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Mechanical Engineering Department

Sub. : - DME-I First midterm test (15-02-2018)

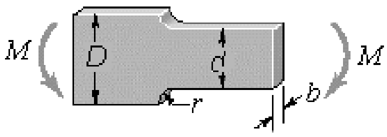
Time 1hr MM: - 10

1. Design a cotter joint to connect piston rod to the crosshead of a double acting steam engine. The diameter of the cylinder is 300 mm and the steam pressure is 1 N/mm^2. The allowable stresses for the material of cotter and piston rod are as follows: σ_t = 50 MPa ; τ = 40 MPa ; and σ_c = 84 MPa 4

OR

Design and draw complete knuckle joints with appropriate design steps.

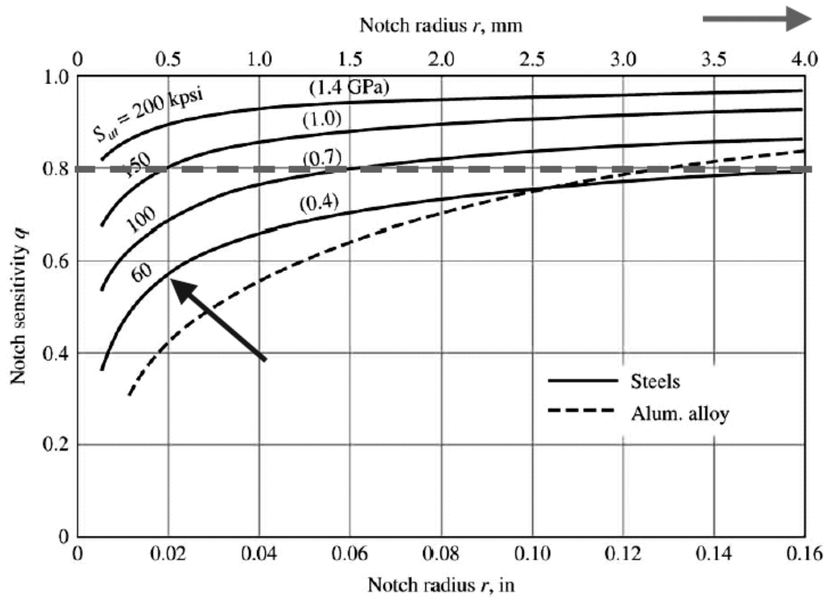
2. A rectangular stepped steel bar is shown in Fig.1. The bar is loaded in bending. Determine the fatigue stress-concentration factor if ultimate stress of the materials is 689 MPa. 2



r = 5mm
D = 50 mm
d = 40 mm
b = 1 mm

Fig. 1

Note: Take Stress- concentration factor $K_t \approx 1.7$ and use following diagram for solving question



3. Explain the following (any two) 4
   (a) Limit, fits and tolerances
   (b) Economic issues of material usage
   (c) Steels Designated on the Basis of Chemical Composition
   (d) General design considerations in forging

Complete detailed solution of DME-I first mid-term test

Solution 1.

Solution. Given : $D = 300$ mm ; $p = 1$ N/mm^2 ; $\sigma_t = 50$ MPa $= 50$ N/mm^2 ; $\tau = 40$ MPa $= 40$ N/mm^2 ; $\sigma_c = 84$ MPa $= 84$ N/mm^2

We know that maximum load on the piston rod,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (300)^2 \, 1 = 70\ 695 \text{ N}$$

The various dimensions for the cotter joint are obtained by considering the different modes of failure as discussed below :

1. *Diameter of piston rod at cotter*

Let d_2 = Diameter of piston rod at cotter, and

t = Thickness of cotter. It may be taken as $0.3\, d_2$.

Considering the failure of piston rod in tension at cotter. We know that load (P),

$$70\ 695 = \left[\frac{\pi}{4} (d_2)^2 - d_2 \times t\right] \sigma_t = \left[\frac{\pi}{4} (d_2)^2 - 0.3\, (d_2)^2\right] 50 = 24.27\, (d_2)^2$$

$\therefore$ $(d_2)^2 = 70\,695 / 24.27 = 2913$ or $d_2 = 53.97$ say 55 mm **Ans.**

and $t = 0.3\, d_2 = 0.3 \times 55 = 16.5$ mm **Ans.**

2. *Width of cotter*

Let b = Width of cotter.

Considering the failure of cotter in shear. Since the cotter is in double shear, therefore load (P),

$$70\ 695 = 2\, b \times t \times \tau = 2\, b \times 16.5 \times 40 = 1320\, b$$

$\therefore$ $b = 70\,695 / 1320 = 53.5$ say 54 mm **Ans.**

3. *Diameter of socket*

Let d_3 = Diameter of socket.

Considering the failure of socket in tension at cotter. We know that load (P),

$$70\ 695 = \left\{\frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] - (d_3 - d_2)\, t\right\} \sigma_1$$

$$= \left\{\frac{\pi}{4}\left[(d_3)^2 - (55)^2\right] - (d_3 - 55)\, 16.5\right\} 50$$

$$= 39.27\, (d_3)^2 - 118\ 792 - 825\, d_3 + 45\ 375$$

or $(d_3)^2 - 21\, d_3 - 3670 = 0$

$$\therefore \quad d_3 = \frac{21 + \sqrt{(21)^2 + 4 \times 3670}}{2} = \frac{21 \pm 123}{2} = 72 \text{ mm} \quad \text{...(Taking + ve sign)}$$

Let us now check the induced crushing stress in the socket. We know that load (P),

$$70\ 695 = (d_3 - d_2)\, t \times \sigma_c = (72 - 55)\, 16.5 \times \sigma_c = 280.5\, \sigma_c$$

$\therefore$ $\sigma_c = 70695 / 280.5 = 252$ N/mm^2

Since the induced crushing is greater than the permissible value of 84 N/mm^2, therefore let us find the value of d_3 by substituting $\sigma_c = 84$ N/mm^2 in the above expression, *i.e.*

$$70\ 695 = (d_3 - 55)\, 16.5 \times 84 = (d_3 - 55)\, 1386$$

$\therefore$ $d_3 - 55 = 70\,695 / 1386 = 51$

or $d_3 = 55 + 51 = 106$ mm **Ans.**

We know the tapered length of the piston rod,

$$L = 2.2\, d_2 = 2.2 \times 55 = 121 \text{ mm} \quad \textbf{Ans.}$$

Assuming the taper of the piston rod as 1 in 20, therefore the diameter of the parallel part of the piston rod,

$$d = d_2 + \frac{L}{2} \times \frac{1}{20} = 55 + \frac{121}{2} \times \frac{1}{20} = 58 \text{ mm} \quad \textbf{Ans.}$$

and diameter of the piston rod at the tapered end,

$$d_1 = d_2 - \frac{L}{2} \times \frac{1}{20} = 55 - \frac{121}{2} \times \frac{1}{20} = 52 \text{ mm} \quad \textbf{Ans.}$$

OR

Knuckle Joint (Pin Joint)

A knuckle joint (as shown in Fig. 1) is used to connect two rods under tensile load. This joint permits angular misalignment of the rods and may take compressive load if it is guided.

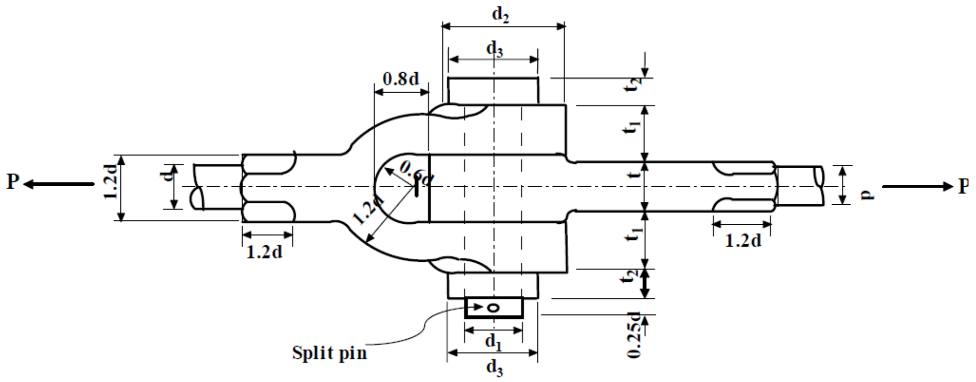


Fig. 1 A typical knuckle joint

These joints are used for different types of connections e.g. tie rods, tension links in bridge structure. In this, one of the rods has an eye at the rod end and the other one is forked with eyes at both the legs. A pin (knuckle pin) is inserted through the rod-end eye and fork-end eyes and is secured by a collar and a split pin. Normally, empirical relations are available to find different dimensions of the joint and they are safe from design point of view. The proportions are given in the Fig. 1.

d = diameter of rod

$d_1 = d$	$t = 1.25d$
$d_2 = 2d$	$t_1 = 0.75d$
$d_3 = 1.5.d$	$t_2 = 0.5d$

Mean diameter of the split pin = 0.25 d

However, failures analysis may be carried out for checking. The analyses are shown below assuming the same materials for the rods and pins and the yield stresses in tension, compression and shear are given by σ_t, σ_c and τ .

S. no.	Failure mode	Design equation
1	Failure of rod in tension:	$\frac{\pi}{4}d^2\sigma_t = P$
2	Failure of knuckle pin in double shear:	$2\frac{\pi}{4}d_1^2\tau = P$
3	Failure of knuckle pin in bending (if the pin is loose in the fork)	$\sigma_t = \frac{16P\left(\frac{t_1}{3}+\frac{t}{4}\right)}{\pi d_1^3}$

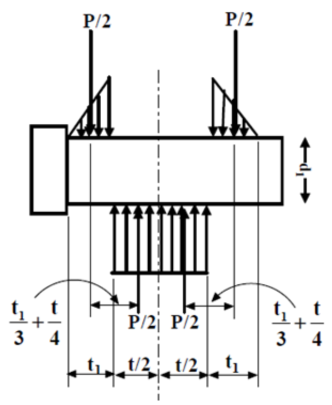


Fig. 2 Bending of a knuckle pin

4	Failure of rod eye in shear:	$(d_2 - d_1)t\tau = P$
5	Failure of rod eye in crushing:	$d_1 t\sigma_c = P$
6	Failure of rod eye in tension:	$(d_2 - d_1)t\sigma_t = P$
7	Failure of forked end in shear:	$2(d_2 - d_1)t_1\tau = P$
8	Failure of forked end in tension:	$2(d_2 - d_1)t_1\sigma_t = P$
9	Failure of forked end in crushing:	$2d_1t_1\sigma_c = P$

The design may be carried out using the empirical proportions and then the analytical relations may be used as checks.

For example using the 2nd equation we have $\tau = \frac{2P}{\pi d_1^2}$. We may now put value of d_1 from empirical relation and then find F.S. = $\frac{\tau_y}{\tau}$ which should be more than one.

Solution 2

From the geometry r/d = 0.125 and D/d = 1.25.

From the stress concentration chart using data handbook or assume Stress-concentration factor as $K_t \approx 1.7$

From given chart

Notch sensitivity index, $q \approx 0.88$

The fatigue stress concentration factor K_f is now given by

$K_f = 1+q\ (K_t - 1) = 1.616$

Solution 3

(a) *The extreme permissible values of a dimension are known as limits. The degree of tightness or looseness between two mating parts that are intended to act together is known as the fit of the parts. The character of the fit depends upon the use of the parts. Thus, the fit between members that move or rotate relative to each other, such as a shaft rotating in a bearing, is considerably different from the fit that is designed to prevent any relative motion between two parts, such as a wheel attached to an axle'.*

Tolerance on the other hand is the total amount that a specific dimension is permitted to vary. It is the difference between the maximum and the minimum limits for the dimension.

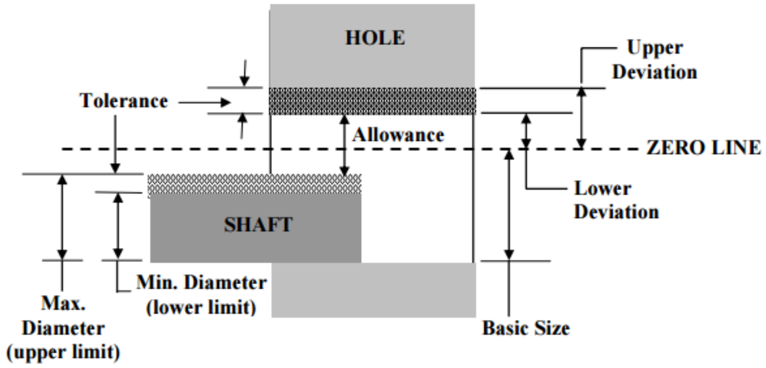


Fig. 1

Fig. 1 explains the terminologies used in defining tolerance and limit.

Clearance Fit: In this type of fit, the shaft of largest possible diameter can also be fitted easily even in the hole of smallest possible diameter.
Transition Fit: In this case, there will be a clearance between the minimum dimension of the shaft and the minimum dimension of the hole. If we look at the figure carefully, then it is observed that if the s haft dimension is maximum and the hole dimension is minimum then an overlap will result and this creates a certain amount of tightness in the fitting of the shaft inside the hole. Hence, transition fit may have either clearance or overlap in the fit.
Interference Fit: In this case, no matter whatever may be the tolerance level in shaft and the hole, there is always an overlapping of the matting parts. This is known as interference fit. Interference fit is a form of a tight fit.

(b) Economic issues of material usage

Engineering materials are important in everyday life because of their versatile structural and physical properties. Selection of the engineering materials is dependent on these properties. However, choice of material has implications throughout the life-cycle of a product, influencing many aspects of economic and environmental performance. In other words, different material issues are needed to be dealt with during material selection if the product is to be commercially successful and competitive in the market. These special considerations are related to their cost, production cost, environmental and social issues like pollution, disposal, recycling, etc.
Engineering profession deals with utilization of scientific and technological advances to design and manufacture components and systems that perform reliably and satisfactorily. However, there are economics as the driving force behind it. Economics of engineering a component / system depends on three factors: component design, material usage, and manufacturing costs. These three factors are inter-related in the sense that one or two might influence the choice of others.
Initially any component / system need to go through conception and then design stage. This includes generation of concept about the component / system. Later on design stage takes care of its size, shape, and configuration which will influence the performance of it during the service. Usually engineers deal with not a single component, but with complex assembly of components / a system. Thus, each component needs to be designed for greater efficiency of the system. This may sometimes act as constraint to optimal design of a component. Hence, design of a component is usually an iterative process. Less the number of iterations, lowe r will be the cost of the component / system.
Design stage is followed by material selection. Material is selected depending on its properties, which are suitable to serve the purpose during the service. Other than the properties, cheaper materials are pr eferred, if choice is available. Thus, usually a family of materials is selected that satisfy the design constraints, then comparisons are made on the basis of cost per unit. This cost also includes unavoidable material wastage during manufacturing stage.
After design and material selection, it is up to manufacturing method to reduce the product cost. Usually manufacturing includes both primary stage and secondary stage. The cost considerations include the capital on tooling, maintenance cost, labour, repair costs, and material wastage. More the number of manufacturing stages, higher will be the product cost. Inspection, assembly, and final packaging will add-on to the product cost.

(c) Steels Designated on the Basis of Chemical Composition

According to Indian standard, IS: 1570 (Part II/Sec I)-1979 (Reaffirmed 1991), the carbon steels are designated in the following order:
(*1*) Figure indicating 100 times the average percentage of carbon content,
(*2*) Letter 'C', and
(*3*) Figure indicating 10 times the average percentage of manganese content. The figure after multiplying shall be rounded off to the nearest integer.
For example 20C8 means a carbon steel containing 0.15 to 0.25 per cent (0.2 per cent on average) carbon and 0.60 to 0.90 per cent (0.75 per cent rounded off to 0.8 per cent on an average) manganese.

(d) GENERAL DESIGN CONSIDERATIONS IN FORGING

1. An expensive operation, if possible avoids using forging.
2. Select materials that are relatively easy to deform.
3. Part shapes that provide smooth and easy external flow paths are desirable.
4. For ease of manufacture, ribs should be widely spaced.
5. Avoid internal undercuts and external undercuts caused by projections