COARSE FILE IN MACHINE DESIGN

FOR 3RD YEAR STUDENTS

PREPARED BY

NAWZAD J. MAHMOOD

ASSISTANT LECTURER

REF. & AIR COND. ENGINEERING DEPARTMENT

MINISTRY OF HIGHER EDUCATION AND SCIENTIFIC RESEARCH

FOUNDATION OF TECHNICAL EDUCATION

TECHNICAL COLLEGE/KIRKUK

REF. &AIR CONDITIONING ENGINEERING DEPARTMENT

Subject	Year of student	Hours In Week			Units
		Theory	Practical	Total	
Machine Design	3rd Year Students	2	3	5	6

The project:-

The student can be able to:

1-Define the basic principles to design different mechanical parts.

2-Study variable loads and thermal stresses to design mechanical instruments and equipment's.

ITEM	WEEK	SYLLABUS
1	1-2	METALS &SIMPLE STRESSES
2	3-4	VARIABLE LOADS &STRESS CONCENTRATIONS
3	5-7	BOLTS,RIVETS AND WELDED JOINTS
4	8-9	PRIMARY LOADS IN SCREWS
5	10-11	POWER SCREW DESIGN
6	12-14	SHAFT DESIGN
7	15	KEY WAYS AND SPLINED SHAFTS
8	16	COUPLINGS
9	17	BELT AND CHAIN DRIVE DESIGN
10	18-20	ROLLING BEARINGS
11	21-22	SLIDING BEARINGS
12	23-24	SPRINGS
13	25-26	PRESSURE VESSELS
14	27-28	STSTIC &DYNAMIC SEALS
15	29-30	SPUR GEAR DESIGN

REFRENCES

1- MACHINE DESIGN, KHURMI 1956-2005 –INDIA

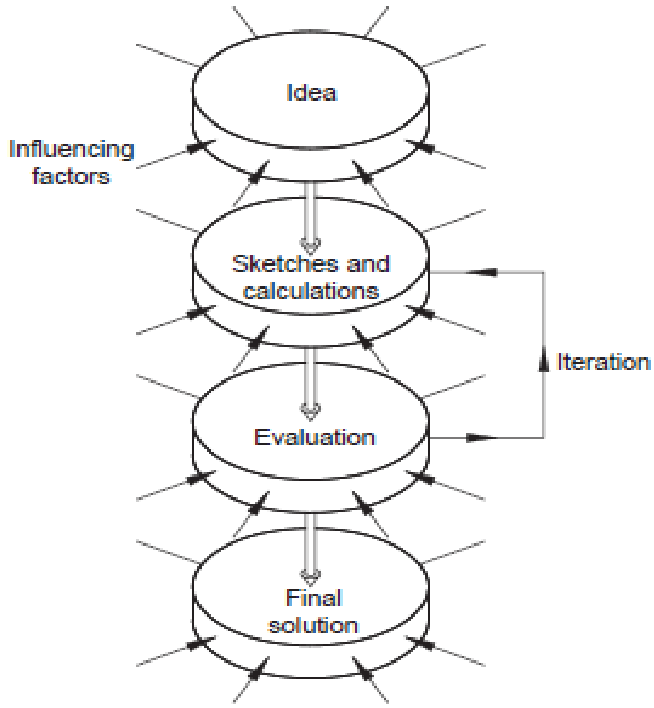
2- MECHANICAL DESIGNS, PETER R.CHILDS, UK.

Tutorial part

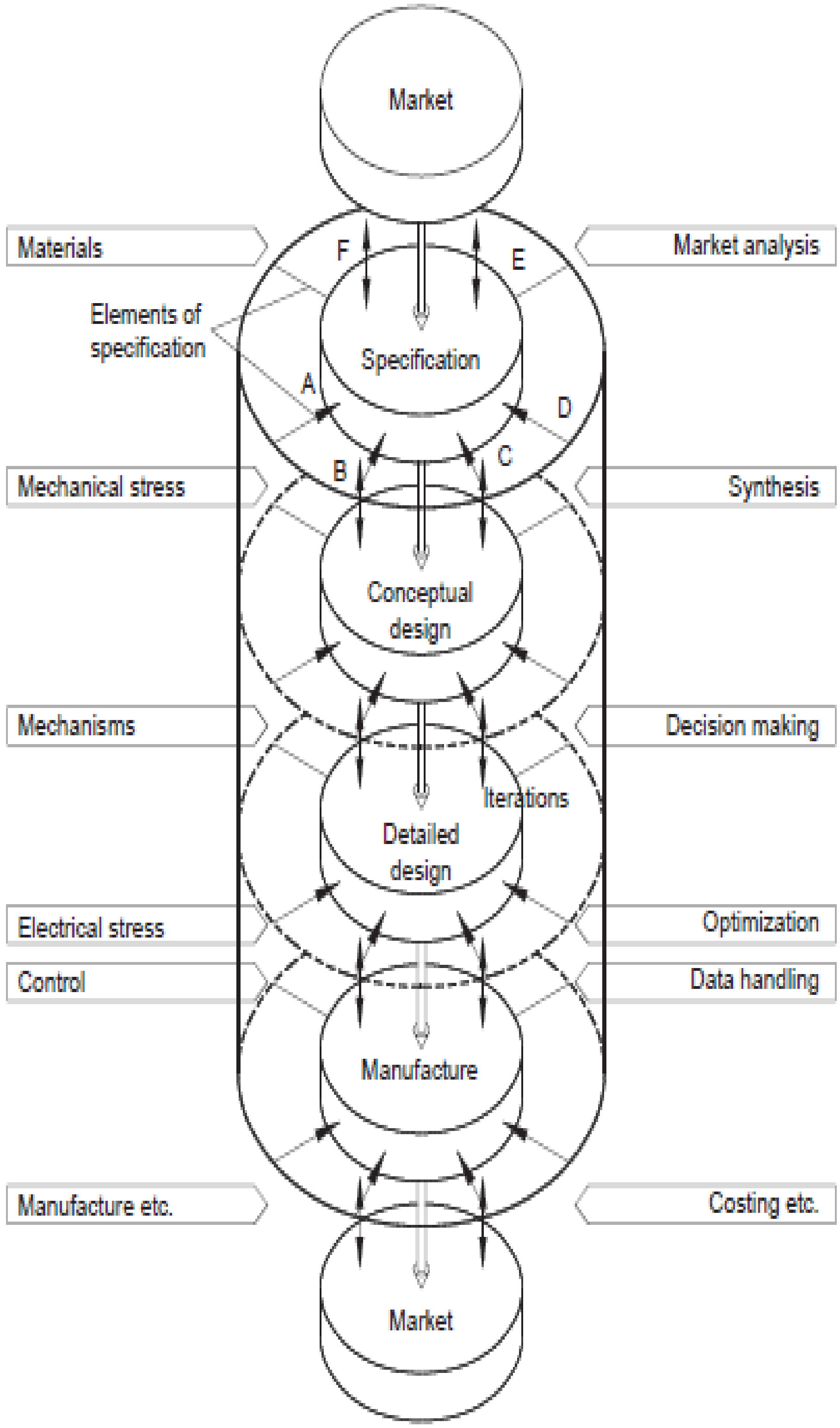
week	Subject
1-15	**Design-CAD and specified Design software**

Lecture 1

Machine design is defined as :-
The total activity necessary to product or process to meet a market need.



The traditional and familiar 'inventor's' approach to design.



The total design process (after Pugh, 1990).

SUFFIXES

a	axial
b	bending
c	compressive
f	endurance
s	strength properties of material
t	tensile
u	ultimate
y	yield

ABBREVIATIONS

AISI	American Iron and Steel Institute
ASA	American Standards Association
AMS	Aerospace Materials Specifications
ASM	American Society for Metals
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing Materials
BIS	Bureau of Indian Standards
BSS	British Standard Specifications
DIN	Deutsches Institut für Normung
ISO	International Standards Organization
SAE	Society of Automotive Engineers
UNS	Unified Numbering system

COMBINED LOADINGS

Plane Stress Element. The geometry of a differential plane stress element is shown in Fig. 4.1, where the dimensions (Δx) and (Δy) are such that the stresses, whether normal (σ) or shear (τ), can be considered constant over the cross-sectional areas of the edges

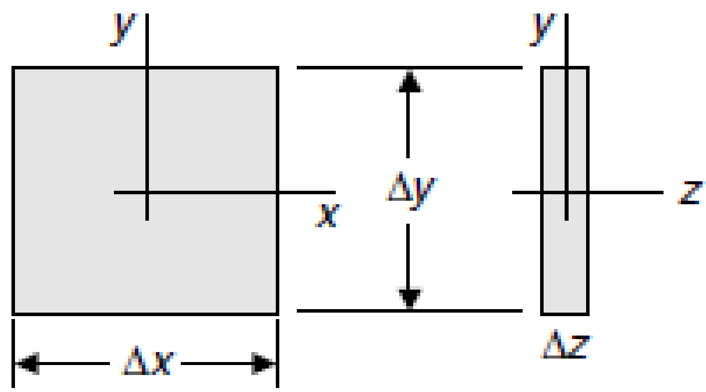


FIGURE 4.1 Geometry of a plane stress element.

Uniaxial Stress Element. For the fundamental loadings of axial, thermal, and bending, a uniaxial stress element is produced and shown in Fig. 4.3,

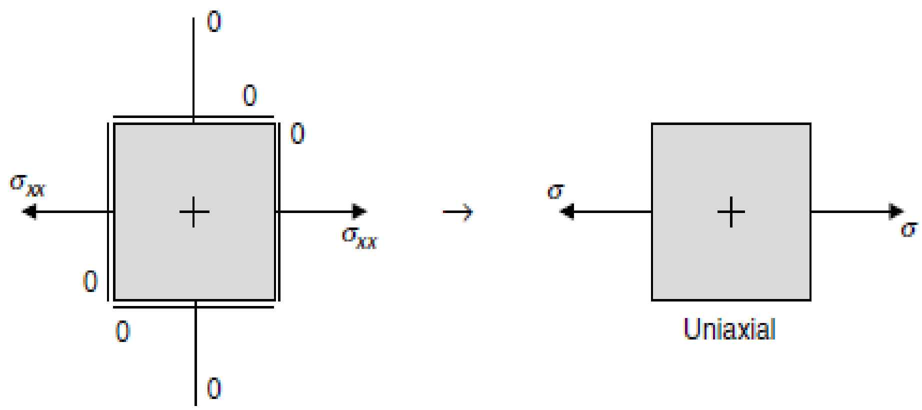


FIGURE 4.3 Uniaxial stress element.

where there is only normal stress (σ) along the axis of interest; and the other stresses, the normal stress (σ_{yy}) and the four shear stresses (τ_{xy}), are zero.

4.2 AXIAL AND TORSION

The first combination of loadings to be considered is axial and torsion. This is a very common loading for shafts carrying both a torque (T) and an end load (P), as shown in Fig. 4.6.

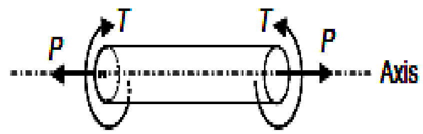


FIGURE 4.6 Axial and torsion loading.

Stress Element. The general stress element shown in Fig. 4.2 becomes the stress element shown in Fig. 4.7, where the normal stress (σ_{xx}) is the axial stress, the normal stress (σ_{yy}) is zero, and the shear stress (τ_{xy}) is the shear stress due to torsion.

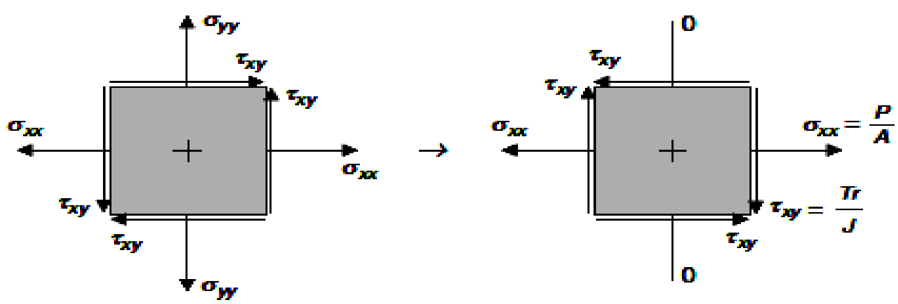


FIGURE 4.7 Stress element for axial and torsion.

The shear stress due to torsion (τ_{xy}) is shown downward on the right edge of the stress element because the torque (T) shown in Fig. 4.6 is counterclockwise looking in from the right side to the left side.

AXIAL AND BENDING

The second combination of loadings to be considered is axial and bending. This is a somewhat common loading for structural elements constrained axially. Shown in Fig. 4.10 is a simply-supported beam with a concentrated force (F) at its midpoint, and a compressive axial load (P).

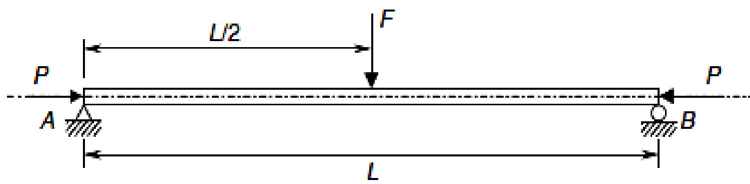


FIGURE 4.10 Axial and bending loads.

In this section, the bending moment (M) and shear force (V) are assumed to be known for whatever beam and loading is of interest. (See Chap. 2 on *Beams*.)

Stress Element. The general stress element shown in Fig. 4.2 becomes the stress element shown in Fig. 4.11, where the normal stress (σ_{xx}) is a combination of the axial stress and bending stress, the normal stress (σ_{yy}) is zero, and the shear stress (τ_{xy}) is the shear stress due to bending.

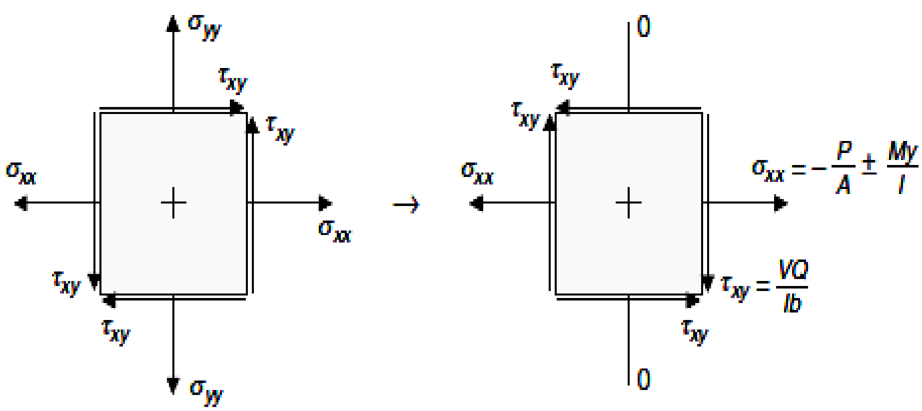


FIGURE 4.11 Stress element for axial and bending loads.

where $Q = A\bar{y}$, first moment of area (A)

A = area out beyond point of interest, specified by distance (y)

$\bar{y}$ = distance to centroid of area (A) defined above

I = area moment of inertia about an axis passing through neutral axis

b = width of beam at point of interest

AXIAL AND THERMAL

The third combination of loading to be considered is an axial load and a thermal load. This type of loading can occur when a machine element is put under a tensile, or compressive, preload during assembly in a factory environment, then subjected to an additional thermal load either due to a temperature drop in the winter or a temperature rise in the summer. Recall that if the machine element is not constrained, then under a temperature change the element merely gets longer or shorter and no stress is developed.

Figure 4.14 shows a thin-walled pipe, or tube, with flanges constrained between two fixed supports. (Note that typically pipe designations are based on inside diameter, whereas tubing is based on outside diameter.) Suppose that the original length of the pipe was shorter than the distance between the supports so that a tensile preload is developed in the pipe when it is installed. Also, suppose that what is of interest is the additional load that will be produced when the pipe is subjected to a temperature drop during the winter.

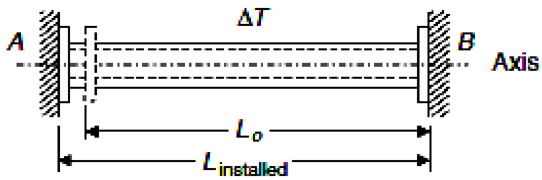


FIGURE 4.14 Axial and thermal loading.

The axial stress due to the lengthening of the pipe during installation is given by Eq. where the axial strain (ε) is multiplied by the modulus of elasticity (E).

$$\sigma_{\text{axial}} = E\,\varepsilon_{\text{axial}} = E\left(\frac{\Delta L}{L}\right) = E\left(\frac{L_{\text{installed}} - L_o}{L_o}\right)$$

The thermal stress due to a temperature drop (ΔT) is given by Eq. (4.6) where the thermal strain (ε_T) is multiplied by the modulus of elasticity (E)

$$\sigma_{\text{thermal}} = E\varepsilon_T = E\alpha\,(\Delta T) \tag{4.2}$$

and (α) is the coefficient of thermal expansion of the pipe.

Combining these two normal stresses, both of which are constant over the cross section of the pipe, gives the single stress (σ_{xx}) shown in Eq. (4.3),

$$\sigma_{xx} = \sigma_{\text{axial}} + \sigma_{\text{thermal}} = E\varepsilon_{\text{axial}} + E\varepsilon_T = E\left[\frac{\Delta L}{L} + \alpha(\Delta T)\right] \tag{4.3}$$

where

$$\frac{\Delta L}{L} = \frac{L_{\text{installed}} - L_o}{L_o} \tag{4.4}$$

Stress Concentration Factors and Notch Sensitivity

Lecture 2

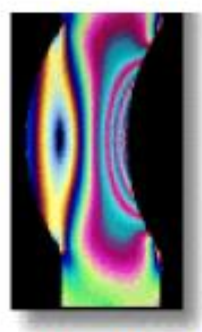
Machine Design

Radiometric Thermoelasticity

Automobile Connecting Rod | Hook and Clevis | Crack Tip

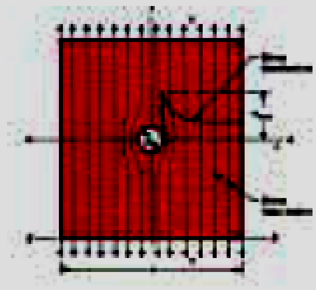
When materials are stressed the change in atomic spacing creates temperature differences in the material. Cameras which sense differences in temperature can be used to display the stress field in special materials.

Photoelasticity
(Continued)



When a photoelastic material is strained and viewed with a polariscope, distinctive colored fringe patterns are seen. Interpretation of the pattern reveals the overall strain distribution.

Geometric Stress Concentration Factors



$$K_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

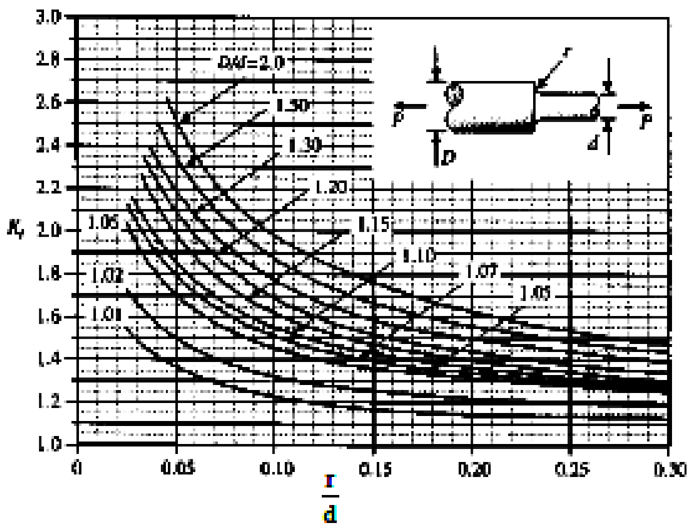
$$\sigma_{nom} = \frac{F}{A_0}$$

$$A_0 = (w - d)t$$

Geometric stress concentration factors can be used to estimate the stress amplification in the vicinity of a geometric discontinuity.

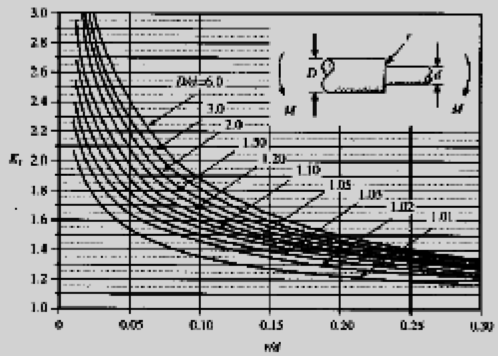
Geometric Stress Concentration Factors

(Tension Example)



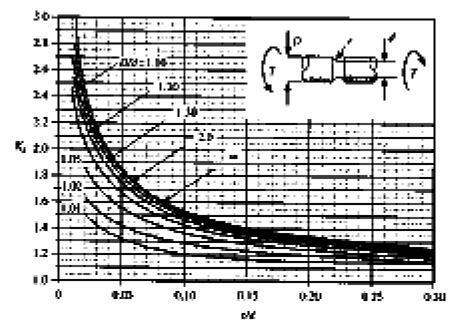
Geometric Stress Concentration Factors

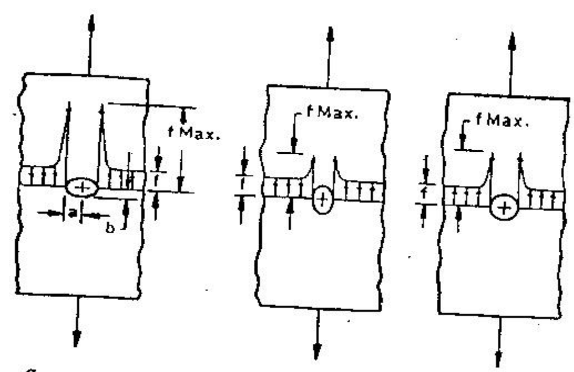
(Bending Example)



Geometric Stress Concentration Factors

(Torsion Example)





$\frac{a}{b} = 2$ $\frac{a}{b} = \frac{1}{2}$ $\frac{a}{b} = 1$

$\sigma_{max} = f_{max} = 5f$ (a) $\sigma = f_{max} = 2f$ (b) $\sigma = f_{max} = 3f$ (c)

The maximum stress is given by:

$$\sigma_{max} = \sigma_0 (1 + 2\frac{a}{b})$$

where :

σ_{max} : Max. Stress. (N/m^2)

σ_0 : Nominal stress. (N/m^2)

a & B : radius. (mm)

Or theoretical stress concentration factor

$$Kt = \frac{\sigma_{max}}{\sigma_0} = (1 + 2\frac{a}{b})$$

Kt : Stress concentration factor .

1. when $\frac{a}{b}$ is large

$\frac{a}{b} = 2 \quad \therefore \quad \sigma_{max} = 5\sigma_0$

2. when $\frac{a}{b}$ is small

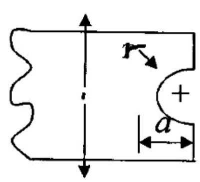
$\frac{a}{b} = \frac{1}{2} \quad \therefore \quad \sigma_{max} = 2\sigma_0.$

3. when $\frac{a}{b} = 1$

$\therefore \quad \sigma_{max} = 3\sigma_0.$

Stress concentration in notched tension member

$$\sigma_{max} = \sigma_0 (1 + \frac{a}{r})$$

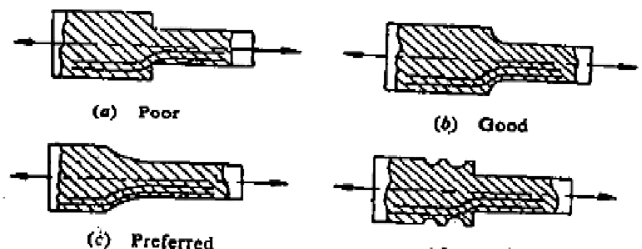


Where

a: notch depth

r: radius at the bottom of the notch.

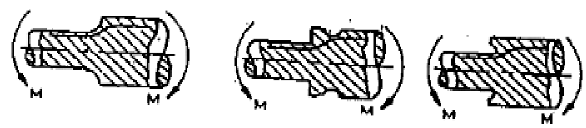
*Methods for reducing stress concentration



(a) Poor (b) Good (c) Preferred (d) Preferred

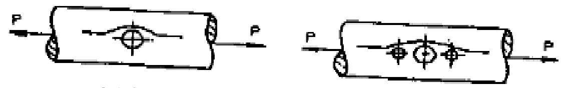
Fig. 6·8

In Fig. 6·8 (a) we see that the stress lines tend to bunch up and cut very close to the sharp reentrant corner. In order to improve the situation, fillets may be provided, as shown in Fig. 6·8 (b), and (c) to give more equally spaced flow lines.



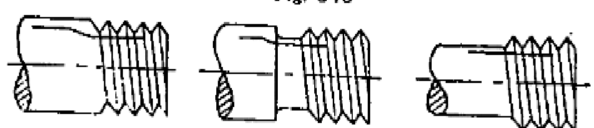
(a) Poor (b) Good (c) Preferred

Fig. 6·9



(a) Poor (b) Preferred

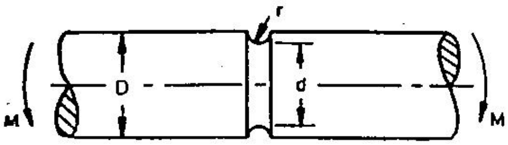
Fig. 6·10



(a) Poor (b) Good (c) Preferred

Fig. 6·11

Theoretical stress concentration factor (K_t) for a grooved shaft in **bending**



$$Z = \frac{\pi d^3}{32}$$

$\frac{D}{d}$	Theoretical stress concentration factor (K_t)									
	r/d									
	0.02	0·04	0·08	0·10	0·12	0·16	0·20	0·24	0·28	0·30
1·01	1·74	1·68	1·47	1·41	1·38	1·32	1·27	1·23	1·22	1·20
1·02	2·28	1·89	1·60	1·53	1·48	1·40	1·34	1·30	1·26	1·25
1·03	2·46	2·04	1·68	1·61	1·55	1·47	1·40	1·35	1·31	1·28
1·05	2·75	2·22	1·80	1·70	1·63	1·53	1·46	1·40	1·35	1·33
1·12	3·20	2·50	1·97	1·83	1·75	1·62	1·52	1·45	1·38	1·34
1·30	3·40	2·70	2·04	1·91	1·82	1·67	1·57	1·48	1·42	1·38
1·50	3·48	2·74	2·11	1·95	1·84	1·69	1·58	1·49	1·43	1·40
2·00	3·55	2·78	2·14	1·97	1·86	1·71	1·59	1·50	1·44	1·41
∞	3·60	2·85	2·17	1·98	1·88	1·71	1·60	1·51	1·45	1.42

Table 6·8

Example **6·1**. ***Find the** maximum stress induced in the **following** cases taking stress **concentration** into account.*

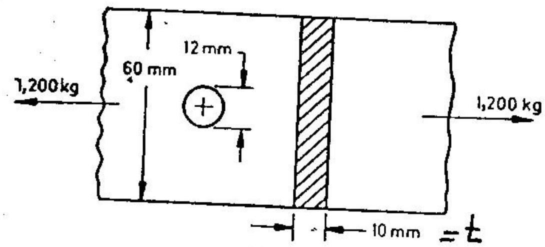
*(1) A **rectangular plate** 60 mm × 10 mm with a hole **12 mm** diameter as **shown in Fig** 6·13 (a) and subjected to a tensile **load of** 1,200 kg.*

*(2) **A stepped shaft** as shown in Fig. 6·13 (b) and carrying a tensile load of 1,200 kg.* *(Oxford University,)*

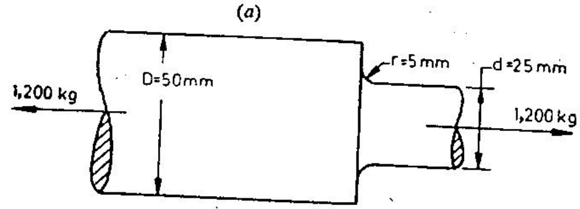
Solution.

Case 1

Given. Width of the plate, $b = 60$ mm
Thickness of the plate, $t = 10$ mm
Diameter of the hole, $d = 12$ mm
Tensile load, $W = 1{,}200$ kg



(a)



(b)

We know that cross-sectional area of the plate,

$$A = (b-d)t$$
$$= (60-12)10 = 480 \text{ mm}^2$$

$\therefore$ Nominal stress $= \frac{W}{A} = \frac{1{,}200}{480} = 2.5 \text{ kg/mm}^2$

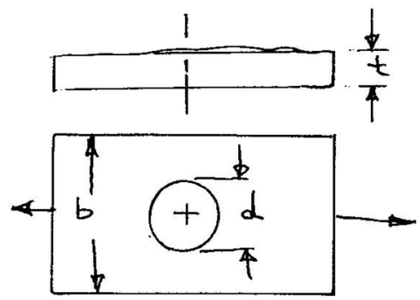
Ratio of diameter of hole to width of plate,

$$\frac{d}{b} = \frac{12}{60} = 0.2$$

From Table ~~5.1~~ 6.1, we find that theoretical stress concentration factor,

$$K_t = 2.5$$

$\therefore$ Maximum stress $= K_t \times \text{Nominal stress}$
$= 2.5 \times 2.5 = 6.25 \text{ kg/mm}^2$ **Ans.**



$A = (b-d)t$

Case 2

Given. Maximum diameter of the shaft,

$$D = 50 \text{ mm}$$

Table 6.1

$\frac{d}{b}$	0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55
K_t	2.83	2.69	2.59	2.5	2.43	2.37	2.32	2.26	2.22	2.17	2.13

Minimum diameter of the shaft,

$$d = 25 \text{ mm}$$

Radius of fillet, $r = 5$ mm

Tensile load, $W = 1{,}200$ kg

We know that cross-sectional area for the stepped shaft,

$$A = \frac{\pi}{4} d^2$$

$$= \frac{\pi}{4} \times 25^2 = 491 \text{ mm}^2$$

$\therefore$ Nominal stress

$$= \frac{W}{A}$$

$$= \frac{1{,}200}{491} = 2 \cdot 4 \text{ kg/mm}^2$$

Ratio of maximum diameter to minimum diameter,

$$\frac{D}{d} = \frac{50}{25} = 2$$

Ratio of radius of fillet to minimum diameter,

$$\frac{r}{d} = \frac{5}{25} = 0 \cdot 2$$

From Table 5·3, we find that theoretical stress concentration factor, $K_t = 1 \cdot 64$

$\therefore$ Maximum stress $= K_t \times$ Nominal stress

$$= 1 \cdot 64 \times 2 \cdot 4 = 3 \cdot 94 \text{ kg/mm}^2$$ **Ans.**