\sigma_{bmin}/2 =(102-102)/2 =0$

And variable stress $\sigma_v=\sigma_{bmax}-\sigma_{bmin}/2$

$=(102-(-102))/2 =204/2 =102N/mm^2$

According to soderberg equation, we have

$$(K_f\sigma_v/\sigma^{''}_e)+(\sigma_m/\sigma_y) =1/f_s$$

WhereKf=1

$\sigma^{''}_e=\sigma_e x K_{sr}$

$=250x0.9$ (assume $K_{sr}=0.9$,ground surface)

$=225N/mm^2$

Sub the in the soderberg equation,

We get,

$(102/225)+(0/400)=(1/f_s)$

$1/f_s=(102/225)$

$F_s=225/102=2.2$

Factor of safety =2.2

5.**A stepped shaft of diameters D and d is subjected to a variable axial load P which cyclically varies between 0 and 10KN. The shaft is made of C20 steel, mirror polished with S_u=500N/mm^2 and S_y=260N/mm^2**

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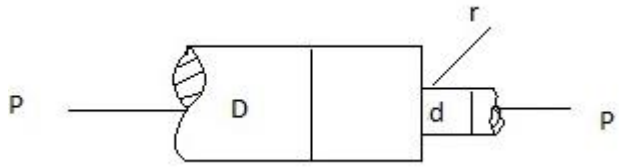
Determine the diameters D and d with D/d=1.5, FOS=2, notch sensitivity factor=0.8 and r/d=0.2 where r is the shoulder radius.

Solution:

Given



P_{max}=10KN=1

0000N P_{min}=0

σ_u=500N/mm^2

σ_y=260N/mm^2

Endurance limit for C20 steel is

σ_e =0.5 S_u=0.5x500=250N/mm^2 which is less then$\sigma_y$because σ_e=0.85σ_y.

Now considering mirror polished surface condition , and axial loading system, assume

K_a=load correction factor for axial loading =0.8

K_{sr}=surface finish factor =1

Now σ'_e=250xK_axK_{sr}

i.e.,σ'_e=250x0.8x1=200N/mm^2

In this problem, since the ultimate stress (σ_u) and the yield stress (σ_y) are given the diameters may be determined using Goodman equation or soderberg equation.

According to soderberg equation.

$K_f\sigma_v=\sigma'_e[1/f_s-\sigma_m/\sigma_y]$

Now

K_f=fatigue stress concentration factor

=1+q(K_f-1)

=1+0.8(1.45-1)=1.36

σ_v =variable stress amplitude

=P_{max}-P_{min}/2 =10000-0/2A =5000/A N/mm^2

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$=P_{max}+P_{min}/2A \quad =10000+0/2A \quad =5000/A \quad N/mm^2$

Where A=area of critical section=$\pi/4xd^2$

Now using soderberg equation

$$1.36x5000/(Ax200) \quad +(5000/(Ax260)) \quad =1/2$$

By solving, we get $A=106.5mm^2$

I.e., $\pi/4xd^2=106.5 \quad d=\{106.5x4/\pi\}^{0.5}$

=12mm (D=1.5d)

D=18mm

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UNIT II -SHAFT AND COUPLINGS

TECHNICAL TERMS

Tolerance

Tolerance is the amount of variation permitted to a basic size. [Or] Difference between maximum and minimum limits of size is called tolerance.

Preferred numbers

Preferred numbers are numbers, which are got by geometric progression with specific step ratios, they include integral power of 10.

They result in the optimized sizes of a product to cover a particular range

Critical speed.

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.

Key

A key is device, which is used for connecting two machine parts for preventing relative motion of rotation with respect to each other.

Woodruff keys

A woodruff key is used to transmit small value of torque in automotive and machine tool industries. The keyway in the shaft is milled in a curved shape whereas the key way in the hub is usually straight.

Couplings

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.

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Flexible couplings

They are used to join the abutting ends of shafts when they are not in exact alignment.

They are used to permit an axial misalignment of the shafts without under absorption of the power, which the shafts are transmitting

Splines

The shaft which is slotted to allow the movement of keys is called as splines.

2. INTRODUCTION

A shaft is a rotating machine element which is used to transmit power from one place to other place. Carbon steels of grade 40C8, 45C8, 50C4, 50C12 are normally used as shaft materials.

Material properties

It should have high strength

It should have good machinability.

It should have low notch sensitivity factor.

It should have good heat treatment properties.

It should have high wear resistance.

2.1.1 TYPES OF SHAFT

1. Transmission shaft: These shafts transmit power between the source and machines absorbing power. The counter shafts, line shafts, overhead shafts all shafts are transmission shafts.

2. Machine shafts: These shafts from an integral part of the machine itself.

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2.1.2 STRESSES IN SHAFT

Following stresses are induced in the shaft.

1. Shear stress due to transmission of torque
2. Bending stress due to forces acting upon machine elements like gears, pulleys etc. 3. Stresses due to combined torsional and bending loads.

2.2DESIGN OF SHAFTS

The shaft may be designed on the basis of 1. Strength 2. Rigidity and stiffness in designing shaft on the basis of strength the following cases may be consider

1Shafts subjected to twisting moment only

2. Shaft subjected to bending moment only

3. Shaft subjected to combined twisting moment and bending moment 4. Shaft subjected to fluctuating loads

2.2.1SHAFTS SUBJECTED TO TWISTING MOMENT ONLY

$$\frac{T}{J} = \frac{\tau}{r}$$

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

For hollow section

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

$k = \frac{d_i}{d_o}$ Where d_i=inside diameter, d_o= outside diameter

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Twisting moment may be obtained by using the following relation

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

For belt tension

$$T = (T_1 - T_2) R$$

T1- Tension in the tight side

T2- Tension in the slack side

R- Radius of the pulley

2.2.2SHAFT SUBJECTED TO BENDING MOMENT ONLY

The bending moment equation is

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

M- Bending moment

I-moment of inertia of cross sectional area of the shaft about the axis of rotation

σb- Bending stress

For Round Solid Shaft

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

For Hollow Shaft

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

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2.2.3 SHAFT SUBJECTED TO COMBINED TWISTING MOMENT AND BENDING MOMENT

When the shaft is subjected to combined twisting moment ad bending moment then the shaft must be designed on the basic of two moments simultaneously

For solid shaft

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

$$\frac{\pi}{32} \times \sigma_{b(max)} \times d^3 = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)$$

For hollow shaft

$$\frac{\pi}{16} \times \tau_{max} \times d_o^{\,3} \times (1 - k^4) = \sqrt{(M^2 + T^2)}$$

$$\frac{\pi}{32} \times \sigma_{b(max)} \times d_o^{\,3} \times (1 - k^4) = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)$$

2.3 KEY

A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them.

2.3.1 TYPES OF KEYS

1. Sunk key,
2. Saddle key,
3. Tangent key,
4. Round key
5. Splines

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2.3.2 SUNK KEYS

The sunk keys are provided half in the keyway of the shaft and half in the keyway of the hub or boss of the pulley.

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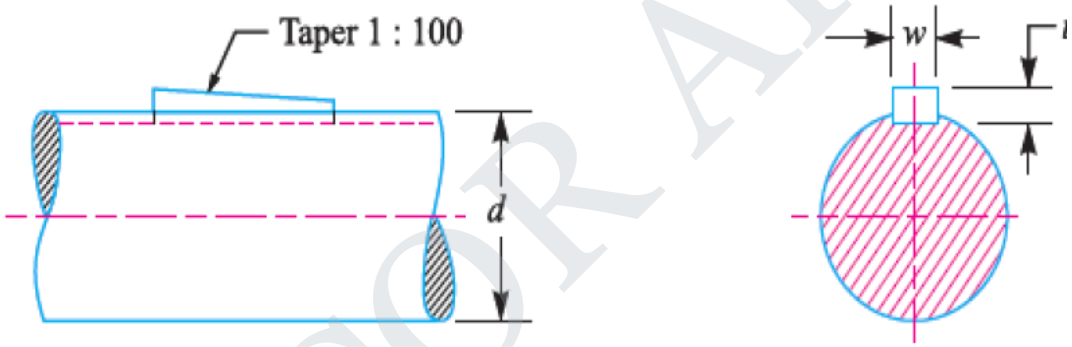

Fig2.1

1. Square sunk key

The only difference from the rectangular sunk key is the width and thickness is equal

$$W=t=D/2$$

2. Parallel sunk key

The parallel sunk key may be of rectangular or square cross section. The cross section is uniform in width and thickness throughout length.

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3. Gib head key

A gib head key is similar to a square or rectangular key but it has a head at one end, generally at the larger end of the taper sunk key. The gib head is used for driving the key while assembling or disassembling.

4. Feather key

Feather key is used where it is necessary to slide a keyed gear, pulley assembly along the shaft. Keys are tight fitted or screwed on the shaft.

5. **Woodruff key**

A woodruff key is used to transmit small amount of torque in automotive and machine tool industries. The keyway in the shaft is milled in a curved shape whereas the keyway in the hub is usually straight. The main advantage of this key is that it will align itself in the keyway.

2.3.4TYPES SADDLE KEYS

1. Flat saddle key

A flat saddle key is a taper key which fits in a keyway in the hub and is flat on the shaft.

2. Hollow saddle key

A hollow saddle key is a tapper key which fits in the keyway in the hub and the bottom of the key is shaped to fit the curved surface of the shaft.

Forces acting on a sunk key

1. Forces due to tight fit of the key and thus compressive stress is induced.
2. Force due to torque transmitted by the shaft and this force produced shearing and crushing stresses in the key.

2.4EFFECT OF KEYWAYS

The keyway 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 cross sectional

area of the shaft. Torsional strength of shaft is reduced. The following relation for the weakening effect of the keyway is based on the experiments results by H.F.Moore.

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$$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

e- Shaft strength factor. It is the ratio of strength of shaft with keyway to the strength of shaft without keyway.

W-width of the keyway

D-diameter of the shaft h-depth of keyway(thickness of key/2)

2.5 DESIGN OF COUPLING

Shaft couplings are used in machinery for several purposes

1. To provide for connection of shaft of units those are manufactured separately.
2. To provide for misalignment of the shaft or to introduce mechanical flexibility.
3. To reduce the transmission of shock loads from one shaft to another.
4. To introduce protection against over loads.

2.5.1 REQUIREMENT OF A GOOD SHAFT COUPLING

1. It should be easy to connect or disconnect.
2. It should transmit the full power from one shaft to the other shaft without losses.
3. It should hold the shaft in perfect alignment.
4. It should have no projecting parts.

2.5.2 TYPES OF SHAFT COUPLINGS

1. Rigid coupling

It 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

2. Flexible coupling

It is used to connect two shafts having lateral and angular misalignments. The types are

(a)Bushed pin type

coupling (b)Universal

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coupling (c)Oldham

coupling

2.5.3 SLEEVE (or) MUFF COUPLING

It is made of cast iron. It consists of a hollow cylinder whose inner diameter is that same as that of the shaft. It is fitted over the ends of two shafts by means of a gib head key. The power transmitted from one shaft two other shafts by means of a key and a sleeve.

Outer diameter of sleeve D=2d+13mm Length of sleeve

L=3.5d d- diameter of shaft

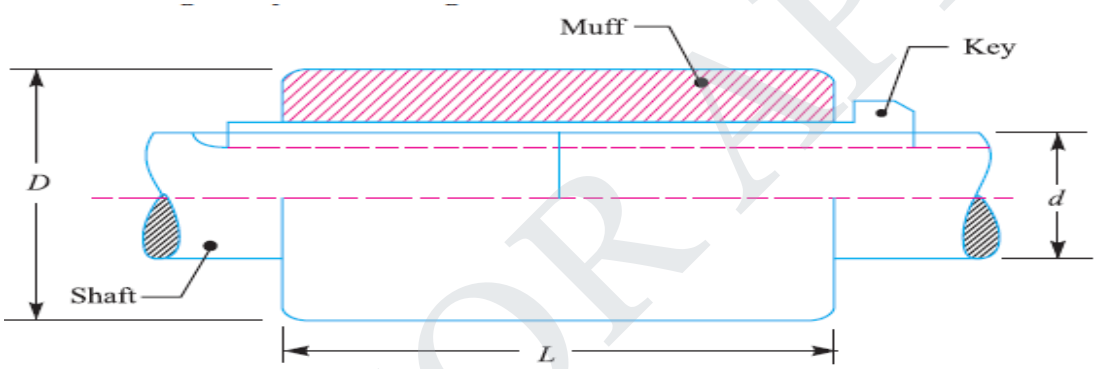

Figure 2.2 muff coupling

2.5.4 DESIGN OF MUFF COUPILNG

1. Design for sleeve

The sleeve is designed by considering it as a hollow shaft

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

2. Design for key

The length of coupling key is at least equal to the length of the sleeve. The coupling key Is usually made into two parts so that the length of key in each shaft

$$l = \frac{L}{2}$$

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Induced crushing stress and shear stress may be checked

$$T = l \times w \times \tau \times \frac{d}{2}$$

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

2.6 FLANGE COUPLING

A flange coupling usually applied to a coupling having two separate cast iron flanges. Each flange is mounted on the shaft and keyed to it. The faces are turned up at right angle to the axis of the shaft. One of the flanges has a projected portion and the other flange has a corresponding recess. This helps to bring the shaft into line and to maintain alignment. The two flanges are coupled together by means of bolt and nuts.

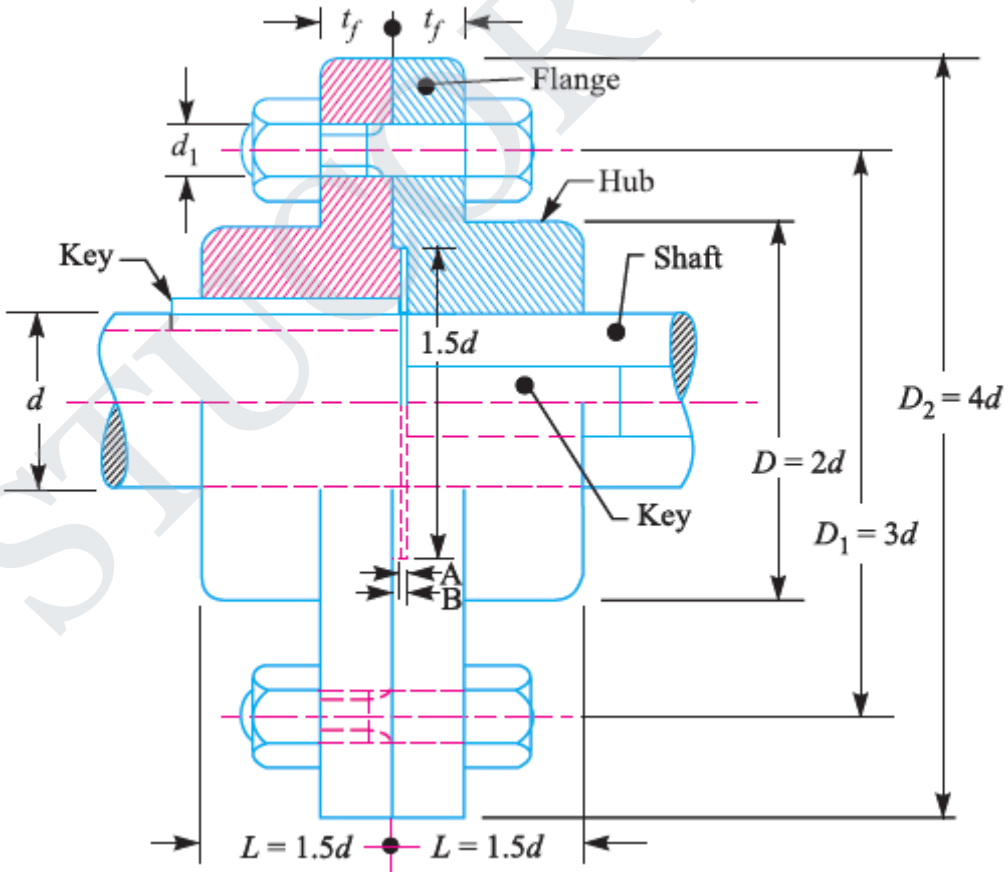

Figure 2.3 Flange coupling

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1. Design for hub the hub is designed by considering it as a hollow shaft

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

D=2*d

Length of hub

L=1.5d

2. Design for key

The key is designed with equal properties and then checked for shearing and crushing stress. The length of key is taken equal to the length of hub

Design for flange

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

t_f- thickness of flange(d/2)

3. Design for bolt

The bolts are subjected to shear stress due to torque transmitted.

The number of bolts (n) depends upon the diameter of shaft and pitch circle diameter is taken

D_1=3d

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$$T = \frac{\pi}{4} \times {d_1}^2 \times \tau_b \times n \times \frac{D_1}{2}$$

D_1- diameter of bolt

Crushing

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

2.7 KNUCKLE JOINT

A knuckle joint is used to connect two rods which are under the action of tensile loads. It consists of mainly three elements a fork or double eye rod, a single eye rod and knuckle pin. Its use may be found in the link of a cycle chain, tie rod joint for roof truss.

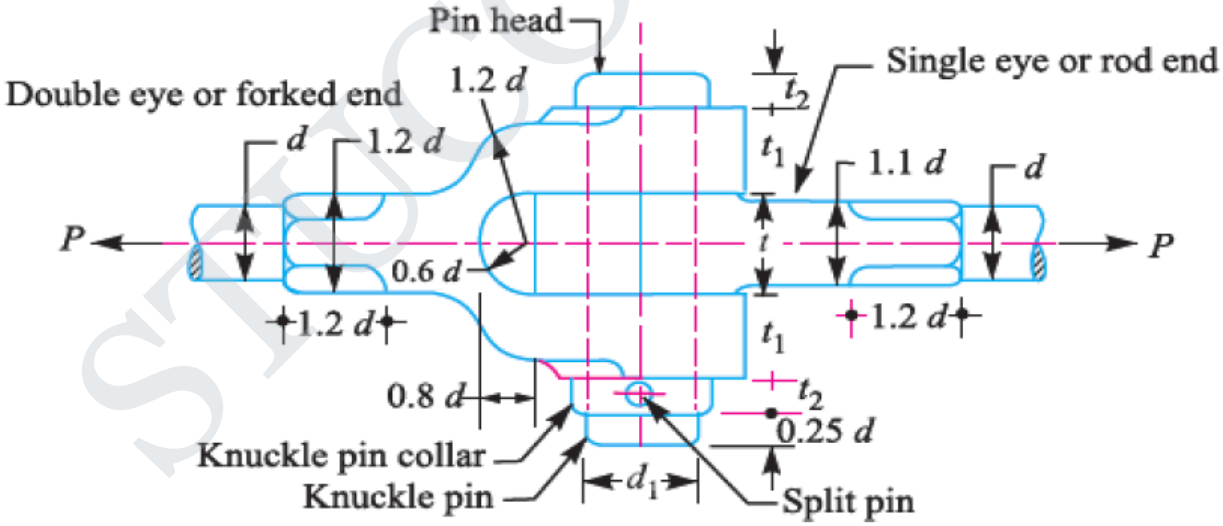

Figure 2.4 knuckle joint

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SHAFTS SUBJECTED TO TWISTIG MOMENT

1) Find the diameter of a solid steel to transmit 20kw at 200r.p.m. The ultimate shear stress for the steel may be taken as 360Mpa and a factor of safety as 8.

If a hollow shaft is to be used in place of the solid shaft, find the inside and outside diameter when the ratio of inside to outside diameters is 0.5.

Given:

P=20KW=20X10^3 W; N=200rpm; τ_u=360MPa=360N/mm^2; F.S.=8; k=d_i/do=0.5.

Solution:

We know that the allowable shear stress,

Diameter of the solid shaft

Let

We know that torque transmitted by the shaft,

$$T = \frac{PX60}{2\pi N} = \frac{20X10^3}{2\pi 200} = 955\text{N-m} = 955\text{X}10^3\text{N-mm}$$

We also know that torque transmitted by the solid shaft (T),

$$955\text{x}10^3 = \frac{\pi}{16}\text{x}\ \tau\ \text{x}\ d^3 = \frac{\pi}{16}\text{x}\ 45\ \text{x}\ d^3 = 8.84\text{b}\ d^3$$

d^3 = 955x10^3/8.84 = 108032 or d = 47.6 say **50 mm**

Diameter of the hollow shaft

Let d_i=Inside diameter, and

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d_o=Outside diameter.

We know that the torque transmitted by the hollow shaft (T),

$$955x10^3 = \frac{\pi}{16} \times \tau (d_o)^3 \times (1-k^4)$$

$$= \frac{\pi}{16} \times 45 (d_o)^3 [1-(0.5)^4] = 8.3 (d_o)^3$$

$$(d_o)^3 = 955x10^3/8.3 = 115060 \text{ or } d_o = 48.6 \text{ say } 50 \text{ mm}$$

And $d_i = 0.5xd_o = 0.5x50 =$ **25 mm**

Result:

Diameter of the solid shaft **= 50 mm** Diameter of the hollow shaft **= 25 mm**

SHAFT SUBJECTED TO BENDING MOMENT ONLY:

2) A pair of wheels of a railway wagon carries a load of 50 kN on each axle box acting at a distance of 100 mm outside the wheels base. The gauge of the rails is 1.4 m. Find the diameter of the axle between the wheels, if the stress is not to exceed 100 Mpa

Given:

W = 50KN $50X10^3$ N; N L= 100 mm ; $x = 1.4$ m; σ_b = 100 Mpa = 100 N/mm^2.

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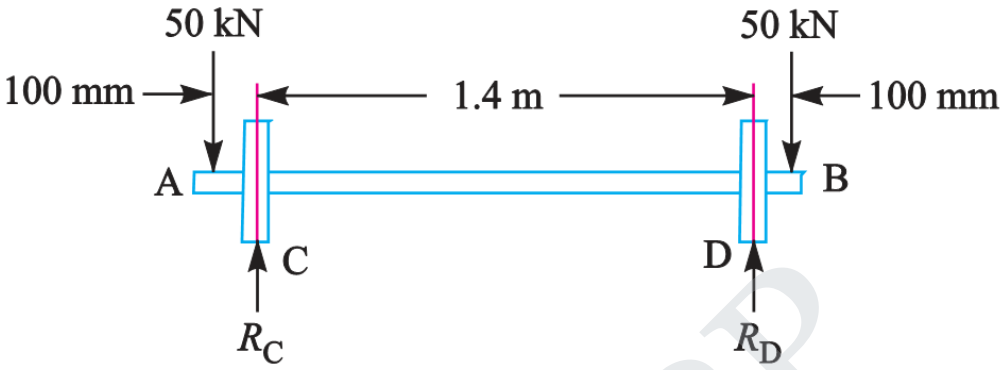

Solution:

A little consideration will show that the maximum bending moment acts on the wheels at C and D. Therefore maximum bending moment,

$$M = W.L = 50x10^3 \times 100 = 5 \times 10^6 \text{ N-mm}$$

Let d = Diameter of the axle.

We know that the maximum bending moment (M),

$$5x10^6 = \frac{\pi}{32} \times \sigma_b \times d^3 = \frac{\pi}{32} \times 100 \times d^3 = 9.82\, d^3$$

$$D^3 = 5x10^6 / 9.82 = 0.51x10^6 \text{ or } d = 79.8 \text{ say } 80\text{mm}$$

Result:

Diameter of the axle = 80 mm

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SHAFTS SUBJECTED TO COMBINED TWISTING MOMENT AND BENDING MOMENT:

3) A solid circular shaft subjected to a bending moment of 3000 N-m and torque of 10000 N-m. The shaft is made of 45 C 8 steel having ultimate tensile stress of 700 MPa and a ultimate shear stress of 500n MPa. Assuming a factor of safety as 6, determine the diameter of the shaft.

Given:

M = 3000 N-m = $3x10^6$ N-mm ; T = 10000 N-m = $10x10^6$ N-mm ; σ_{tu} = 700 MPa = 700 N/mm^2 ; τ_u = 500 MPa = 500 N/mm^2

Solution:

We know that the allowable tensile stress,

$$\sigma_t \text{ or } \sigma_b = \frac{\sigma tu}{F.S.} = \frac{700}{6} = 116.7 \text{ N/mm}^2$$

and allowable shear stress,

$$\tau = \frac{\tau u}{F.S.} = \frac{500}{6} = 83.3 \text{ N/mm}^2$$

Let d = Diameter of the shaft in mm.

According to maximum shear stress theory, equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(3X10^6) + (10X10^6)} = 10.44 \text{ X } 10^6 \text{ N-mm}$$

We also know that equivalent twisting moment (T_e),

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$$10.44 \times 10^6 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 83.3 \times d^3 = 16.36\, d^3$$

$$d^3 = 10.44 \times 10^6 / 16.36 = 0.636 \times 10^6 \text{ or } d = 86 \text{ mm}$$

According to maximum normal stress theory, equivalent bending moment,

$$M_e = \frac{1}{2}\left(M + \sqrt{(M^2 + T^2)}\right) = \frac{1}{2} \times (M^2 + T_e)$$

$$= \frac{1}{2}(3 \times 10^6 + 10.44 \times 10^6) = 6.72 \times 10^6 \text{ N-mm}$$

We also know that the equivalent bending moment (M_e),

$$6.74 \times 10^6 = \frac{\pi}{32} \times \sigma_b \times d^3 = \frac{\pi}{32} \times 116.7 \times d^3 = 11.46\, d^3$$

$$d^3 = 6.72 \times 10^6 / 11.46 = 0.586 \times 10^6 \text{ or } d = 83.7 \text{ mm}$$

Taking the larger of two values, we have

d = 86 say **90 mm**

Result: Diameter of shaft = 90 mm

4) A shaft made of mild steel is required to transmit 100 KW at 300 rpm. The supported length of the shaft is 3 meters. It carries two pulleys each weighing 1500 N supported at a distance of 1 meter from the ends respectively. Assuming the safe value of stress, determine the diameter of the shaft.

Given:

P = 100 KW = 100 x 10^3 W; N = 300 rpm; L = 3 m; W = 1500 N;

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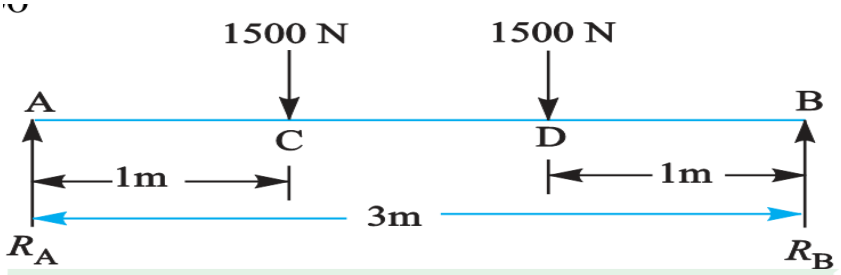

Solution:

We know that the torque transmitting by the shaft,

$$T = \frac{P\,X\,60}{2\pi N} = \frac{100X10^3}{2\pi\,X\,300} = 3183 \text{ N-m}$$

The shaft carrying the two pulleys is like a simply supported beam as shown in fig.

The reaction at each support will be 1500 N,

$R_A = RB = 1500$ N

A little consideration will show that the maximum bending moment lies at each pulley at C and D

Maximum bending moment,

M = 1500 x 1 = 1500 N-m

Let

d = Diameter of the shaft in mm.

We know that equivalent twisting moment,

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$$T_e = \sqrt{(M^2 + T^2)} = \sqrt{(1500^2)} + (3183^2)\ 35190 \text{ N-m}$$

We also know that equivalent twisting moment (Te)

$$3519 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 60 \times d^3 = 11.8\ d^3$$

$$d^3 = 3519 \times 10^3 / 11.8 = 298 \times 10^3 \text{ or } \mathbf{d = 66.8 \text{ say } 70 \text{ mm}}$$

SOLVED PROBLEMS FOR FLANGE COUPLING

Problems

1. **Design a cast iron protective type flange coupling to transmit 10Kw at 960 rpm Select suitable material**

Given data

Power, P=10kw

Speed, N=960rpm

To find: Design a protective type flange coupling

Solution:

(i) Design of a shaft

Power, $p = 2\pi NT/60$ $T = px60/2\pi N = 10x10^3x60/2x\pi x960 = 99.4718$N-m

Torque transmitted by the shaft,

$T = \pi/16 \times \tau_s \times d^3$

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Assume that the shaft, key and bolts are made of mild steel which is having

Allowable shear strength=50N/mm^2

Allowable crushing strength=90N/mm^2

99471.8= π/16x 50x d^3

d=21.638mm

(ii) Dimensions of the flange couplings

(a) Outside Dai of hub, D=2xd+2x25=50mm

(b) Length of hub, L=1.5d=1.5x25=37.5mm

(c) Pitch circle diameter of bolts, D_1=3d=3x25=75mm

(d) Thickness of flange, t_f =0.5d =0.5x25=12.5mm

(e) Outer diameter of flange, D_2=4d =4x25=100mm

(f) Thickness of the protective circumferential flange, tp

tp=0.25d=0.25x25=6.25mm

(iii) Design of hub

T= π/16x τ_h x (D^4-d^4/D)

99471.8= π/16x τ_h x (50^4-25^4/50)

τ_h=4.325 N/mm^2

τ_h=5 N/mm^2

Induced shear stress of the hub is less than the permissible stresses.

Therefore, the design of hub is safe

(iv) Design of key:

Length of the key, l=l=37.5mm

Take width of the key, b=9mm

Thickness of the key,h=7.5mm

(a) Check for shearing:

T= lxbx τ_k x d/2

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$99471.8=37.5x9x\ \tau_k\ x25/2$

$\tau_k=23.57\ N/mm^2$

Induced shear stress of the key is less than the permissible stresses

Therefore, the design of key is safe.

(a) Check for crushing:

$T= l\ x\ h/2x\ \sigma_{ck}\ x\ d/2$

$99471.8=37.5x7.5/2x\ \sigma_{ck}\ x25/2$

$\sigma_{ck}\ =57\ N/mm^2$

Induced crushing stress of the key is less than the permissible stresses therefore; the design of key is safe.

(v) Design of flange:

$99471.8= \pi D^2/2x\ \tau_h x t_f$

$\tau_h=2.06\ N/mm^2$

Induced crushing stress of the flange is less than the permissible stresses therefore; the design of flange is safe.

(vi) Design for bolts

(a) Check for shearing:

$T=\pi/4xd_b^2x\tau_b xnxD_1/2$

$99471.8= \pi/4xd_b^2x50x3x75/2$ (n=3 for D up to50mm)

$d_b=4.75mm$

Say, bolt dia 6mm

(a) Check for crushing:

$T= n\ x\ d_b\ x\ \sigma_{cb}\ x\ D1/2$

$99471.8=3\ x\ 6x\ \sigma_{cb}\ x75/2$

$\sigma_{cb}=11.789\ N/mm^2$

Induced crushing stress of the bold is less than the permissible stresses therefore; the design of bolt is safe.

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Design of muff coupling

1. Design of muff or coupling for a shaft to transmit 35kw at 350rpm. The safe shear stress for the steel shaft is 50N/mm^{2}and for a cast iron muff it is 15N/mm^2. The allowable shear stress and crushing stress for the material are 42N/mm^2 and 120N/mm^2 respectively

Given:

Power to be transmitted, p=35x10^{3}W

Speed, n=350rpm

Allowable shear stress for shaft τ_s=50N/mm^2

Allowable shear stress for muff τ_m=15N/mm^2

Allowable shear stress for shaft τ_k=42N/mm^2

Allowable shear stress for shaft σ_c=120N/mm^2

To find

Design a sleeve or muff coupling

(i) Design of a shaft

Power, p =2πNT/ 60 T=px60/2πN = 35x10^{3}x60/2xπx350=954.92N-m

Torque transmitted by the shaft,

T=π/16x τ_s x d^3

Allowable shear strength=50N/mm^2

954.92x10^3= π/16x 50x d^3

d=45.98 say d=46mm

(2) Design of sleeve

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The usual proportions are

D= 2d +13=2x46+13=105mm

Length of the sleeve

L=3.5 d=3.5 x46=161mm

Check for induced shear stress in muff

$T= \pi/16x\ \tau_m\ x\ (D^4-d^4/D)$

$954.92x10^3= \pi/16x\ \tau_m x\ (105^4-46^4/105)$

$\tau_m=4.36\ N/mm^2$

Induced shear stress of the muff is less than the permissible stresses

Therefore, the design of muff is safe

(3) Design for key

From **PSG** data book 5.16 for the shaft dia d=46mm

Width of the key, b=14mm

Height of the key, h=9mm

Length of the key, l=L/2=161/2=80.5mm

(a) Check for shearing:

$T= lxbx\ \tau_k\ x\ d/2$

$954.92x10^3=80.5\ x14x\ \tau_k\ x46/2$

$\tau_k=36.83\ N/mm^2$

Induced shear stress of the key is less than the permissible stresses

Therefore, the design of key is safe.

(a) Check for crushing:

$T= l\ x\ h/2x\ \sigma_{ck}\ x\ d/2$

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$954.92x10^3=80.5x9/2x\ \sigma_{ck}\ x46/2$

σ_{ck} =114.6N/mm^2

Induced crushing stress of the key is less than the permissible stresses therefore; the design of key is safe.

2.7.1 Dimension of various parts of the knuckle joint

d- Diameter of rod

d1- diameter of pin outer dia of eye

d2=2d diameter of knuckle pin head and collar

d3=1.5d

thickness of single eye or rod end t=1.25d

thickness of fork t1=0.75d

thickness of pin head t2=0.5d

KNUCKLE JOINT PROBLEMS

1.A knuckle joint is to transmit a force of 140KN . Allowable stresses in tension,shear and compression are 75N/sq.mm,65N/sq.mm respectively.design the joint.

Given data:

Force P=140KN=140*$^{10^3}$ N

Allowable tensil stress , $\sigma_t = 75N/mm^2$

Allowable shear stress , τ=65 N/mm^2

Allowable compressive stress , =140 N/mm^2

Solution:

Failure of solid rod in tension is given by

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$$p = \frac{\pi}{4} d^2 \sigma_t$$

$$140*10^3 \text{ N} = \frac{\pi}{4} d^2 * 75$$

Diameter of rod,d =48.75mm =50mm

Various standard proportion pf the knuckle joint are

Diameter of pin , $d_1 = d = 50mm$

Outer diameter of eye, $d_2 = 2d = 2 * 50 = 100mm$

Diameter of pin head, $d_3 = 1.5d = 1.5 * 50 = 75mm$

Thickness of eye,t=1.25d=1.25*50=62.5mm

Thickness of fork, t_1=0.75d=0.75*50=37.5mm

Thickness of fork' t_1=0.75d=0.75*50=37.5mm

Thickness of pin head, t_2=0.5d=0.5*50=25mm

Check for τ

1.failure of knuckle joint by double shear

$$p = 2 * \frac{\pi}{4} d^2 \tau$$

$$140*10^3 = 2 * \frac{\pi}{4} * 50^2 * \tau$$

T=35.65 N/mm^2 which is less than the permissible shear stress (65 N/mm^2),

Hence,the design is satisfactory.

2.failure of single eye or rod end in double shear

P=(d_2-d_1)t* τ

$$140*10^3 = (100 - 50) * 62.5 * \tau$$

τ =44.8 N/mm^2 which is less than the permissible shear stress.

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Hence,the design is satisfactory.

3.failure of forced ends in double shear

P=(d_2-d_1)t* τ*2

$140*10^3 = (100 - 50) * 62.5 * \tau * 2$,

$\tau = 22.4\ N/mm^2$ which is less than the permissible shear stress.

Hence,the design is satisfactory.

Check for σ_t

4.failure of the single eye or rod end intension

P=(d_2-d_1)tσ_t

$140*10^3 = (100 - 50) * 62.5 * \sigma_t$

$\sigma_t = 44.8\ N/mm^2$ which is less than the permissible shear stress(75 N/mm^2).

Hence,the design is satisfactory.

5.failure of forced end in tension

P=(d_2-d_1)t$* 2 * \sigma_t$

$140*10^3 = (100 - 50) * 37.5 * 2 * \sigma_t$

$\sigma_t = 37.3365 \frac{N}{mm^2}$

which is less than the permissible shear stress(75 N/mm^2).

Hence,the design is satisfactory.

Check for σ_c:

6.failure of the single eye or rod end in crushing

P=t$* d_1 * \sigma_c$

$140*10^3 = 50 * 62.5 * \sigma_c$

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σ_c=44.8 N/mm^2 which is less than the permissible shear stress(75 N/mm^2).

Hence,the design is satisfactory.

7.failure of forked end in crushing

P=$t_1 d_{1*} 2 * \sigma_c$

$140*10^3 = 50 * 37.5 * 2 * \sigma_c$

σ_c=37.33 N/mm^2 which is less than the permissible shear stress(75 N/mm^2).

Hence,the design is satisfactory.

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UNIT III -TEMPORARY AND PERMANENT JOINTS

TECHNICAL TERMS

Screw- It secures and transmit power

Pitch- It is axial distance at a point on one thread to corresponding point on next thread

Lead- Distance the screw moves in one turn

Thread forms-ACME,Square,ISO,BSW,BA

Differential screw- It gives motion equal to difference of two pitches

Compound screw- It gives motion equal to sum of two pitches

Welding- Process of joining two similar or dissimilar metal with or without application of pressure

Butt joint- Joint made by welding edges of two plates

Lap joint - Joint made by welding overlap of two plates

Tee joint- Joint made by welding at right angles of two plates

Corner joint- Joint made by welding at right angles of two plates such that it forms an angle

Edge joint- Plates less than 6mm thickness overlapping plates can be directly welded

Stress concentration factor- Due to changes in cross section it occurs at toe and heel points

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3. INTRODUCTION

Welding is the most commonly used process for permanently joining machine parts. Welding is done by fusing the metallic parts with heat at their junction with or without pressure. Advantages of welding over riveting

1. Possibility of joining curvilinear parts
2. Cost is less
3. Tightness of joint
4. Noiseless process
5. Greater strength

3.1 DESIGN OF WELDED JOINTS

In order to determine the strength of fillet joint, it is assumed that the section of fillet is a right angle triangle ABC with hypotenuse AC making equal angles with other two sides AB and BC. The length of each side is known as leg or size of the weld and the perpendicular distance of hypo tenuous from intersection of legs (BD) is known as the throat thickness.

In the triangle BDA

$$\frac{BD}{AB} = \sin\theta$$

$$BD = AB\sin\theta$$

$$= s \sin 45 = 0.707s$$

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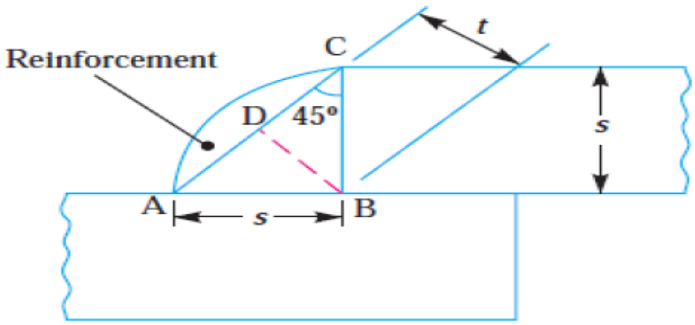
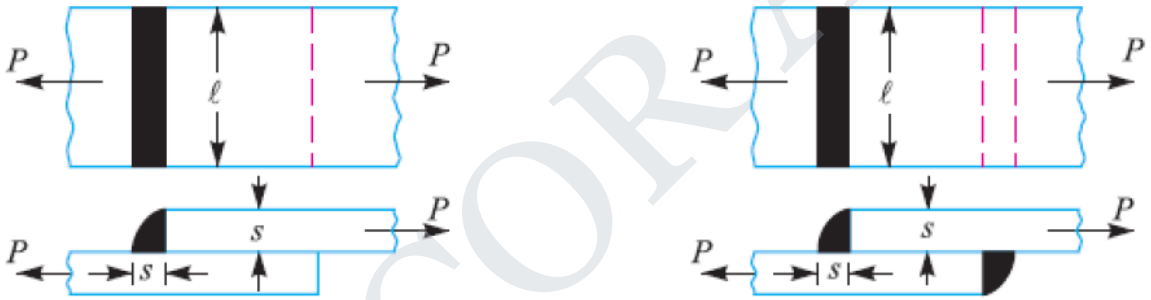


3.2 STRENGTH OF TRANSVERSE FILLET WELDED JOINT

The fillet or lap joint is obtained by overlapping the plates and then welding the edges of the plates. The transverse fillet welds are designed for tensile strength.



(*a*) Single transverse fillet weld. (*b*) Double transverse fillet weld.

Assumed that the weld section is right angle triangle ABC with hypotenuse AC making equal with other two sides AB and BC

t- Throat thickness

s- Thickness of plate

L-length of weld throat thickness

t=s*sin45=0.707s

Minimum area of the weld **A=t*l=0.707s*l**

Allowable tensile stress for the weld then tensile strength of joint for single fillet weld

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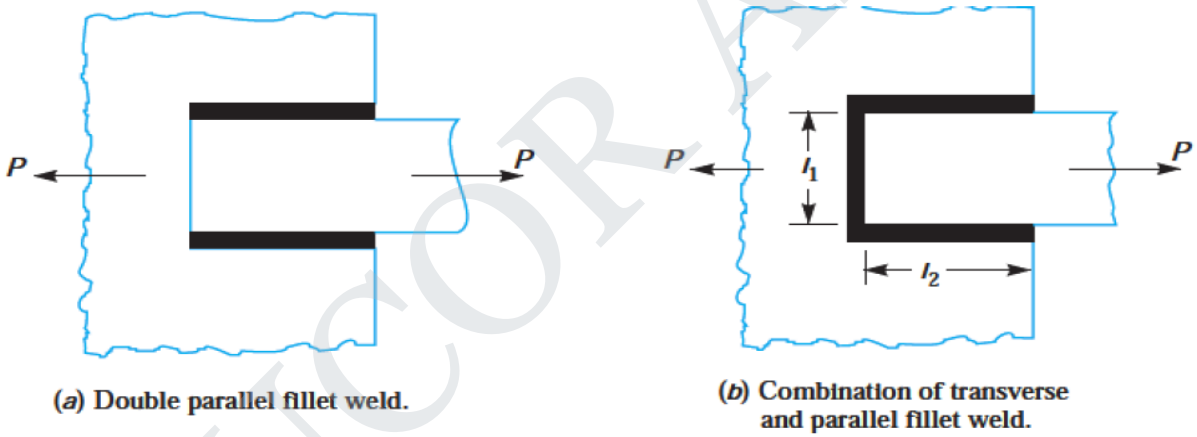
$$P = 0.707s \times l \times \sigma_t$$

Tensile strength of joint for double fillet weld

$$P = 2 \times 0.707s \times l \times \sigma_t$$

3.3 STRENGTH OF PARALLEL FILLET WELD

The parallel filet weld joints are designed for shear strength. Consider a double parallel fillet weld as shown in figure



(*a*) Double parallel fillet weld.

(*b*) Combination of transverse and parallel fillet weld.

Fig 3.1

Shear strength of joint due to double parallel fillet weld

$$P = 2 \times 0.707s \times l \times \tau$$

$$= 1.414 \times s \times l \times \tau$$

Combination of single transverse and double parallel fillet weld then the strength of joint

$$P = 0.707s \times l_1 \times \sigma_t + 1.414 \times s \times l_2 \times \tau$$

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3.4 STRESS CONCENTRATION FACTOR FOR WELDED JOINTS

The reinforcement provided to the weld produces stress concentration at the junction of the weld and parent metal. When the parts are subjected to fatigue loading the stress concentration factor as given in the following table is taken into account

Type of joint	Stress concentration factor
Reinforced butt welding	1.2
Toe of transverse fillet weld	1.5
End of parallel fillet weld	2.7
T-butt joint with sharp corner	2.0

3.5 AXIALLY LOADED UNSYMMETRICAL WELD SECTIONS

Unsymmetrical sections such as angles, channels T-sections etc, welded on the flange edges are loaded axially as shown in figure. In such case the length of weld should be proportioned in such a way that the sum of resisting moments of the welds about the gravity axis is zero.

l_a- length of weld at the top

l_b- length of weld at the bottom

l- Total length of weld a- distance of top weld from gravity axis

b- Distance of bottom weld from gravity axis

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$$l_a = \frac{l \times b}{a+b} \, , l_b = \frac{l \times a}{a+b}$$

$$l = l_a + l_b$$

3.6 ECCENTRICALLY LOADED WELD JOINTS

When a welded joint is eccentrically loaded the principle stress will be applied because the welded part undergoing direct load and a bending moment Maximum normal stress

3.6.1 Maximum normal stress

$$\sigma_{t(\max)} = \frac{\sigma_b}{2} + \frac{1}{2}\left[\sqrt{(\sigma_b)^2 + 4\tau^2}\right]$$

3.6.2 Maximum shear stress

$$\tau_{\max} = \frac{1}{2}\left[\sqrt{(\sigma_b)^2 + 4\tau^2}\right]$$

Case (i)

Consider a t-T-joint fixed at one end and subjected to an eccentric load P at a distance e, the joint subjected to two types of stresses

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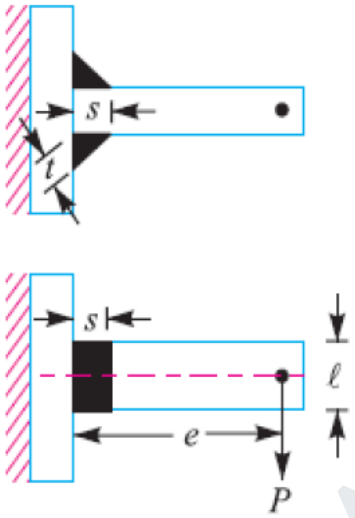

Figure 3.4 eccentrically loaded weld joint

Direct shear due to the shear force (P)

Bending stress due to bending moment **P*e** Aera of throat (A) =throat thickness*length of weld

=t*l*2 =1.414*s*l

3.6.3 Shear stress in the weld

$$\tau = \frac{P}{A}$$

3.6.4 Section modulus of the weld metal through the throat

$$z = \frac{s \times l^2}{4.242}$$

3.6.5 Bending stress

$$= \frac{M}{z} = \frac{P \times e}{\frac{s \times l^2}{4.242}}$$

Case (ii)

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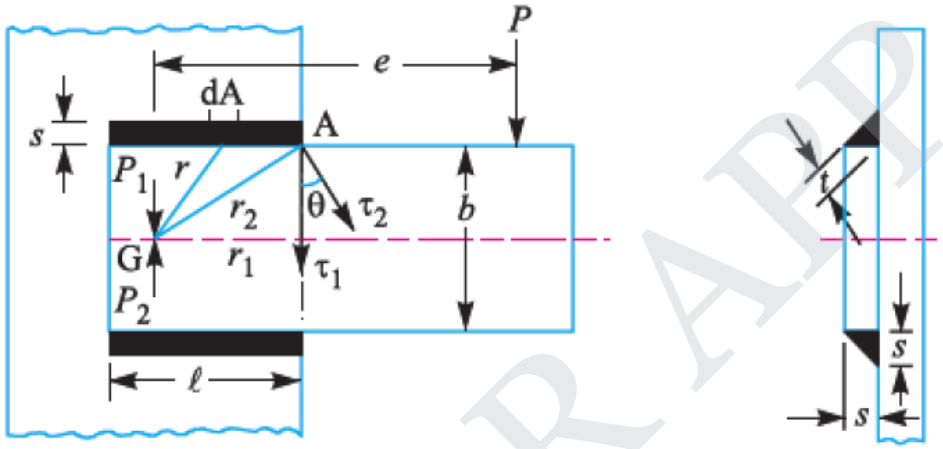
When a welded joint is loaded eccentrically as shown in figure subjected to two types of stresses

1. Direct or primary shear stress
2. Shear stress due to turning moment



P- Eccentric load

e- Eccentricity (i.e.) perpendicular distance between the line of action of load and center of gravity(G) of throat section

l- Length of single weld

t- Throat thickness

Direct or primary shear stress

$$\tau_1 = \frac{P}{1.414 \times s \times l}$$

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3.6.6 Shear stress due to turning moment (i.e.) secondary shear stress

$$\tau_1 = \frac{P \times e \times r_2}{J}$$

The polar moment of inertia of the throat area (A) about the center of gravity (G) is obtained by the parallel axis theorem x- perpendicular distance between two parallel axis

$$J = 2A\left(\frac{l_2}{12} + x^2\right)$$

$$A = 0.707 \times s \times l$$

3.6.7 Resultant shear stress at A

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

3.6.8 θ-angle between τ1 and τ2

$$\cos\theta = \frac{r_1}{r_2}$$

3.7 SCREWED FASTENER

A screw thread is obtained by cutting a continuous helical groove on a cylindrical surface (external thread). The threaded portion engages with corresponding threaded hole (internal thread) forming screwed fastener. Screwed fasteners such as bolts, studs and nuts in combination, machine screws are used for fastening components that require frequent assembly and disassembly. Screwed fasteners are to hold parts together and to transmit power.

3.7.1 SCREW THREAD NOMENCLATURE

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1. Major diameter this is the largest diameter of a screw thread, touching the crests on external thread or the roots on internal thread. This is also called nominal diameter.

2. Minor diameter this is the smallest diameter of a screw thread, touching the roots or core of external thread or the crests of internal thread. This is also called core or root diameter.

3. Pitch diameter This is the diameter of an imaginary cylinder, passing through the threads at the points where the width of thread is equal to the space between threads

4. Pitch It is the distance measured parallel to the axis, between corresponding points on adjacent screw threads.

5. Lead It is the distance, a screw advances axially in one turn. For a single threaded screw, the pitch and lead are equal. For a double threaded screw the lead is twice the pitch and so on.

3.7.2FORMS OF THREADS

Bureau of Indian Standards (BIS) adopts ISO (International Organization for Standards) metric threads which are followed by number of countries.

1. V-Thread: This thread profile has a larger contact area providing more frictional resistance to motion. It is used where effective positioning is required.

2. British Standard Whitworth (B.S.W) thread: This thread is adopted in Britain in inch units. The profile has rounded ends making it less liable to damage than sharp V-threads.

3. Square thread: This is an ideal thread form for power transmission. In this as the threaded flank is at right angle to the axis. The normal force between the threads acts parallel to the axis with zero radial components. This enables the nut to transmit very high pressure as in case of a screw jack and other similar applications.

4. Buttress thread: This thread form is combination of V-thread and square thread. It exhibits the advantages of square threads like the ability to transmit power and low frictional resistance and the strength of a V-thread. It is used where power transmission takes place in one direction only.

5. ACME thread: It is a modification form of square thread. It is much stronger than square thread because of the wider base and it is easy to cut. The inclined sides of thread facilitate quick and easy engagement and disengagement as for example the split nut with the lead screws of lathe.

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3.7.3THREAD SERIES

BIS recommends two thread series, coarse and fine series. Based on the relative values of pitches. It must be noted that the concept of quality is not associated with these terms.

3.7.4 THREAD DESIGNATION

The diameter pitch combination of an ISO metric screw thread is designated by the letter M followed by the value of nominal diameter and pitch, the two values being separated by the sign ‘x’. For example a diameter pitch combination of nominal diameter 10mm and pitch 1.25mm is designated as M10x1.25.

3.7.5 STRESSES IN SCREW THREADS

The following types of stresses are induced in screwed fasteners under static loading

(i) Stresses due to initial tightening of the nut

(ii) Stresses due to external forces

(iii) Stresses due to the combination of above

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3.8 SOLVED PROBLEMS

WELDED JOINT

1. A plate 100 m Wide and 10 mm thick is to be welded to another plate by means of double parallel fillets. The plates are subjected to a static load of 80 k N. Find the length of weld if the permissible shear stress in the weld does not exceed 55 MPa.

Given :

Width=100 mm ;

Thickness = 10 mm

P = 80 kN = 80 × 10^3 N ;

L= 55 MPa = 55 N/mm^2

SOLUTION:

Let L =Length of weld, and

s = Size of weld = Plate thickness = 10 mm

We know that maximum load which the plates can carry for double parallel fillet weld (P),

80 × 10^3 = 1.414 × s × l ×L

= 1.414 × 10 × l × 55 = 778 l

= 80 × 10^3 / 778 = 103 mm

Adding 12.5 mm for starting and stopping of weld run, we have

L = 103 + 12.5 = 115.5 mm Ans.

Problem-2

A plate 90mm wide and 15mm thickness is welded on to another plate by a single transverse weld and a double parallel fillet weld. Find the length of the parallel fillet weld if the plate is loaded by static tensile load . take the allowable tensile stress of the plate as 70 N/mm^2 and weld shear stress as 55 N/mm^2

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Plate width b=90mm

Plate thickness h=50mm

Allowable tensile stress for the plate σt =70 N/mm^2

Allowable shear stress for the weld τ =55 N/mm^2、

SOLUTION

Maximum pull that can be applied to the plate

= area of the cross section of the plate * Allowable tensile stress = b*h* σt

=90*15*70

=94500 N

Strength of the weld = (2*l*t+b*t)*τ (t=$h/\sqrt{2}$)

=(1.414*l*h+0.707b*h)τ

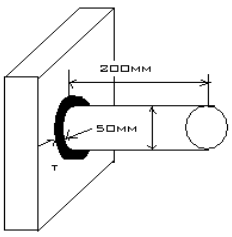
=94500

L= 36.00 mm

Total length of the weld = 90+(2*36.00)=162.00mm

3) **A 50mm diameter solid shaft is welded at its one edge to the vertical side of rectangular pillar by an all round fillet weld. A load of 10kn applied at the free end of the shaft which at a distance of 200mm from the fixed end. Find the size of the weld, assuming the permissible stress or the weld material in tension as 94mpa.**

Solution:



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Applied load, P =10KN =10000N
Diameter of shaft, d =50mm
Eccentricity for the load, e =200mm
Permissible tensile stress, (σ_t) = 94 Mpa
= 94×10^6 N/m^2 = 94 N/mm^2

To find the size of weld:

Let t =Throat thickness of weld
h = Size of weld

Since the load is applied eccentrically with respect to the welding location, two types of stresses will be induced in the weld such as

1. Direct shear stress due to the load, P
2. Bending stress due to bending moment, P x e

Now,

Direct shear stress, $\tau = \frac{Load}{Total\ throat\ area} = \frac{P}{\Pi dt}$

$= \frac{100000}{\Pi \times 50 \times t} = \frac{63.7}{t}$ N/mm^2

Bending stress, $\sigma_b = \frac{Bending\ moment}{Section\ modulus} = \frac{M}{Z}$

$= \frac{\frac{P.e}{\Pi d^2 t}}{4} = \frac{4Pe}{\Pi d^2 t} = \frac{4X10000X200}{\Pi X50^2 Xt} = \frac{1018.6}{t}$ N/mm^2

Applying maximum principal stress theory, the maximum normal stress can be calculated as,

$\sigma_{(max)} = \frac{1}{2}\left[\sigma_b + \sqrt{\sigma_b{}^2 + 4\tau^2}\right]$

$= \frac{1022.6}{t}$ N/mm^2

Equating this maximum normal stress to permissible stress, we get $\frac{1022.6}{t}$ N/ mm^2

$t = \frac{1022.6}{94} = 10.9$mm

$h = \sqrt{2}$ X t = $\sqrt{2}$ X 10.9 = 15.4mm

=16mm

Size of the weld = 16mm.

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Problem-4

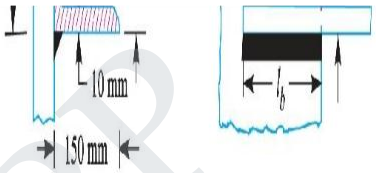
A 200 × 150 × 10 mm angle is to be welded to a steel plate by fillet welds as shown in Fig. If the angle is subjected to a static load of 200 kN, find the length of weld at the top and bottom. The allowable shear stress for static loading may be taken as 75 MPa.

Given :

a + b = 200 mm ;

P = 200 kN = 200 × 103 N ;

* = 75 MPa = 75 N/mm2



SOLUTION

l_a = Length of weld at the top,

l_b = Length of weld at the bottom, and

L = Total length of the weld = la + lb

Since the thickness of the angle is 10 mm, therefore size of weld, s = 10 mm

We know that for a single parallel fillet weld, the maximum load (P),

200 × 103 = 0.707 s × l × *

L = 0.707 × 10 × l × 75 = 530.25 l

∴ L = 200 × 103 / 530.25 = 377 mm..

Result :The length of the weld is 377mm.

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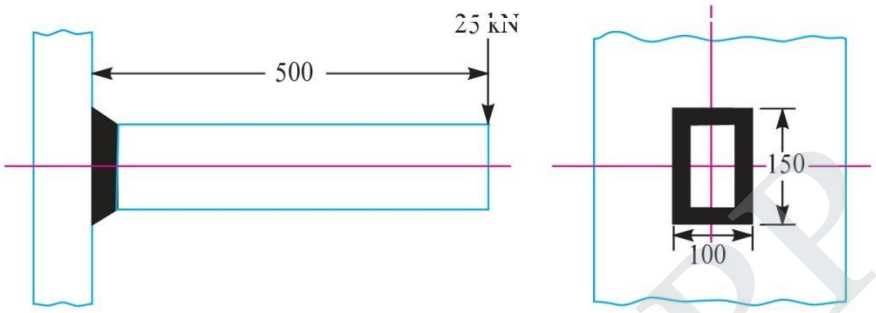
A rectangular cross-section bar is welded to a support by means of fillet welds as shown in Fig.

Determine the size of the welds, if the permissible shear stress in the weld is limited to 75 MPa.



All dimensions in mm

Solution. Given : $P = 25$ kN $= 25 \times 10^3$ N ; $\tau_{max} = 75$ MPa $= 75$ N/mm^2 ; $l = 100$ mm ; $b = 150$ mm ; $e = 500$ mm

Let s = Size of the weld, and

t = Throat thickness.

The joint, as shown in Fig. 10.26, is subjected to direct shear stress and the bending stress. We know that the throat area for a rectangular fillet weld,

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

$$= 0.707s\,(2 \times 150 + 2 \times 100) = 353.5\,s \text{ mm}^2 \quad \ldots (\because\ t = 0.707s)$$

$$\text{Direct shear stress, } \tau = \frac{P}{A} = \frac{25 \times 10^3}{353.5\,s} = \frac{70.72}{s} \text{ N/mm}^2$$

We know that bending moment,

$$M = P \times e = 25 \times 10^3 \times 500 = 12.5 \times 10^6 \text{ N-mm}$$

From Table 10.7, we find that for a rectangular section, section modulus,

$$Z = t\left(b.l + \frac{b^2}{3}\right) = 0.707\,s\left[150 \times 100 + \frac{(150)^2}{3}\right] = 15\,907.5\,s \text{ mm}^3$$

$$\text{Bending stress, } \sigma_b = \frac{M}{Z} = \frac{12.5 \times 10^6}{15\,907.5\,s} = \frac{785.8}{s} \text{ N/mm}^2$$

We know that maximum shear stress (τ_{max}),

$$75 = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\,\tau^2} = \frac{1}{2}\sqrt{\left(\frac{785.8}{s}\right)^2 + 4\left(\frac{70.72}{s}\right)^2} = \frac{399.2}{s}$$

$$s = 399.2 / 75 = 5.32 \text{ mm}$$ **Ans.**

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UNIT IV-ENERGY STORING ELEMENTS AND ENGINE COMPONENTS

CONTENTS

Spring

A spring is an elastic member, which deflects, or distorts under the action of load and regains its original shape after the load is removed

Leaf springs

A leaf spring consists of flat bars of varying lengths clamped together and supported at both ends, thus acting as a simply supported beam.

Belleville Springs

They are made in the form of a cone disc to carry a high compressive force. In order to improve their load carrying capacity, they may be stacked up together. The major stresses are tensile and compressive.

Spring index (C)

The ratio of mean or pitch diameter to the diameter of wire for the spring is called the spring index.

Pitch

The axial distance between adjacent coils in uncompressed state

Surge in springs

The material is subjected to higher stresses, which may cause early fatigue failure. This effect is called as spring surge.

Buckling of springs

The helical compression spring behaves like a column and buckles at a comparative small load when the length of the spring is more than 4 times the mean coil diameter

Nipping

Pre-stressing of leaf springs is obtained by a difference of radii of curvature known as nipping.

Spring rate (stiffness).

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The spring stiffness or spring constant is defined as the load required per unit deflection of the spring.

Spiral spring.

A flat spiral spring is a long thin strip of elastic material wound like a spiral. These springs are frequently used in watch springs, gramophones, e.t.c

Spring is an elastic body whose function is to distort when loaded and to recover its original shape when the load is removed.

4.1 APPLICATION OF SPRINGS

1. To apply forces as in brakes, clutches and spring loaded valves.
2. To store energy as in watches, toys.
3. To measure forces as in spring balance and engine indicators.
4. To cushion, absorb or control energy due to either shock or vibration as in car.

4.2 TYPES OF SPRINGS

1. Helical springs
2. Conical and volute spring
3. Torsion spring
4. Laminated or leaf spring
5. Disc or bellevile spring
6. Special purpose spring

4.3HELICAL SPRINGS

The helicall springs are made up of a wire coiled in the form of helix and is primarily intended for tensile or compressive loads. The cross section of the wire from which the spring made may be circular, square or rectangular. The two forms of helical springs are compression spring and helical tension springs.

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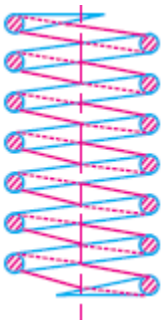

(*a*) Compression helical spring.

(*b*) Tension helical spring.

Fig 4.1 helical spring in compression and tension

4.3.1TERMS USED IN COMPRESSION SPRING

SOLID LENGTH

When the compression spring is compressed until the coils come in contact with each other the spring is said to be solid. The solid length of a spring is the product of total number of coils and the diameter of the wire.

LS=n'*d

n'- total number of coils

d- diameter of the wire

FREE LENGTH

It is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils.

$$LF=n'*d+\delta max+0.15\ \delta max$$

SPRING INDEX

It is defined as the ratio of the mean diameter of the coil to the diameter of the coil to the diameter of the wire.

$$C=D/d$$

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D- mean diameter of coil

d- diameter of wire

SPRING RATE

It is defined as the load required per unit deflection of the spring.

$$q=P/y$$

P- applied load

y- deflection of the spring

PITCH

The pitch of the coil is defined as the axial distance between adjacent coil in uncompressed state.

$$\text{Pitch length}=\text{free length}/(n'-1)$$

4.3.2 ENDS FOR COMPRESSION HELICAL SPRING

In all springs the end coils produce an eccentric application of the load, increasing the stress on one side of the spring. Under certain conditions, especially where the number of coils is small, this effect must be taken into account. The nearest approach to an axial load is secured by squared and ground ends, where the end turns are squared and then ground perpendicular to the helix axis. It may be noted that part of the coil which is in contact with the seat is in contact with the seat does not contribute to spring action and hence are termed as inactive coils. The turns which impact spring action are known as active turns. As the load increases, the number of inactive coils also increased due to seating of the end coils and the amount of increase varies from 0.5 to 1 turn at usual working loads.

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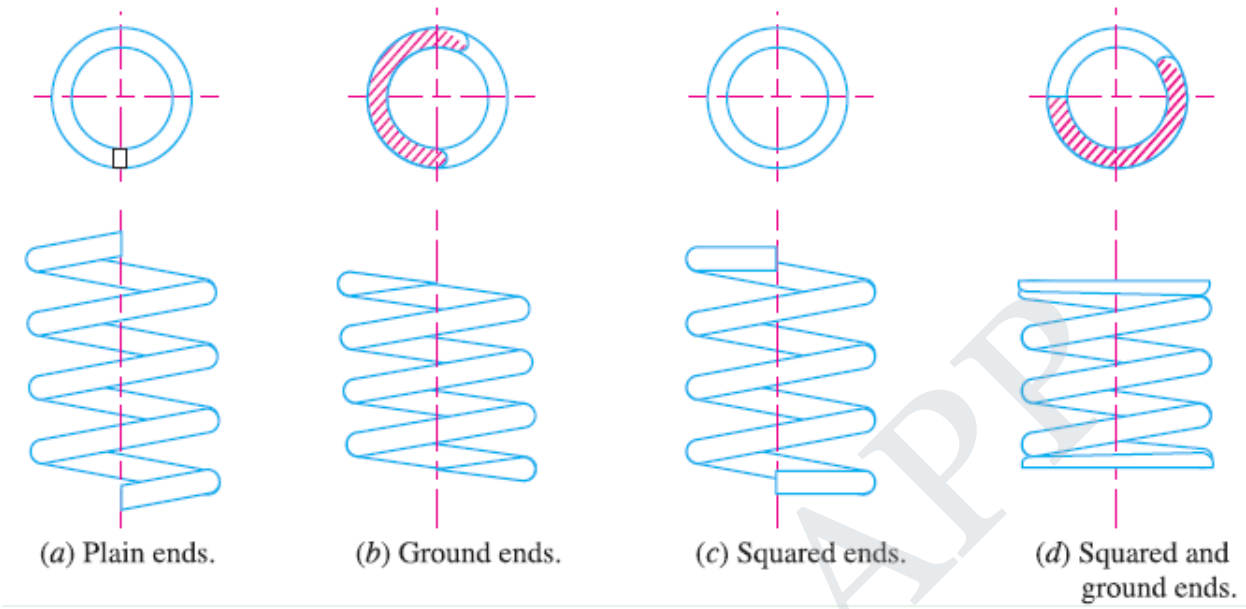

Fig 4.2 various ends in springs

Table 4.1 ends in springs

Type of end	*Total number of turns (n')*	*Solid length*	*Free length*
1. Plain ends	n	$(n + 1)\ d$	$p \times n + d$
2. Ground ends	n	$n \times d$	$p \times n$
3. Squared ends	$n + 2$	$(n + 3)\ d$	$p \times n + 3d$
4. Squared and ground ends	$n + 2$	$(n + 2)\ d$	$p \times n + 2d$

n- number of active turns

p-pitch of coils

d-diameter of spring

4.4 DESIGN PROCEDURE OF HELICAL SPRING

1. Calculate spring stiffness
2. Calculate spring index

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3. Calculate solid length (ls)
4. Calculate free length(lf)
5. Calculate pitch(P)

4.4.1STRESSES IN HELICAL SPRING

Consider a helical compression spring made of circular wire and subjected to an axial load W

Maximum shear stress induced in the wire =$\mathbf{k_s x 8 x w x d/\pi d^3}$

Ks- Shear stress factor

Ks When the springs are subjected to static loading the effect of wire curvature may be neglected because yielding of material will relive the stresses. In order to consider the effect of both direct shear as well as curvature of the wire Wahl's stress factor is introduced.

$$\mathbf{Ks=1+1/2C}$$

Maximum shear stress introduce in wire =K X 8 X WX D/ $\boldsymbol{\pi d^3}$

$$\mathbf{K=4C-1/4C-4+0.615/C}$$

4.4.2DEFLECTION OF HELICAL SPRING OF CIRCULAR WIRE

$$Y= K\ X8XWXD/\ \mathbf{Gd^3}$$

4.4.3STIFFNESS OF SPRING (or) SPRING RATE

$$q=w/deflection =8XDxn^3/\ \mathbf{Gd^3}$$

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The laminated or leaf spring consists of a number of flat plates of varying lengths held together by means of clamps and bolts. These are mostly used in automobiles.

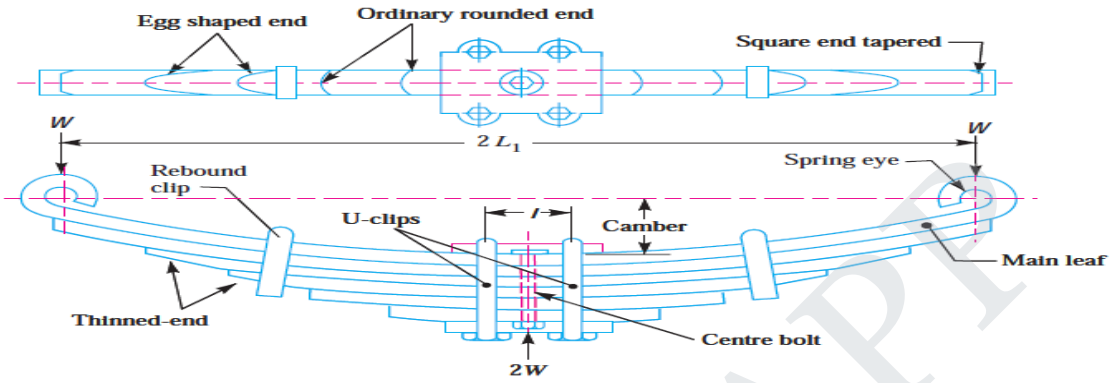

FIG 4.2

A leaf spring commonly used in automobiles is of semielliptical form. It is built up of a number of plates. The leaves are usually given an initial curvature or cambered so that they will tend to straighten under the load. The leaves are held together by means of a band shrunk around them at the center or by a bolt passing through the center. Since the band exerts stiffening and strengthening effect, therefore the effective length of the spring for bending will be overall length of spring minus width of band. The spring is clamped to the axle housing by means of U bolts. The longest leaf known as main leaf or master leaf has its ends formed in the shape of an eye through which the bolts are passed to secure the spring to its support. Usually the eyes through which the spring is attached to the hanger or shackle, are provided with bushings of some antifriction materials such as bronze or rubber. The other leaves of the springs are known as graduated leaves. Rebound clips are located at intermediate positions in the length of the spring so that the graduated leaves also share the stress induced in the full length of leaves when the spring rebounds.

2L1- length of span (or) overall length of spring

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L-width of band (or) distance between centers of U bolt, ineffective length of springh effective length of spring

2L=2L1-L Length of smallest leaf

Length of smallest leaf =Effective length /n-1+ ineffective length

Length of next leaf =Effective length /n-1x2+ ineffective l

Length of master leaf=2L1+π (d+t)*2

Bending stress in the spring σ_b = **$6WL/nbt^2$**

Deflection of the spring = **$6WL^3/nEbt^2$**

When the leaves are not initially stressed therefore maximum stress or bending stress for full length leaves □ □

$$\sigma_b = 18WL/bt^2 (2n_g+3n_f)$$

n - Number of leaves

nG - number of graduated leaves

nF - number of extra full length leaves

For graduated leaves

$$\sigma_g = 12WL/bt^2 (2n_g+3n_f)$$

Deflection

$$\delta = \frac{12WL^3}{Ebt^3(2n_G + 3n_F)}$$

Initial gap (or) Nip

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$$C = \frac{2WL^3}{nEbt^3}$$

Load on the bolt to close the nip

$$P_b = \frac{2Wn_G n_F}{n(2n_G + 3n_F)}$$

4.6 SOLVED PROBLEMS

HELICAL SPRING:

1) **Design a helical compression spring for a maximum load of 1000 N for a deflection f 25 mm using the value of spring index as 5. The maximum permissible shear stress for spring wire is 420 MPa and modulus of rigidity is 84 kN/mm^2.**

Given:

W = 1000 N ; δ = 25 mm ; C = D/d = 5 ; τ = 420 MPa = 420 N/mm^2 ;
G = 84 KN/mm^2 = 84 x 10^3 N/mm^2

Solution:

Mean diameter of the spring coil:

Let D = Mean diameter of the spring coil, and

d = Diameter of the spring wire.

We know that Whal's stress factor,

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4X5-1}{4X5-4} + \frac{0.615}{5} = 1.31$$

And maximum shear stress (τ),

$$420 = K\ X\ \frac{8\ X\ W\ X\ C}{\pi\ d^2} = 1.31\ X\ \frac{8\ X\ 1000\ X\ 5}{\pi\ d^2} = \frac{16.667}{d^2}$$

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$d^2 = 16.667 / 420 = 39.7$ or $d = 6.3$ mm

From PSG Design data book page no: 13.1

We shall take a standard wire of size SWG3 having diameter (d) = 6.401 mm

Mean diameter of the spring coil,

$D = C.d = 5\ d = 5 \times 6.401 = 32.005$ mm

And outer diameter of the spring coil,

$D_o = D + d = 32.005 + 6.401 = 38.406$ mm

Number of turns of the coils:

Let n = Number of active turns of the coils,

We know that compression of the spring (δ),

$$25 = \frac{8.W.C^3 n}{G.d} = \frac{8 \times 1000 \times (5)^3 n}{84 \times 10^3\ 6.401} = 1.86\ n$$

$n = 25 / 1.86 = 13.44$ say 14

For squared and ground ends, the total number of turns,

$n' = n + 2 = 14 + 2 = 16$

Free length of the spring:

We know that free length of the spring

$= n' . d + \delta + 0.15\ \delta = 16 \times 6.401 + 25 + 0.15 \times 25$

$= 131.2$ mm

Pitch of the coil:

We know that pitch of the coil,

$$= \frac{Free\ length}{n' - 1} = \frac{131.2}{16-1} = 8.75 \text{ mm}$$

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CONCENTRIC SPRING:

2) A concentric spring for an aircraft engine valve is to exert a maximum force of 5000N under an axial deflection of 40 mm. Both the spring have free length, same solid length and are subjected to equal maximum shear stress of 850 MPa. If the spring index for both the springs is 6, find (a) the load shared by both spring, (b) the main dimensions of the both spring, and (c) the number of active coils in each spring.

Assume G = 80 KN/mm^2 and diametral clearance to be equal to the difference between the wire diameters.

Given:

W = 5000 N ; δ = 40 mm ; $\tau_1 = \tau_2$ = 850 MPa = 850 N/mm^2 ; C = 6 ;

G = 80 KN/mm^2 = 80 x 10^3 N/mm^2.

Solution:

(a) Load shared by each spring:

Let W_1 and W_2 = Load shared by outer and inner spring respectively

d_1 and d_2 = diameter of spring wires for outer and inner spring respectively, and

D_1 and D_2 = Diameter of the outer and inner springs respectively.

Since the diametral clearance is equal to the difference between the wire diameters, therefore,

$$(D_1 - D_2) - (d_1 - d_2) = d_1 - d_2$$

$$(D_1 - D_2) = 2\, d_1$$

We know that $D_1 = C \,.d_1$, and $D_2 = C \,.\, d_2$

$$C \,.d_1 - C \,.\, d_2 = 2\, d_1$$

Or $$\frac{d_1}{d_2} = \frac{C}{C-2} = \frac{6}{6-2} = 1.5$$

We also know that $\frac{W_1}{W_2} = \left(\frac{d_1}{d_2}\right)^2 = (1.5)^2 = 2.25$

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And $W_1 + W_2 = W = 5000N$

W_1 = 3462 N , and W_2 = 1538 N

(b) Main dimensions of both the springs:

We know that Whal's stress factor for both the springs,

$$K_1 = K_2 = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4X6-1}{4X6-4} + \frac{0.615}{6} = 1.2525$$

And maximum shear stress induced in the outer spring (τ_1),

$$850 = K_1 X \frac{8 X W_1 X C}{\pi d_1^2} = 1.2525 X \frac{8 X 3462 X 6}{\pi (d_1^2)} = \frac{66243}{(d_1^2)}$$

$(d_1)^2 = 66243 / 850 = 78$ or $d_1 = 8.83$ say 10 mm

$D_1 = C. d_1 = 6 \times 10 = 60$ mm

Similarly, maximum shear stress induced in the inner spring (τ_2),

$$850 = K_2 X \frac{8 X W_2 X C}{\pi d_2^2} = 1.2525 X \frac{8 X 1438 X 6}{\pi (d_2^2)} = \frac{29428}{(d_2^2)}$$

$(d_2)^2 = 29428 / 850 = 34.6$ or $d_2 = 5.88$ say 6 mm

$D_2 = C . d_2 = 6 \times 6 = 36$ mm

(c) Number of active coils in each spring:

Let n and n = Number of active coils of the outer and inner spring respectively

We know that the axial deflection for the outer spring (δ),

$$40 = \frac{8 X W_1 X C^3 X n1}{G . d_1} = \frac{8 X 3462 X 6^3 X n_1}{80 X 10^3 X 10} = 7.48\ n_1$$

$n_1 = 40 / 7.48 = 5.35$ say 6

assuming square and ground ends for the spring, the total number of turns of the outer spring

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$n_1' = 6 + 2 =$ 8 solid length of the outer spring

$$L_{S1} = n_1' . d_1 = 8 \times 10 = 80 \text{ mm}$$

Let n_2' be the total number of turns of the inner spring. Since both the spring have the same solid length, therefore,

$$n_2' . d_2 = n_1' . d_1$$

$$n_1' = \frac{n1' . d_1}{d_2} = 8 \times 10 / 6 = 13.3 \text{ say } 14$$

$$\mathbf{n_2} = 14 - 2 = \mathbf{12}$$

since both the springs have the same free length, therefore free length of outer spring,

= free length of inner spring

$= L_{S1} + \delta + 0.15\,\delta = 80 + 40 + 0.15 \times 40 =$ **126 mm**

Other dimensions of the spring are as follows, Outer diameter of the outer spring

$= D_1 + d_1 = 60 + 10 =$ **70 mm**

Inner diameter of the outer spring

$= D_1 - d_1 = 60 - 10 =$ **50 mm**

Outer diameter of the inner spring

$= D_2 + d_2 = 36 + 6 =$ **42 mm**

Inner diameter of the inner spring

$= D_2 - d_2 = 36 - 6 =$ **30 mm**

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LEAF SPRING:

3) **A truck spring has 12 number of leaves, two of which are full length leaves. The spring supports are 1.05 m apart and the central band is 85 mm wide. The central load is to be 5.4 KN with a permissible stress of 280 MPa. Determine the thickness of and width of the steel spring leaves. The ratio of the total depth to the width of the spring is 3. Also determine the deflection of the spring.**

Given:

$n = 12$; $n_F = 2$; $2L_1 = 1.05$ m = 1050 mm ; $l = 85$ mm ; $2W = 5.4$ KN = 5400 N or $W = 2700$ N ; $\sigma_F = 280$ MPa = 280 N/ mm^2

Solution:

Thickness and width of the spring leaves

Let t = thickness of the leaves, and

b = width of the spring leaves.

Since it is given that the ratio of the total depth of the spring (n x t) and width of the spring (b) is 3, therefore

$\frac{n \times t}{b} = 3$ or b = n x t / 3 = 12 x t / 3 = 4t

We know that the effective length of the spring,

$2L = 2L_1 - l = 1050 - 85 = 965$ mm

$L = 965 / 2 = 482.5$ mm

And number of gradual leaves,

$n_G = n - n_F = 12 - 2 = 10$

Assuming that the leaves are not initially stressed, therefore maximum stress or bending stress for full length leaves (σ_F),

$$280 = \frac{18 . W . L}{b.t^2 (2n_G + 3n_F)} = \frac{18 \times 2700 \times 482.5}{4t \times t^2 (2 \times 10 + 3 \times 2)} = \frac{225476}{t^3}$$

$t^3 = 225476 / 280 = 805.3$ or t = 9.3 say **10mm**

b = 4 t = 4 x 10 = **40 mm**

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Deflection of the spring

We know that deflection of the spring,

$$\delta = \frac{12 \cdot W \cdot L^3}{E.b.t^3\ (2n_G+3n_F)} = \frac{12\ X\ 2700\ X\ (482.5)^3}{210\ X\ 10^3 X 40\ X\ 10^3\ (2\ X\ 10+3\ X\ 2)}$$

δ = 16.7 mm

4) A locomotive semi – elliptic laminated spring has an overall length of 1 m and sustains a load of 80 KN at its centre. The spring has 3 full length leaves and 15 graduated leaves with a central band of 100 mm width. All the leaves are to be stressed to 400 MPa, when fully loaded. The ratio of the total spring depth to that of width is 2. E = 210 KN/mm^2. Determine,

(a). The thickness and width of the leaves.

(b). The initial gap that should be provided between the full length and gradual leaves before the band load is applied.

(c). The load exerted on the band after the spring is assembled.

Given:

$2L_1$ = 1 m = 1000 mm ; 2W = 70 KN or W = 35 KN = 35 x 10^3 N ; n_F = 3 ; n_G = 15; l = 100 mm ; σ = 400 MPa = 400 N /mm^2 ; E = 210 KN/ mm^2 = 210 x 10^3 N/mm^2

Solution:

Thickness and width of leaves

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We know that the total number of leaves,

$$n = n_F + n_G = 3 + 15 = 18$$

since it is given that ratio of the total spring depth (n x t) and width of leaves is 2, therefore

$$\frac{n\ X\ t}{b} = 2 \text{ or } b = n \times t / 2 = 18 \times t / 2 = 9\ t$$

We know that the effective length of the leaves,

$$2L = 2L_1 - l = 900 \text{ mm or } L = 900/2 = 450 \text{ mm}$$

Since all the leaves are equally stressed, therefore final stress (σ),

$$400 = \frac{6\ .W\ .\ L}{n\ .b\ .t^2} = \frac{6\ X\ 35\ X\ 10^3\ X\ 450}{18\ X\ 9t\ X\ t^2} = \frac{583\ X\ 10^3}{t^3}$$

$$t^3 = 583 \times 10^3 / 400 = 1458 \text{ or } \mathbf{t} = 11.34 \text{ say } \mathbf{12\ mm}$$

$$\mathbf{b} = 9\ t = 9 \times 12 = \mathbf{108\ mm}$$

Initial gap

We know that the initial gap (C) that should be provided between the full length and gradual leaves before the band load is applied, is given by

$$\mathbf{C} = \frac{2\ .W\ .\ L^3}{n\ .E\ .b\ .t^3} = \frac{2\ X\ 35\ X\ 10^3\ X\ (450)^3}{18\ X\ 210\ X\ 10^3\ X\ (12)^3} = \mathbf{9.04\ mm}$$

Load exerted on the band after the spring is assembled

We know that the load exerted on the band after the spring is assembled,

$$\mathbf{W_b} = \frac{2\ .n_F.n_G\ W}{n\ (2n_G+3n_F)} = \frac{2\ X\ 3\ X\ 15\ X\ 35\ X\ 10^3}{18\ X\ (2\ X\ 15+3\ X\ 3)} = \mathbf{4487\ N}$$

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UNIT V -DESIGN OF BEARINGS

TECHNICAL TERMS

Bearing

Bearing is a stationery machine element which supports a rotating shafts or axles and confines its motion.

Journal bearing

A journal bearing is a sliding contact bearing which gives lateral support to the rotating shaft.

Self – acting bearing

The pressure is created within the system due to rotation of the shaft, this type of bearing is known as self – acting bearing

Anti friction bearing

Rolling contact type bearings are called as anti friction bearing since it offers less resistance during starting and running.

Square bearing

If the length to diameter ratio is unity then the bearing is called as square bearing.

Flywheel

A flywheel used in machine serves as a reservoir which stores energy during the period when the supply of energy is more than the requirement and releases it dulling the period when the requirement of energy is more than the supply.

Fluctuation of speed' and 'fluctuation of energy'.

The ratio of maximum fluctuation of speed to the mean speed is called co efficient of fluctuation of speed.

The ratio of fluctuation of energy to the mean energy is called coefficient of fluctuation of energy.

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5. Bearing introduction

A bearing is a machine element which supports another moving machine element. It permits a relative motion between the contact surfaces of the members while carrying the load. Due to the relative motion between the surfaces a certain amount of power is wasted in overcoming frictional resistance and if the rubbing surfaces are indirect contact there will be rapid wear. In order to reduce frictional resistance and wear resistance in some cases to cases carry away the heat generated a layer of fluid may be provided.

Depend upon the nature of contact

(i) Sliding contact bearing

The sliding takes place along the surfaces of contact between the moving element and fixed element

(ii) Rolling contact bearing

The steel balls or rollers are interposed between the moving and fixed element. The balls offer rolling friction at the two points for each ball or roller

5.1JOURNAL BEARING

A sliding contact bearing that supports a load in a radial direction is known as journal bearing. It consists of two main parts, a shaft and a hollow cylinder. The portion of the shaft inside the hollow cylinder also known as bearing is called as journal. In most applications the journal rotates while the bearing is stationary. However there are some applications where the journal is stationary and the bearing rotates and even somewhere both the journal and bearing rotates. This journal bearing may be classified as full journal bearing and partial journal bearing depending upon whether the journal is fully or partially covered by bearing.

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1. Diameter clearance: It the different between the diameter of journal and the bearing.

C=D-d

2. Radial clearance: it is the different between the radial of the bearing and the journal. c_1=R-r=D-d/2

3. Diameteral clearance ratio: It is the ratio of the diameteral clearance to the diameter of the journal. =c/d=(D-d)/d

4. Eccentricity: It is the radial distance between the center of the bearing and the displaced center of the bearing under load.

5. Minimum oil film thickness: It is the minimum distance between the bearing and the journal under complete lubrication condition. It is denoted by ho and occurs at the line of centers.

5.1.2 COEFFICENT OF FRICTION FOR JOURNAL BEARINGS

To determine the coefficient of friction for well lubricated full journal bearings, the following empirical relation established by McKee based on the experimental data

$$\mu=33/10^8(ZN/P)(d/P)+K$$

μ- Coefficient of friction Z- Absolute viscosity of oil

N- Speed of journal in r.p.m

p- Bearing pressure on the projected bearing area

d- Diameter of journal

l- Length of bearing

c- Diameter clearance

k- Factor to correct for end leakage.

5.1.3 CRITICAL PRESSURE OF THE JOURNAL BEARING

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The pressure at which The oil film breaks down so the metal to metal contact begins, is known as critical pressure or the minimum operating pressure of the bearing. It may be obtained by the following relation

$$P = (ZN/47.5X10^{6})(d/C^{2})(l/D+l)$$

5.1.4SOMMERFELD NUMBER

The sommerfeld number is also a dimensionless parameter used extensively in the design of journal bearings.

$$\text{Sommerfeld number} = (ZN/P)(d/C)^{2}$$

5.1.5 HEAT GENERATED IN A JOURNAL BEARING

The heat generated in a bearing due to the fluid friction and friction of the parts having relative motion.

$$Qg=WV\mu$$

After the thermal equilibrium is reached heat will be dissipated at the outer surface of the bearing at the same rate at which it is generated in the oil film. The amount of heat will depend upon the temperature difference size and mass of the radiating surface and on the amount of air flowing around the bearing. However for the convenience in bearing design the actual heat dissipating area may be expressed in terms of the projected area of the journal.

5.1.6 Heat dissipated by the bearing Qd= CA (T_b-T_a)

C- heat dissipation coefficient

A- projected area of the bearing

t_b-temperature of the bearing surface

t_a- temperature of the surrounding air

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1. Calculate diameterof the journal p=2TNπ/60
2. Select the value of l/d ratio
3. Calculate the clearance ratio c/d
4. Determine the length of bearing.
5. Calculate the bearing pressure. P=w/ld
6. Select the type of oil used.
7. Calculate Somerfield number
4. Determine the amount of heat generated.
5. Determine the amount of heat rejected.

5.3ROLLING CONTACT BEARING

The advent of automobiles and many high speed machineries make very much use another type of bearings known as rolling contact bearings. The friction produced in these bearing is very low. These bearings also called as antifriction bearings. They differ from sliding contact bearings in their structure and usage.

5.3.1. COMPONENTS OF ROLLING CONTACT BEARINGS

The rolling bearing consists of four main components (1) the inner ring, (2) outer ring, (3) the balls or rollers, (4) the retainers or separators. The inner ring is forced to fitted with machine shaft and outer ring is fitted with machine housing. The shaft rotates because of relative rotations of balls or rollers. The retainers are used to prevent the balls or rollers from ejecting out during operation.

5.3.2CLASSIFICATION OF ROLLING CONTACT BEARINGS

The rolling contact bearings are classified into two major groups with respect to their structure

(1) Ball bearings

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(2) Roller bearings

Basically the structure of ball bearings are similar expect that whether the rolling element between the inner ring and outer ring are balls or rollers. Also these ball bearings are many types such as deep grove ball bearings, angular contact bal bearings and so on. Both type of bearing can carry radial loads and axial loads acted individually or in combined form. Generally the ball bearings are used for light loads and the roller bearings are usually used for heavier loads. Also in the case of ball bearings the nature of contact is the point contact hence the friction produced is very less compared to roller bearings where the nature of contact is the line contact which produce more friction.

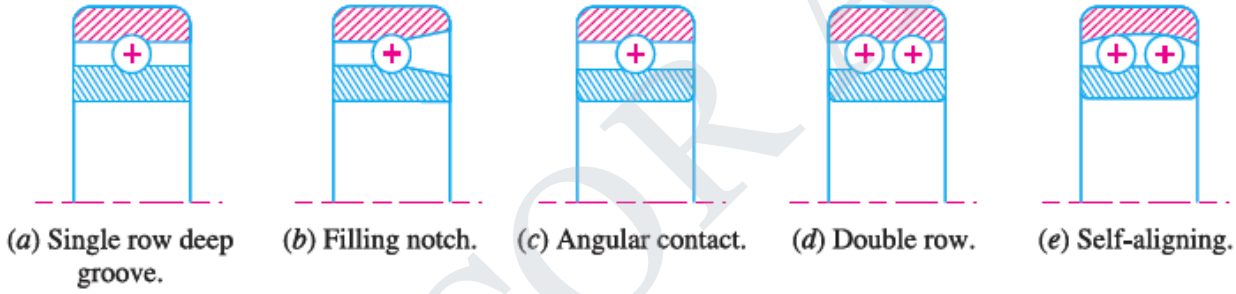

Figure 5.1 Types of radial ball bearing

5.3.3SELECTION OF BEARINGS FOR STEADY LOADING

The size of bearing required is judged by the magnitude and nature of applied load, life and reliability. The bearing load is composed of weights involved forces derived from power transmitted and additional force based on method of operation.

$$C=(L/L10)^{1/K}xP$$

c-basic dynamic load rating

L- life of bearing in million revolutions

L10- life of bearing for 90% survival at 1 million revolutions

P- Equivalent load

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k- exponent (3 for ball bearing, 10/3 for roller bearings)

P=(XFr+YFa)S

Fr- radial load

Fa- axial load

X- Radial load factor

Y- Axial load factor

S- Service factor

5.3.4SELECTION OF BEARING FOR VARIABLE LOADING

The rolling contact bearing are frequently operate under variable load and speed conditions. This is due to many causes like power fluctuation in electrical machineries or requirement of different cutting forces for different kinds of machining tools, or running with loading and unloading condition as in automobiles. Such as variable loaded bearings are designed by considering all these different loaded conditions of work cycle and not solely upon most sever operating conditions. The work cycle may be divided into a number of portions in each of which operation condition may be taken as constant.

$$\mathbf{P_m = (p_1^3 n_1 + p_2^3 n_2 + p_3^3 n_3 / \sum_n)}$$

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A flywheel used in machines serves as a reservoir which stores energy during the period when the supply of energy is more than the requirement and release it during the period when the requirement of energy is more than supply.

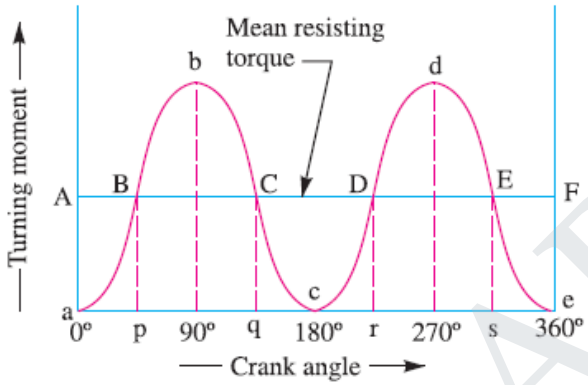

Figure 4.3 turning moment diagram for single cylinder double acting steam engine

The fluctuation of energy may be determined by the turning moment diagram for one complete cycle of operation. Consider a turning moment diagram for a single cylinder double acting steam engine. The vertical ordinate represents the turning moment and the horizontal ordinate represents the crank angle. a little consideration will show that the turning moment is zero when the crank angle is zero. It rises to maximum value when crank angle reaches 90° and it again zero when crank angle reaches 180°. This is shown by curve ABC in figure and it represents the turning moment for outstroke. The curve CDE is turning moment diagram for in stroke and is somewhat similar to the curve ABC. The work done is the product of turning moment and angle turned, therefore the area of the turning moment diagram represents the work done per revolution. The difference between the maximum and minimum speeds during a cycle is called the maximum fluctuation of speed. The ratio of the maximum fluctuation of speed to the mean speed is called coefficient of fluctuation of speed

Coefficient of fluctuation of speed.

$$C_S = \frac{N_1 - N_2}{N} = \frac{2(N_1 - N_2)}{(N_1 + N_2)}$$

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Maximum fluctuation of energyΔ_E= maximum energy-minimum energy

coefficent of fluctuation of energy $C_E = \frac{\max imum fluctuation of energy}{work done per cycle}$

work done per cycle=T_{mean}*θ

θ =angle turned in radians per revolution

=2π in case of stream engine and two stroke I.C engine

=4π in case of four stroke I.C engines

energy stored in flywheel rim

Maximum fluctuation of energyΔ_E= $I\omega^2\left(\frac{\omega_1-\omega_2}{\omega}\right)=I\omega^2C_S=mk^2\omega^2C_S=2EC_S$

k may be equal to mean radius of rim(R) because the thickness of rim is very small as compared to the diameter of rim substituting k=R

$\Delta_E= mR^2\omega^2C_S=mv^2C_S$

Mass of the flywheel rim $m=2\pi R4\rho$

A=b*t

b-width of rim , t- thickness of rim

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1) A ball bearing is rated by manufacture to have an average life of 1000hrs when carrying a load of 45kN at 1800 rpm

(i) What is the 90% life expectancy for these bearing ?

(ii) What is the rated load capacity of this bearing for an average life of 50000hrs at 1800rpm

Solution:

From relation $\frac{L}{L'_{10}} = \left[\frac{\ln(1/p)}{\ln(1/p_{10})}\right]^{1/b} = \left[\frac{\ln(1/0.5)}{\ln(1/0.9)}\right]^{1/1.17}$

$$\frac{L}{L'_{10}} = 5$$

$$L'_{10} = \frac{L}{5} = \frac{1000}{5} = 200hrs$$

$$\frac{L}{L'_{10}} = \left[\frac{\ln\left(1/0.5\right)}{\ln\left(1/0.9\right)}\right]^{1/1.17}$$

Rated life, $L'_{10} = \frac{L}{5} = \frac{50000}{5} = 10000hrs$.

Life in million rev.$=\frac{60nL_h}{10^6} = \frac{60X1800X10000}{10^6}$

$$L=1080mr$$

We known that,

$$L=\left(\frac{C}{P}\right)^b$$

$$1080=\left(\frac{C}{45000}\right)^3$$

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$C=461693.51N$ (Ans)

Result:

(i) Life expectancy ,L_h=200hrs
(ii) Rated load capacity , C= 461693.51N

2) Select a suitable ball bearing for a drilling machine spindle of diameter 40mm rotating at 3000rpm. It is subjected to a radial load of 2000N and axial thrust of 1000N. it is to work for 45hours a week for one year.

Given data:

Diameter of shaft =40mm

Speed , N=3000rpm

Radial load, F_r=2000N

Thrust load, F_a=1000N

Life, L_h=45*hrs a week for one year*

To find :

Selection of bearing

Solution :

Since $\frac{F_a}{F_r}$<0.7, so a single row deep groove ball bearing may be suitable.

For the given diameter of shaft 400mm, from PSGDB4.14, select SKF6308 bearing

.

Static load rating, C_o=2200kgf=32000N

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Dynamic load rating, C=3200kgf=32000N

$$\frac{F_a}{C_o} = \frac{1000}{22000} = 0.0625$$

PSGDB 4.4, corresponding to $\frac{F_a}{C_o} = 0.045$

e=0.245(by interpolation)

Since $\frac{F_a}{F_r} = \frac{1000}{2000} = 0.5 > e$, from table 5.2 or PSGDB 4.4,

The radial load factor X=0.56 and Y=1.75 (by interpolation).

The service factor is selected from table 5.3 or PSGDB 4.2 AS 1.3

Equivalent load,

$P=(X F_r + Y F_a) S = (0.56 X 2000 + 1.75 X 1000) 1.3$

P=3731N

Life ,L_h=45*hrs* a week for one year = 45X 52 = 2340 *hrs*.

From graph in fig. 5.8 or PSGDB 4.6, corresponding to 3000 rpm and 2340 hrs of life

Loading ratio

$$\frac{C}{P} = 7.8$$

C=7.8 X P = 7.8 X 3731 = 29101.8 N

Since the dynamic load rating of the SKF6208 bearing is more than the required dynamic load capacity, the selected bearing is suitable.

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3) Select a suitable Conrad-type deep-groove ball bearing for the following data, the radial load is 7500N and axial load is 4500N, the shaft speed is 2000rpm, the L_{10} life required is $4.9X10^8$ revolutions: the inner ring of the bearing rotates.

Given data:

Speed, N=2000rpm

Radial load ,F_r=7500N

Thrust load, F_a=4500N

Life, L_{10}=$4.9X10^8$ revolution =490 million rev.

To find :

Selection of bearing.

Solution:

Since the diameter of shaft is not given in the problem, assume the diameter of shaft as 80mm. from PSGDB 4.14, select SKF6416 bearing.

Static load rating, C_o=12800kgf=128000N

Dynamic load rating, C = 13000kgf = 130000N

$$\frac{F_a}{C_o} = \frac{4500}{128000} = 0.035$$

PSGDB 4.4, corresponding to $\frac{F_a}{C_o} = 0.035$

E=0.235(by interpolation)

Since $\frac{F_a}{C_o} = \frac{4500}{7500} = 0.6 > e$, from table 5.2 or PSGDB 4.4,

The radial load factor X = 0.56 and Y =1.85 (by interpolation)

The service factor is selected from table 5.3 or PSGDB 4.2 and 1.2

Equivalent load,

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$$P = (XF_r + Y F_0)S = (0.56X\ 7500 + 1.85\ X\ 4500)\ 1.2$$

$$P = 15030N$$

The relation between life in million revolutions and life in working hours is given by

$$L = \frac{60nL_n}{10^6}$$

$$490 = \frac{60X2000XL_h}{10^6}$$

$$L_h = 4083.3 \text{ hrs}$$

PSGDB 4.6, corresponding to 2000rpm and 4083.3 hrs of life

Loading ratio, $\frac{C}{P} = 7.8$

$$C = 7.8 \times P = 7.8\ X\ 15.3. = 119234\ N$$

Since the dynamic load rating of the SKF6416 bearing is more than the required dynamic load capacity, the selected bearing is suitable

Result :

The selected bearing is SKF6416 deep groove ball bearing.

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JOURNAL BEARING :

1) **Design a journal bearing for a centrifugal pump to the following specification:**

Diameter of the journal =75mm

Speed of the journal =1140 rpm

Load of each journal =11500N

Given data:

Diameter of the journal =75mm

Speed of the journal =1140 rpm

Load oc each journal =11500N

Solution:

Step I: From PSGDB 7.31 for centrifugal pumps, the $\frac{L}{D}$ ratio is $\frac{L}{D}$=1 to 2

Let us take $\frac{L}{D} = 1.5$

Length of journal, L = 1.5 X 75 = 112.5 mm

Step 2: pressure developed

$$p=\frac{W}{LD}=\frac{11500}{75X112.5} = 1.363 \text{ N/mm}^2$$

this pressure is within the safe limit (0.7 to 1.4 N/mm^2)from PSGDB 7.31

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Selection of lubricating oil the minimum value of $\frac{Zn}{p}$ from PSGDBE 7.31 is

$$\left(\frac{Zn}{p}\right)_{min} = 2844.5$$

$$Z_{min} = \frac{2844.5 \times 13.63}{1140}$$

Zmin = 34 centipoise

From PSGDB, at 34CP and 60degree cilices , the next standard higher value of SAE 40oil will be selected

The actual viscosity of SAE 40 oil is

Z + 40.5 CP

Step 4:

Calculation of coefficient of friction,

$$\mu = \frac{33.25}{10^{10}}\left(\frac{Zn}{p}\right)\left(\frac{D}{C}\right) + K$$

where,

C-Diametral clearance, it may be assumed as 150microns from PSGDB 7.32

$$\frac{D}{C} = \frac{75}{150 \times 10^{-3}} = 500$$

K= 0.002 for $0.75 < \frac{L}{D} < 2.8$

$$\mu = \frac{33.25}{10^{10}}\left(\frac{40.5 \times 1140}{1.363}\right) \times 500 + 0.002$$

μ = 0.0583

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step 5: Heating calculation

Heat generated, $H_g = \mu W V$

Where V =rubbing velocity = $\frac{\pi Dn}{60} = \frac{\pi \times 0.075 \times 1140}{60}$

$V = 4.476$ m/s

$$H_g = \frac{(\Delta t+18)^2 LD}{K}$$

$$= \frac{((60-28)+18)^2 \times 0.075 \times 0.1125}{0.484}$$

$= 45.58$ W

Since the heat generated is more than the heat dissipated, artificial cooling arrangement must be provided. This cooling arrangement can be done by providing cooling fans os by circulated water.

Diameter of bearing, $D_b = D + C = D + \frac{D}{1000}$

$= D (1+ 0.001) = 75 \times 1.001$

$= 75.075$mm

Material: From PSGDB 7.30 for compressors material chosen is Lighter liner on steel or bronze backing

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