ME18503 DESIGN OF MACHINE ELEMENTS

OBJECTIVES

- This course will impart the knowledge on various types of stress- selection of materials
- This course will make acquainted design principles on shaft, fits and tolerances.
- This course will familiarize the design principles of springs under dynamic and static conditions.
- This course will enable to check strength of fasteners kind of both rivet and welding.
- This course will facilitate to select and examine the rolling and sliding contact bearings.

OUTCOMES

1. Able to depict the design process, material selection, calculation of stresses under static and variable loading conditions with the effect of stress concentrations.

2. Adept the design of solid, hollow shafts keys and couplings. Also having knowledge of fits and tolerance.

3. Examining the close coil helical springs under variable loading . Acquiring the knowledge of leaf, disc and torsion springs.

4. Proficient in Design of riveted joint and welding joints under eccentric loading.

5. Have design knowledge on sliding and rolling contact bearing. EHD Journal Bearing

TEXT BOOKS:

1. Bhandari,V.B ,"Design of Machine Element", 3rd edition, TMH Publications.New Delhi,2017.
2. Khurmi R.S., and Gupta J.K., "*Machine Design*", 14 th Edition,S Chand&Co NewDelhi,2005.
3. Sundararajamoorthy, T.V. and Shanmugam, N., "*Machine Design*", Anuradha Agencies, Chennai,2003.

REFERENCES :

1. Bhandari V.B., "*Design of Machine Elements*", 4 th edition TATA McGraw HillNew delhi,2017.
2. Khurmi R.S., and Gupta J.K., "*Machine Design"*, 14th Edition, S Chand and Co NewDelhi,2005.
3. Lingaiah K., "*Machine Design Data Book"*, 2 nd Edition Tata McGraw – Hill.New delhi,2010.
4. Robort.L.Norton," Machine Design", 5 th Edition, Pearson Publisher, New delhi,2018.
5. Sharma,P.C and Agarwal,D.K,"MachineDesign",Agrawal-Kataria and Sons Publication, NewDelhi,2014.
6. Shigley, J.E., Charles, R.M. and Richard, G.B., "*Mechanical Engineering Design", 10 th* ed., McGraw-Hill, New Delhi,2014.
7. Spotts M.F., "*Design of Machine Elements*", 8 th Edition,Pearson Education,NewDelhi, 2019.

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GENERAL DISCUSSION

PRE REQUISITE

Knowledge on

1. Strength of Material
2. Theory of Machines
3. Materials science/Metallurgy

SOME Questions for you

STRESS?	σ = load/Area N/mm^2
STRAIN?	ε = change in dimension/Original dimension no units
TYPES of STRESSES	Tensile stress , compressive stress ,shear stress
Principle stress	Stress acting on principle plane
Principle plane	Plane in which shear is zero

Types of Loads

1. Point load
2. UD load
3. UV load

Others

1. **Dead load**
2. **Transverse load**
3. **Axial load**
4. **Tangential load**

Name the load

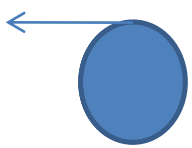
Point load --- transverse load

Name the load

Axial load

Name the load

Tangential load



Last session 22- 06- 2020 -discussed

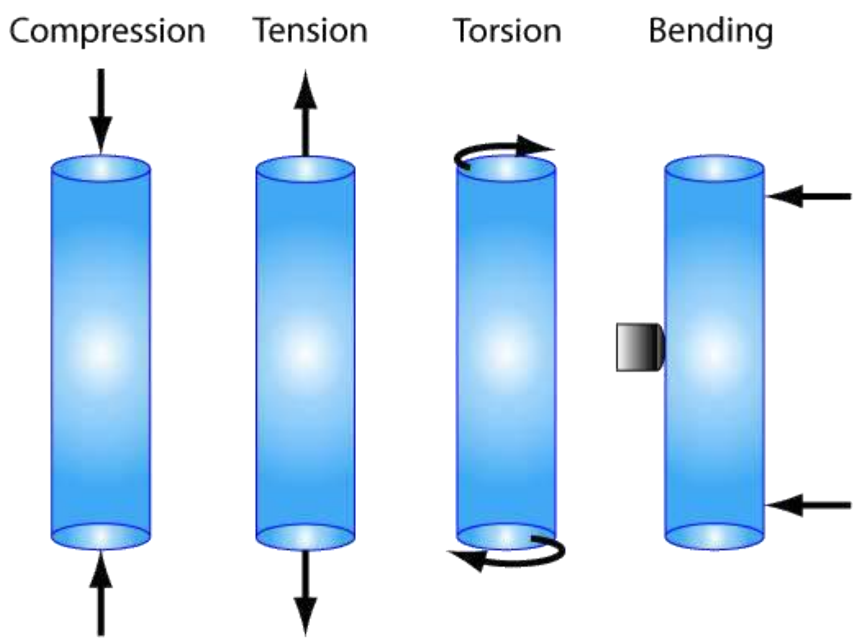
Objectives

Outcomes

types of simple stresses

Types of loads

Diagrams for different type of simple stresses



σ_t τ σ_b

σ_c

What is design?

Creating/innovate of an idea

Modify the existing

With the application of scientific principles and mathematical approaches

Types of Design

1. Adaptive design

2. Modifying design

3. New design--/ -- Innovation

Examples to understand the types of Design

Electric fan --- with 3 blades → adaptive

Electric fan---- with 4 blades → Modified design

Discussion –forum for some examples towards the types of design

Assignment to find the other types of design

Industrial design,
Experimental design,
Aesthetic,
Ergnomics design

Syllabus

UNIT I
BASICS OF DESIGN PRINCIPLES

UNIT II
DESIGN SHAFT, KEYS AND FITS AND TOLERANCE AND COUPLINGS

UNIT III
DESIGN FOR SPRINGS

UNIT IV
DESIGN FOR RIVETED AND WELDING JOINTS, FASTNERS

UNIT V
DESIGN OF BEARINGS

UNIT I BASICS OF DESIGN PRINCIPLES

Design Process-

Types of Stress,

Cyclic stresses ,

Factor of Safety,

Stress concentration factor in tension, bending and torsion,

Theories of failures.

Notch sensitivity,

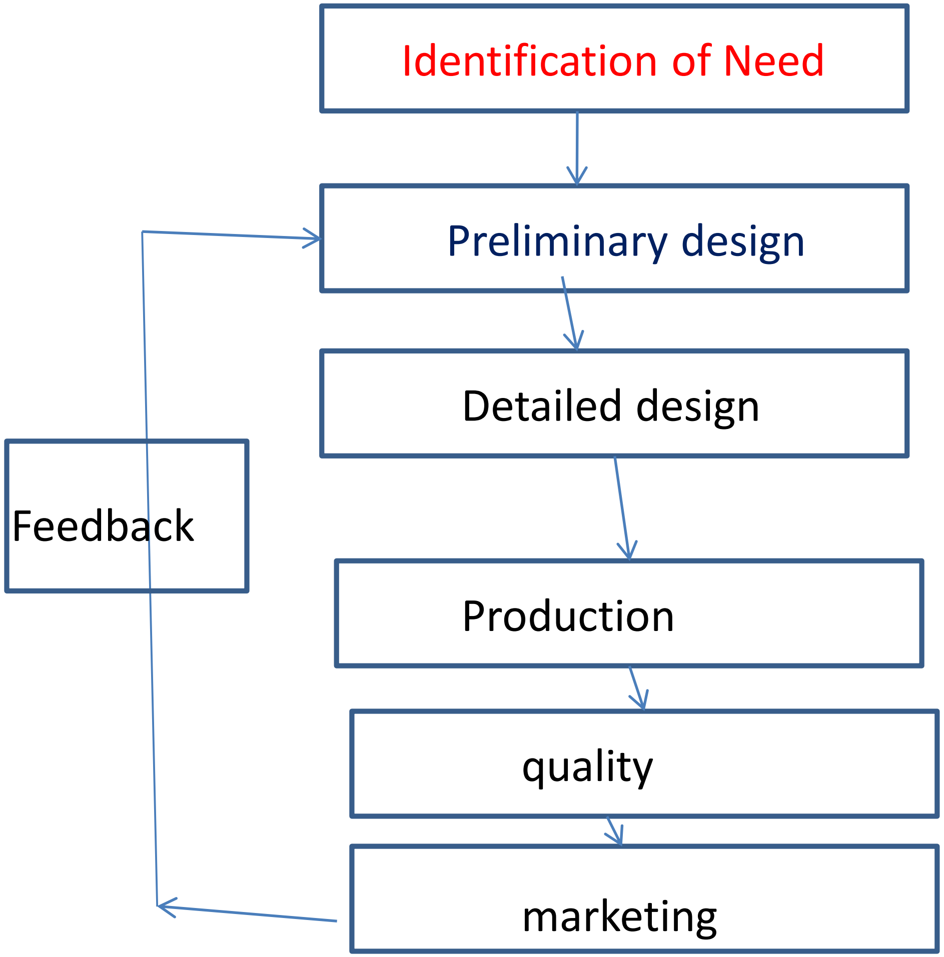
Design for variable and repeated loadings,

Fatigue stress concentration factor,

Endurance diagrams,

Introduction to fracture mechanics

What is Design process?



It is also known as design cycle

Mechanical properties

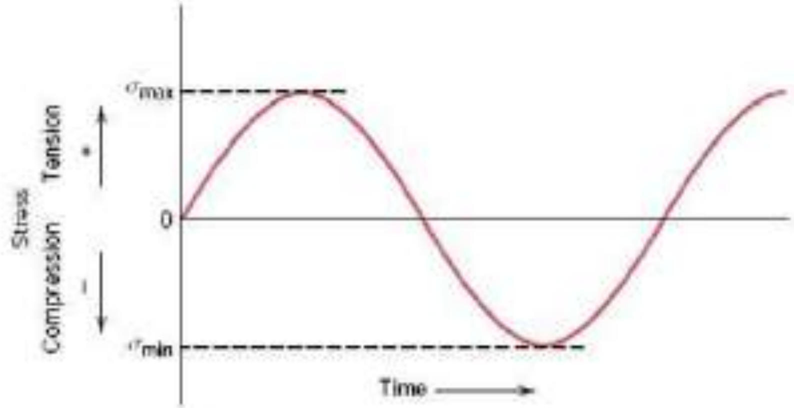
Materials → Ductile material → Elastic limit, yield, Plastic ten break

Materials → Brittle Material → Only break at ultimate load

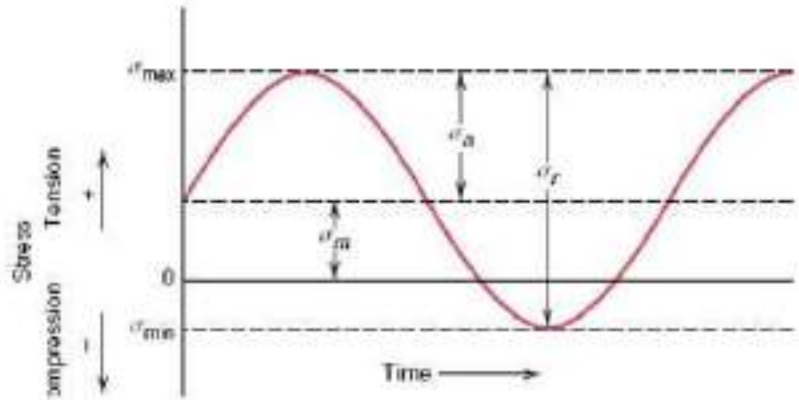
Learn some mechanical properties

- strength
- hardness
- Plasticity
- Elasticity
- Resilience
- Toughness
- Creep

Cyclic stress



Reversed stress cycle

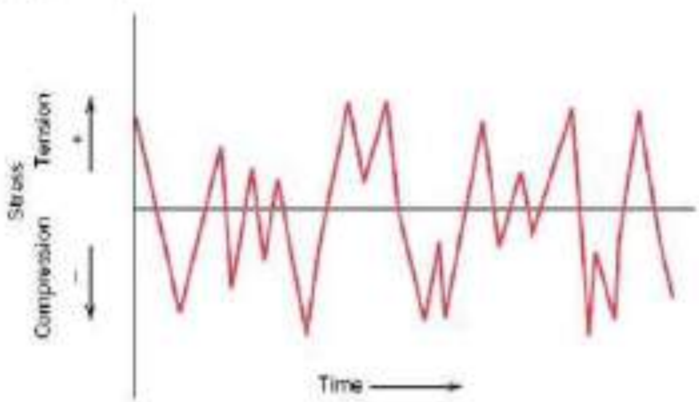


Repeated stress cycle

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

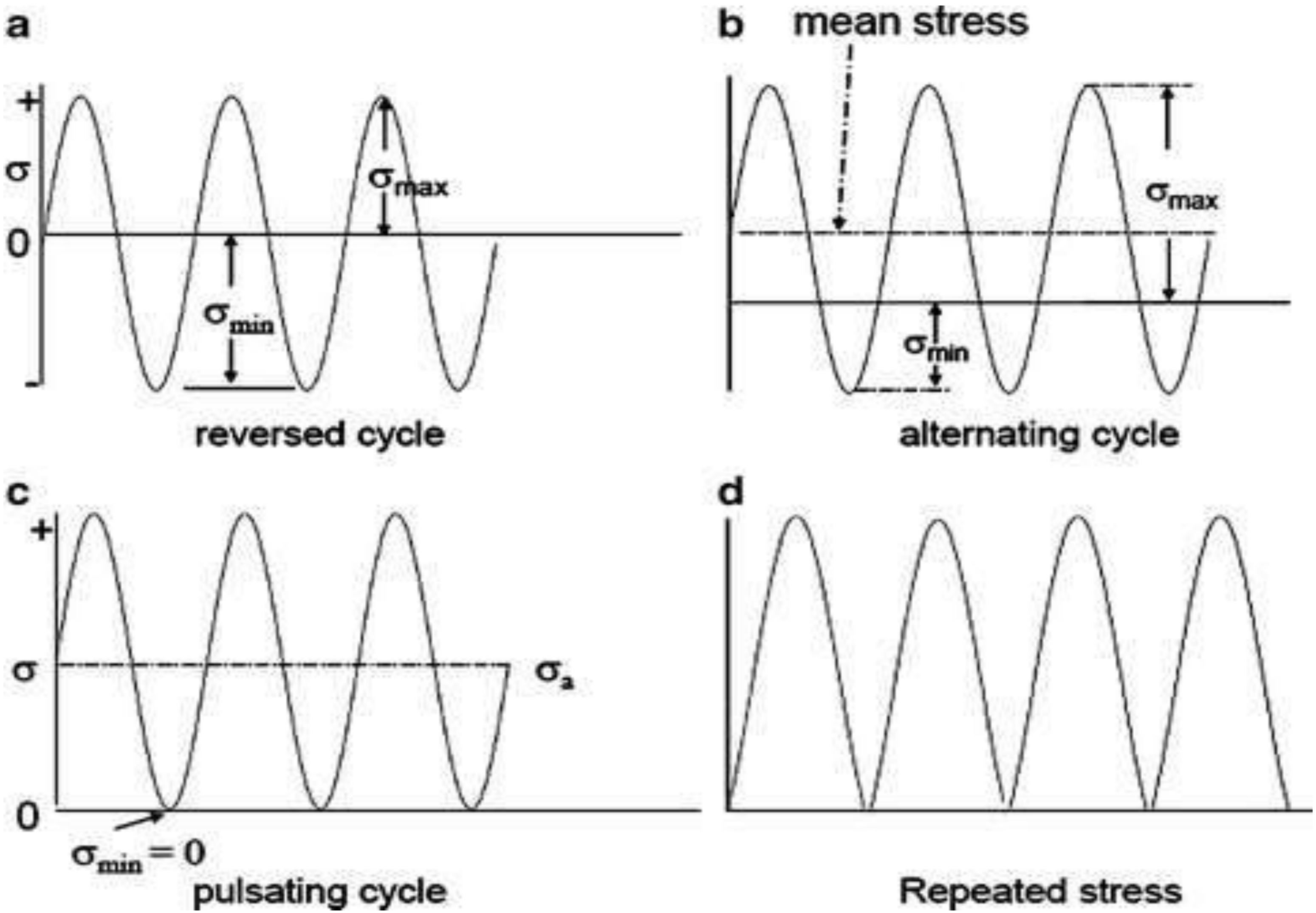
$$\sigma_r = \sigma_{max} - \sigma_{min}$$

$$\sigma_a = \frac{\sigma_r}{2} = \frac{\sigma_{max} - \sigma_{min}}{2}$$



Random stress cycle

Cyclic stresses



Last session topics

Types of simple stresses,

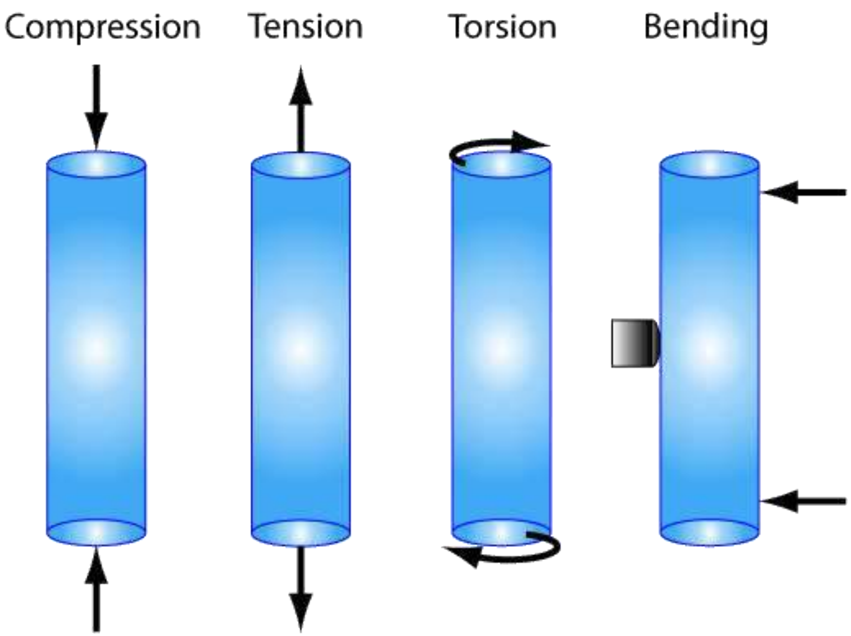
cyclic stresses

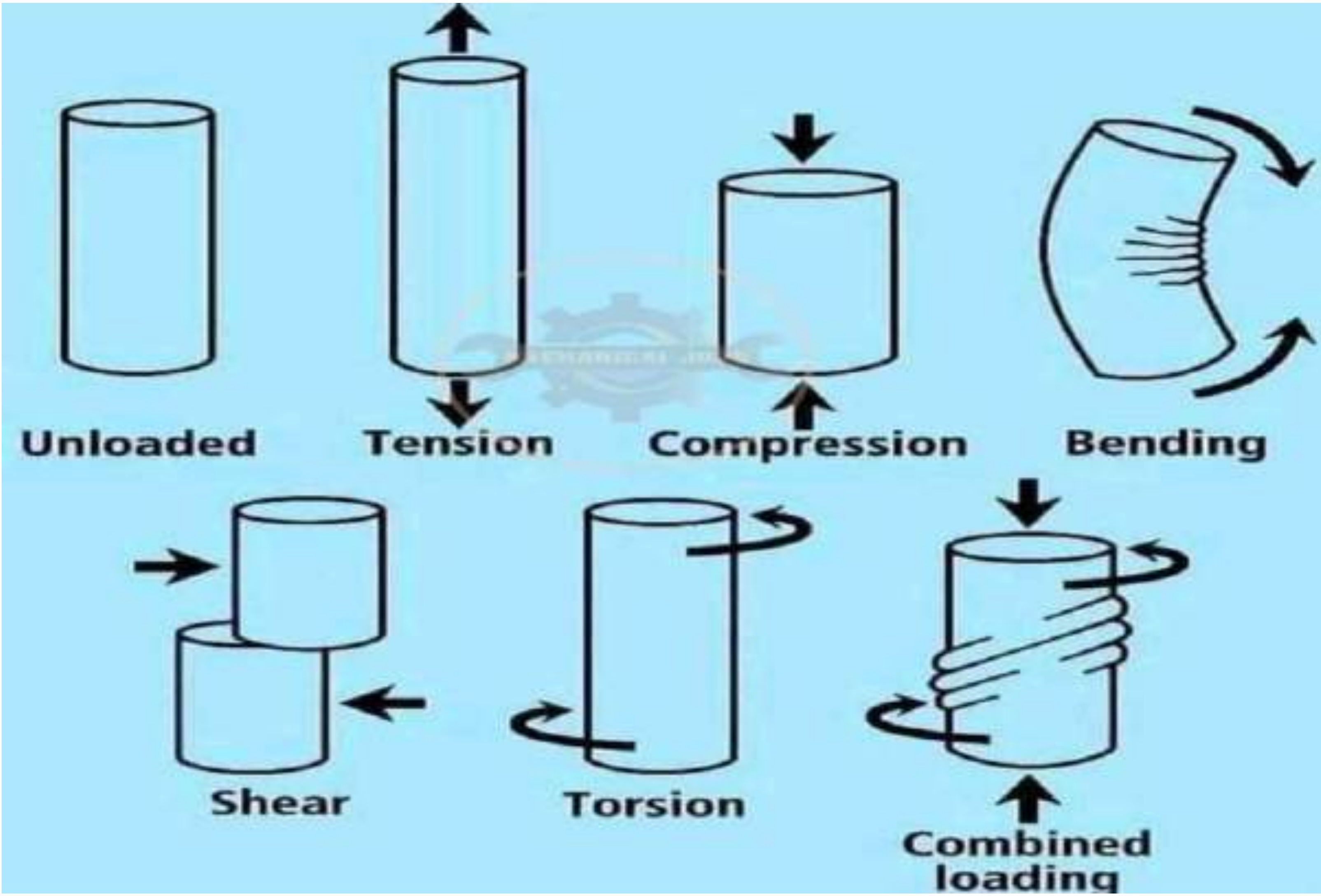
Design processes/cycle

Types of design – adaptive, modified , & new design and others

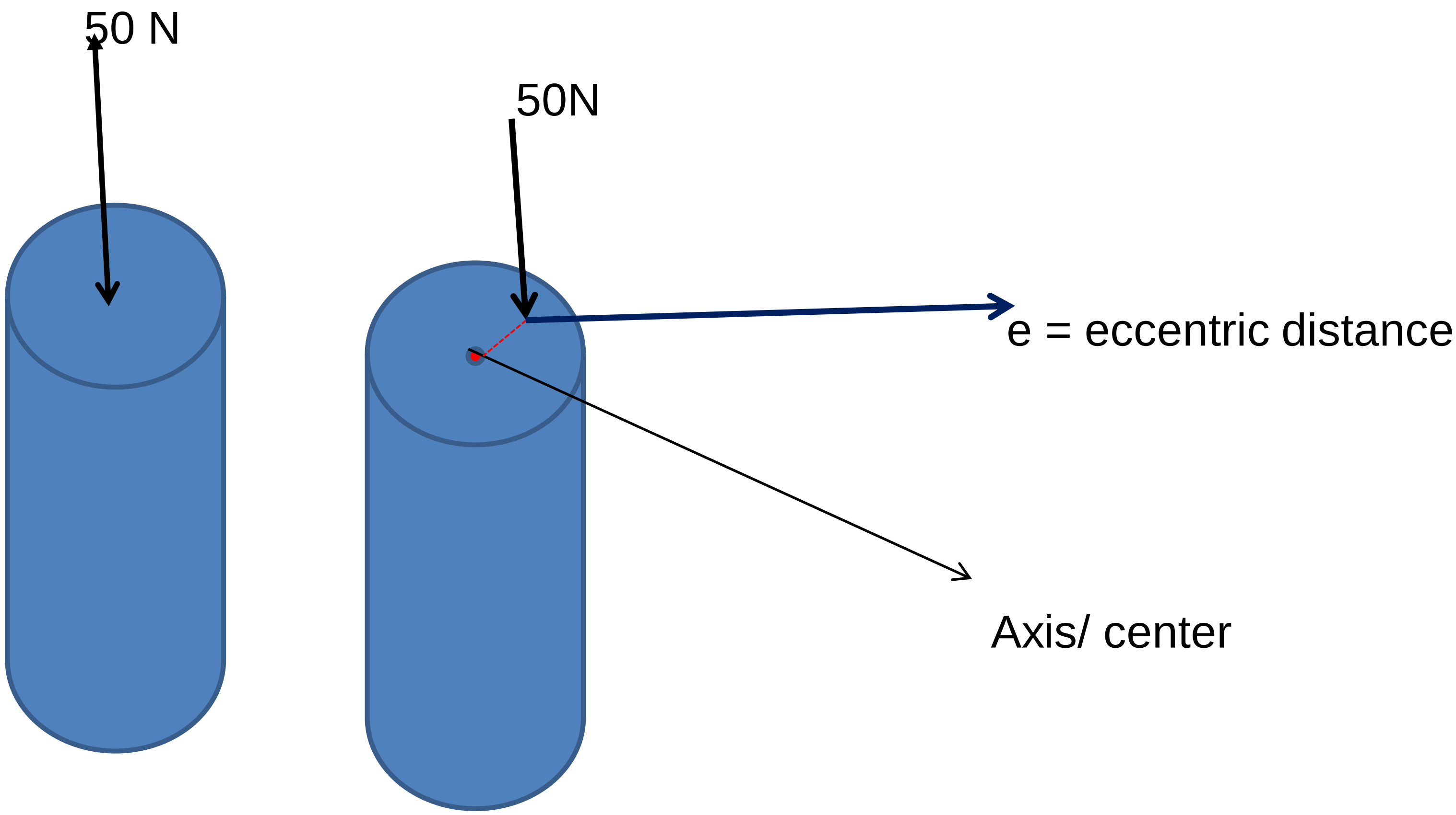
Some important mechanical properties

Failures



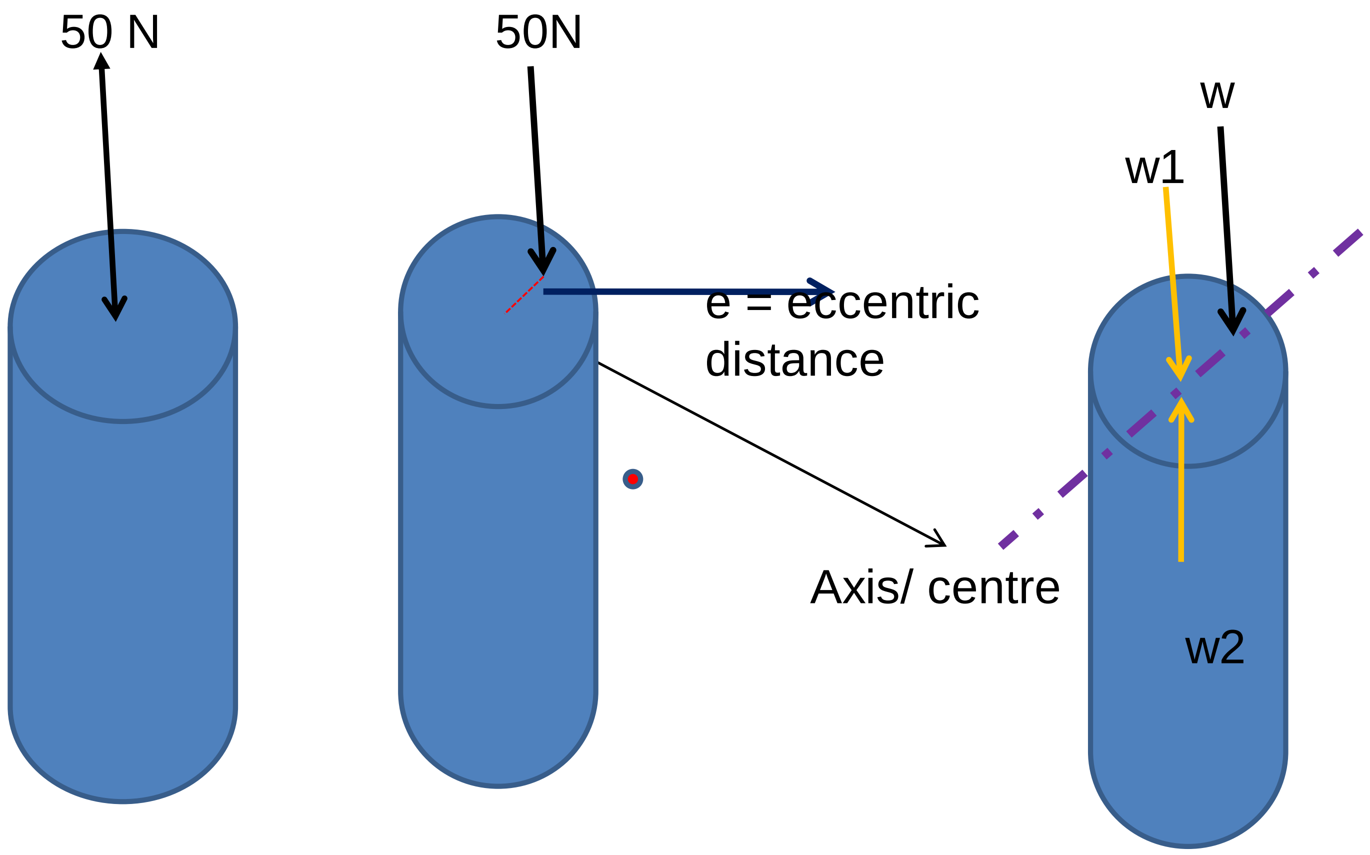

Unloaded
Tension
Compression
Bending
Shear
Torsion
Combined
loading

Eccentric load concept



What is the stress?
What is the load?

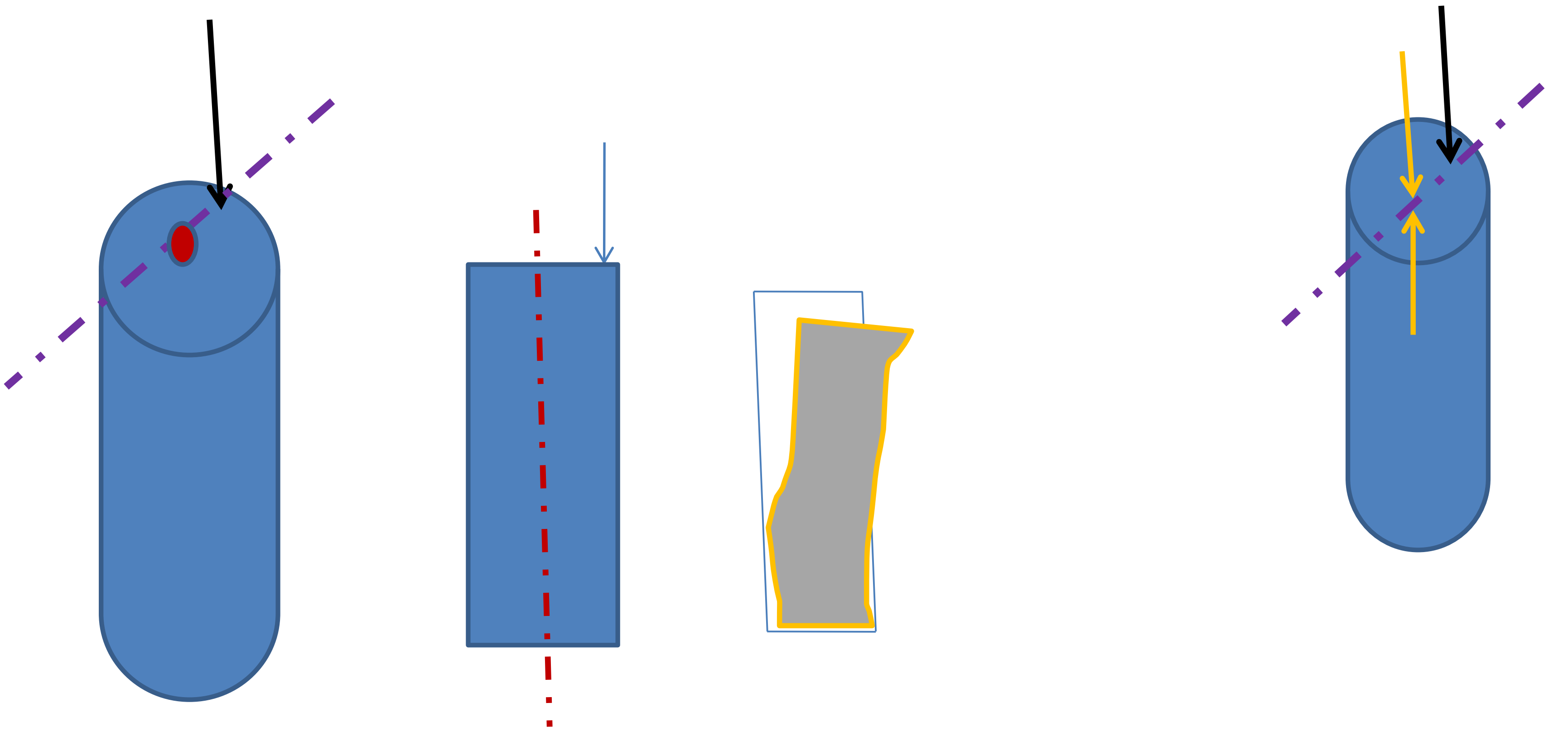
Eccentric load concept



What is the stress?--
compressive or direct stress

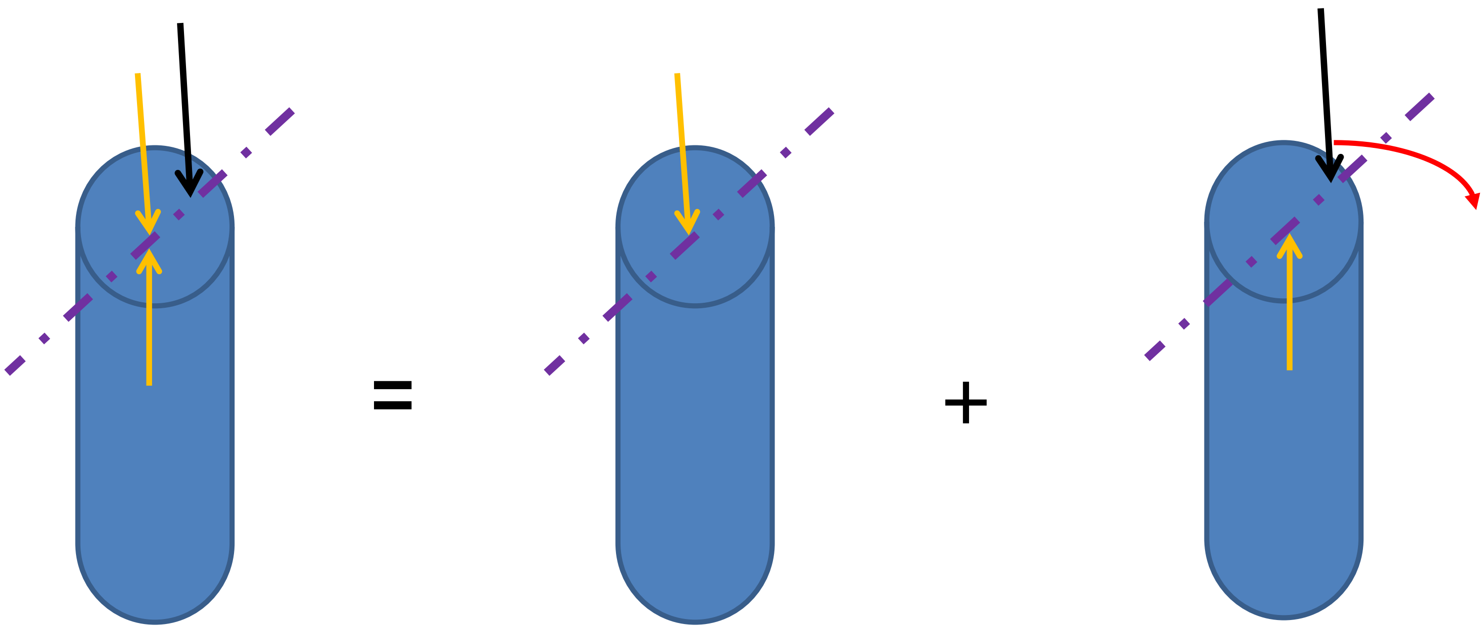
What is the load?--- Axial load

W-- original load
W1- introduced
W2 introduced opposite



How the member will fail---???
???? due to   which stress

How to g et resolution for the effect



σ_{com}

σ_{d} = Direct load/area

σ_b = M/z

Solving the problem

Step1: **Introduce imaginary loads equal and opposite at axis(equal to external load value)**

Step2: **Prepare the equivalent diagram , to have combined stress**

Step3: **Calculate Direct stress (by axial load)**

$$\sigma_{d} = \text{load/area}$$

Step4: **calculate bending Stress (By combination of Original load and introduced load)**

$$\sigma_b = M/z$$

where M = moment due to external load= w x e

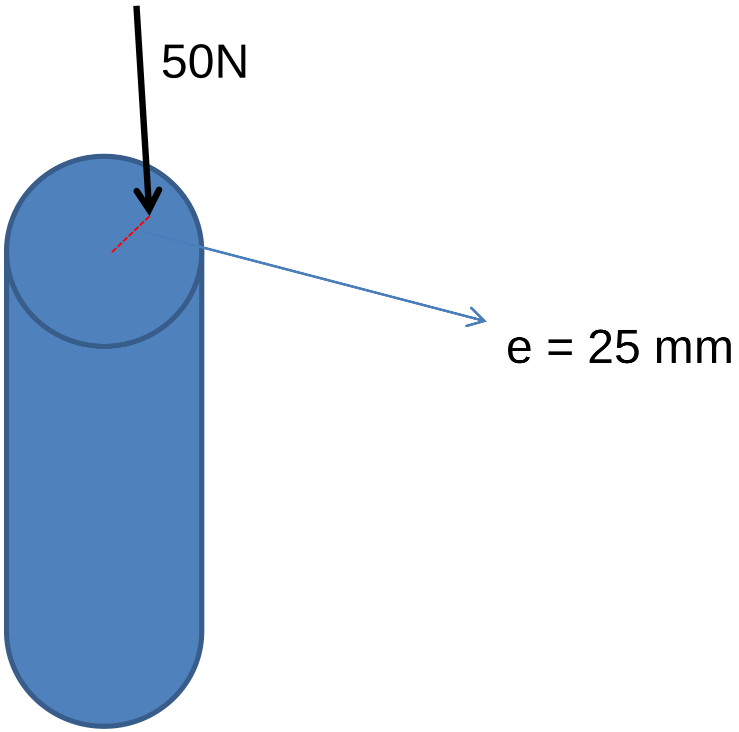
z= section modulus

Step5: **Calculate combined stress ' σ_{com} '**

$$\sigma_{com} = \sigma_d \pm \sigma_b$$

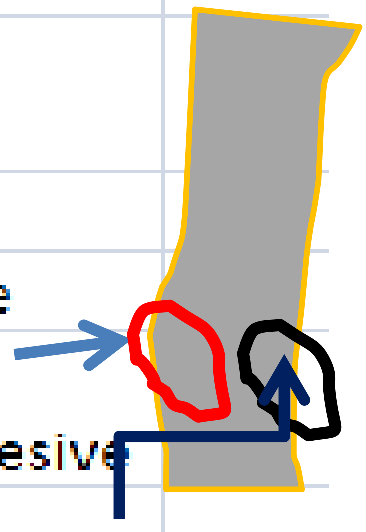
Problem :

A circular member of diameter 40 mm is subject to a load of 50 N, eccentrically 25 mm from the axis as shown in figure. Determine the stresses induced in the member.

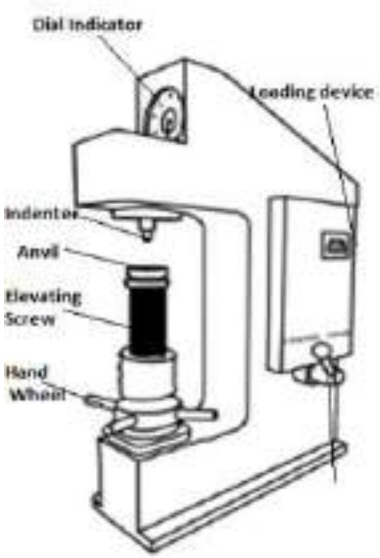


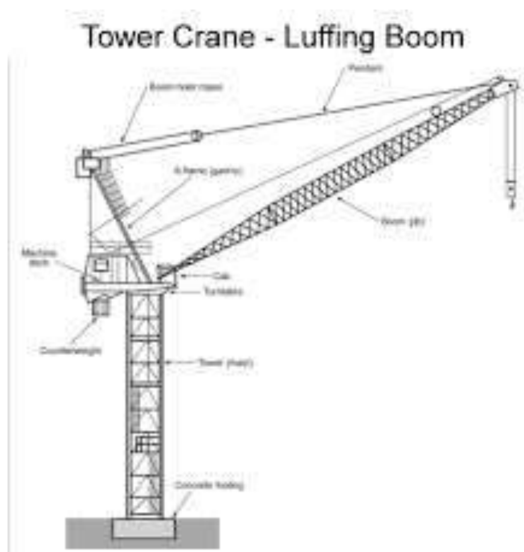
	Data		
	diameter	40	mm
	Load	50	N
	e	25	mm
	area		
	Pi	3.14	
	A	1256	sq.mm
1	Direct stress		
	Stress	0.04	N/sq.mm
2	Bending		
	σb=	M/Z	
	M	Load x e	
		1250	N-mm
	z =	I/y	
		I=	π*d^4/64
		Y=	d/2

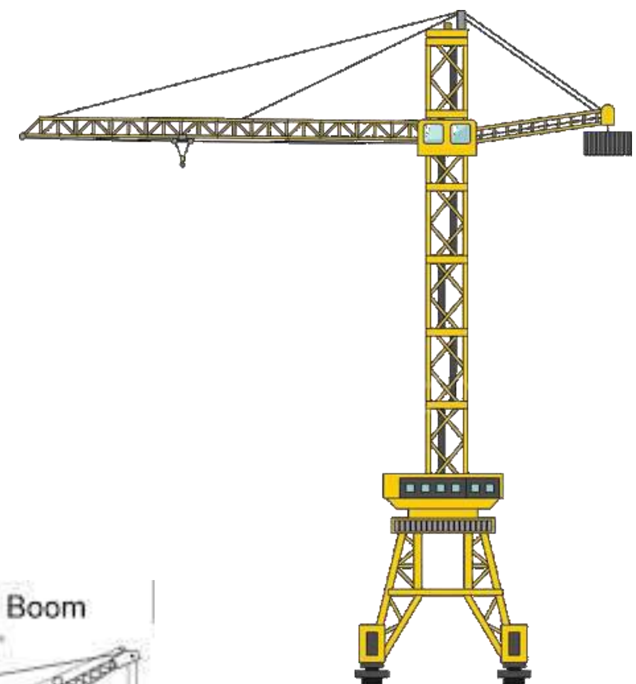
		I =	125640	mm^4
		y =	20	mm
	therfore	z =	6282	mm^3
	Now	σb=	0.198981	N/sq.mm
3	Combined stress			
	σcom=	σd + σ max		0.24 Tensile
		or		
		σd - σl min		-0.16 compresive



Some applications with the eccentric loading concepts

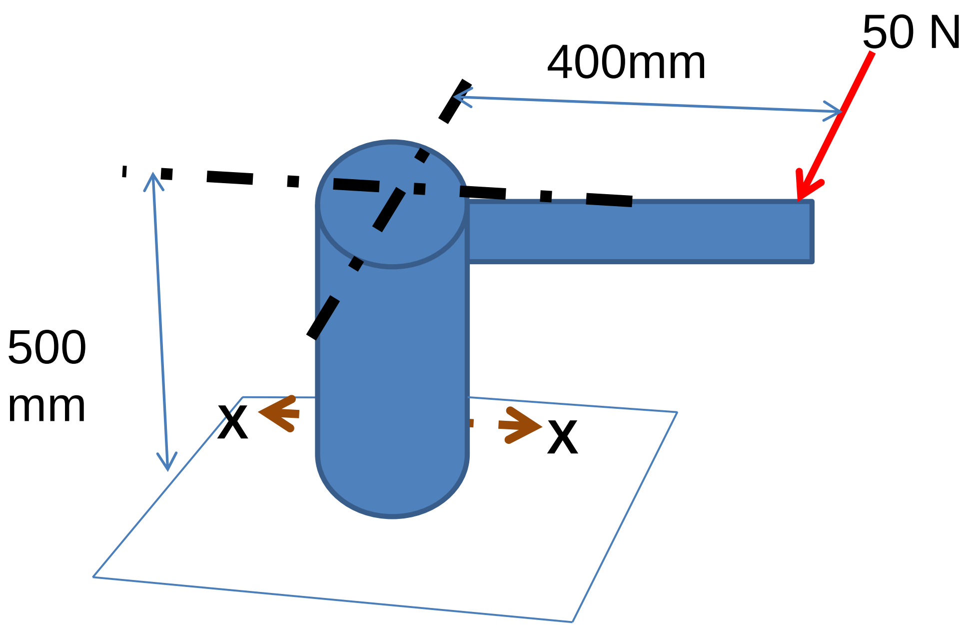




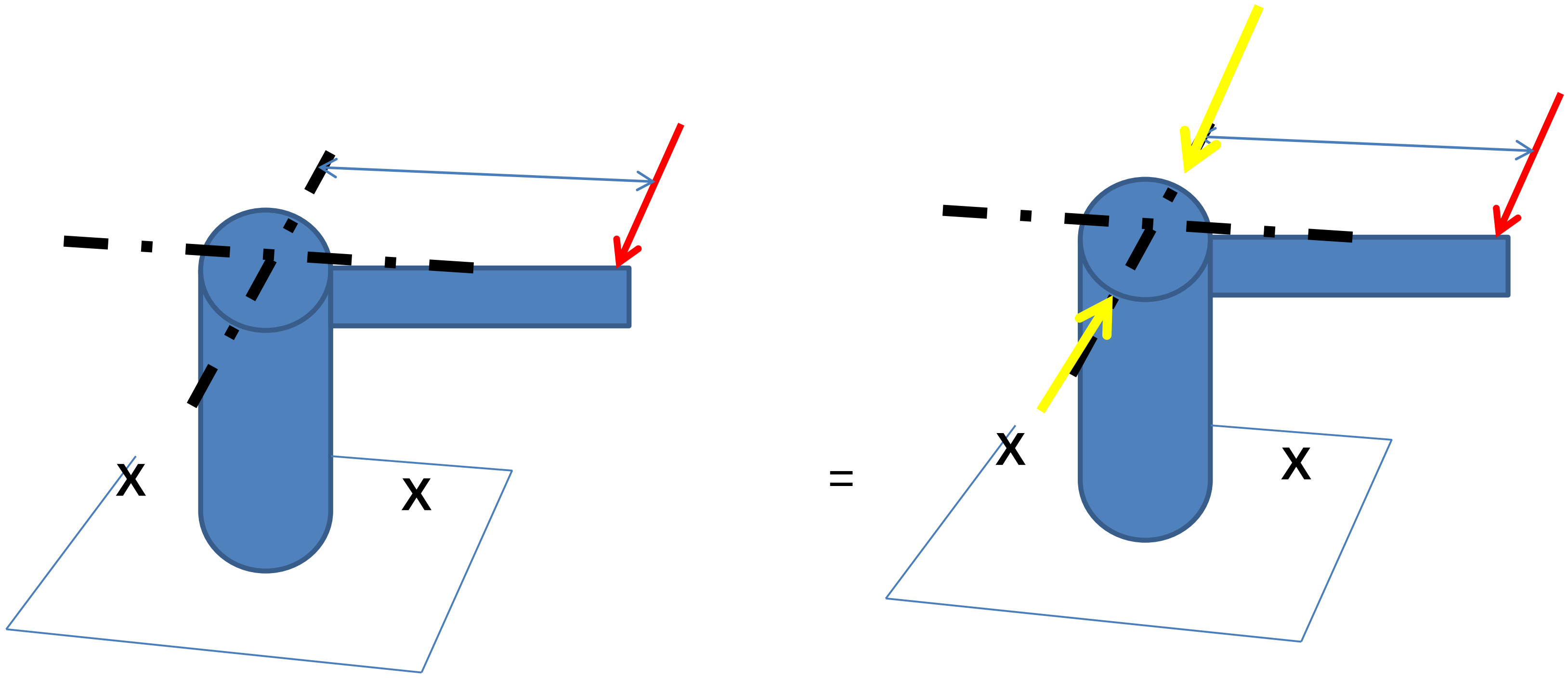


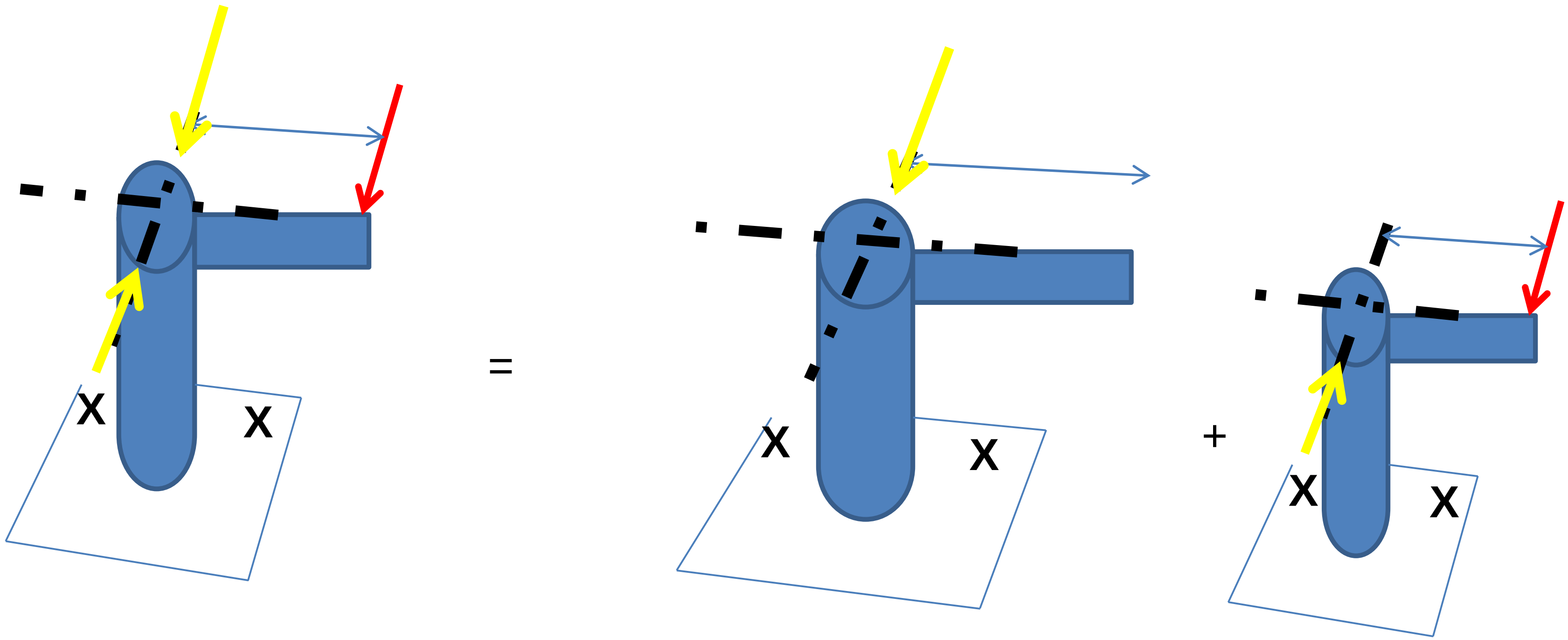
Lifting and Riging

Problem 2



Determine the max and min stresses induced at Section XX, for the member shown in the figure.. Take the diameter of the member is 50 mm


X
X
=
X
X


X
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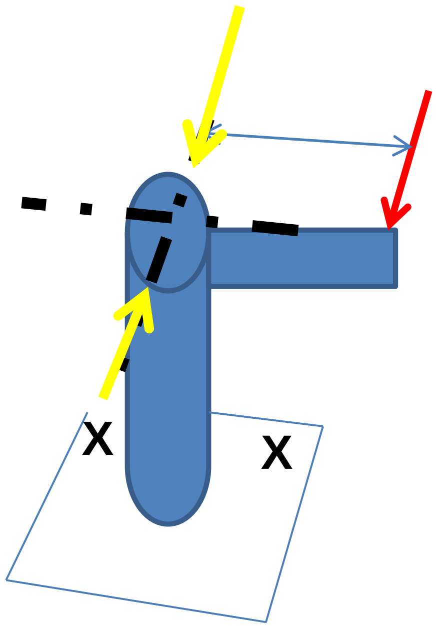
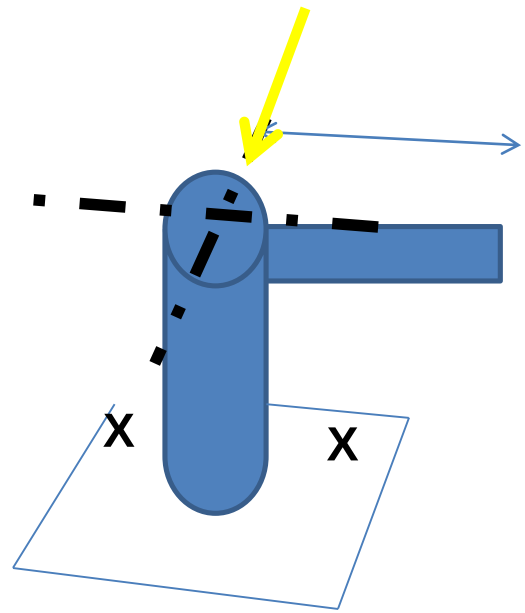


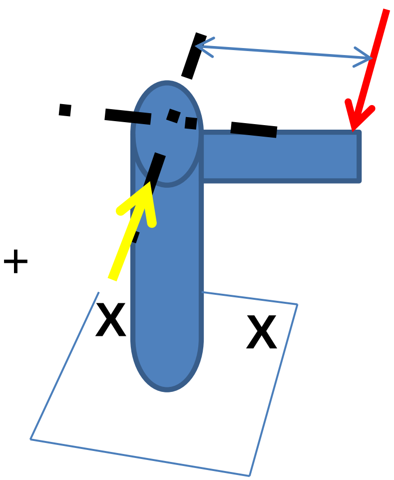
Combined stress

=



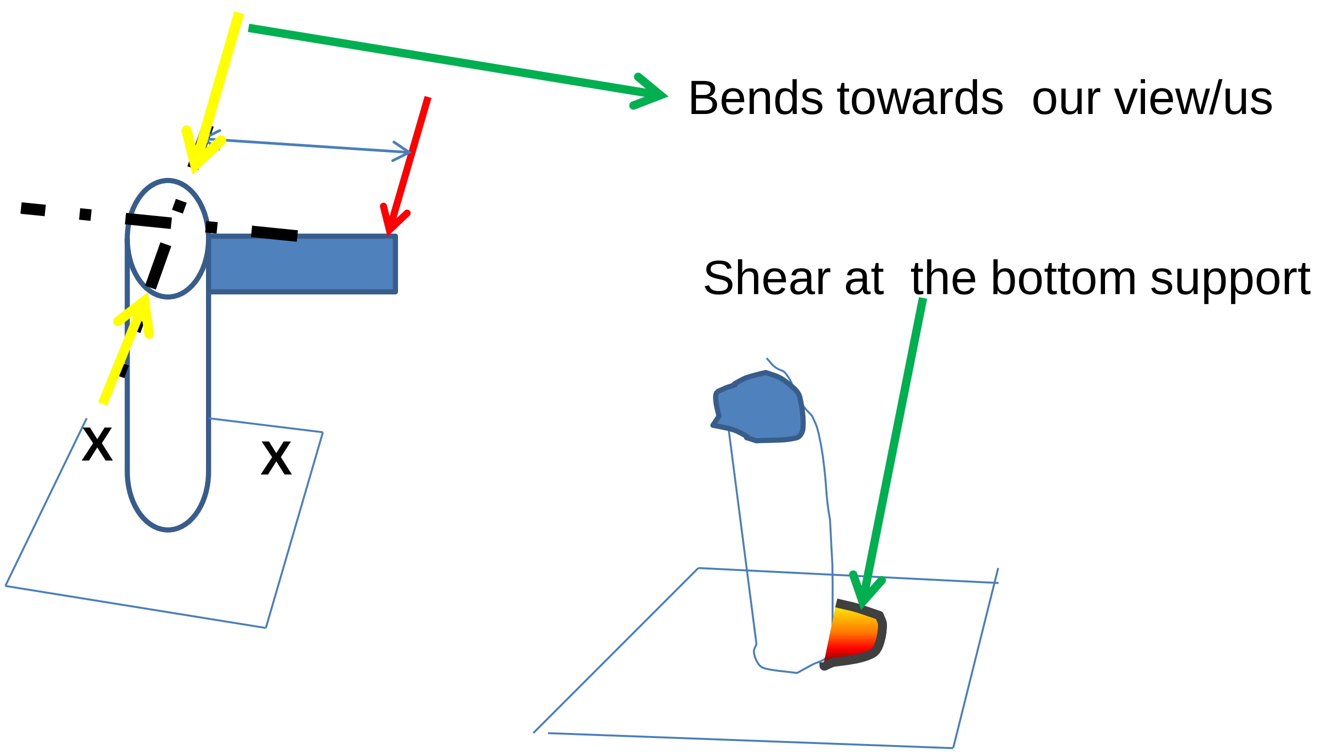
Bending stress

+



Shear stress

Prediction of failure of the member



Solving the problem

Step1: **Introduce imaginary loads equal and opposite at axis(equal to external load value)**

Step2: **Prepare the equivalent diagram , to have combined stress**

Step3: **Calculate Direct stress (by axial load)**

$$\sigma d = \text{load/area}$$

Step4: **calculate bending Stress (By combination of Original load and introduced load)**

$$\sigma_b = M/z$$

where M = moment due to external load= w x e

z= section modulus

Step 5 Calculate Shear stess, T = π/16 x Ʈ x d^3

Step5: **Calculate combined stress ‘ σ_{com} ‘**

$$\sigma_{com} = \sigma_d \pm \sigma_b \quad / \quad = Ʈ \pm \sigma_b$$

Data			
diameter	50	mm	
Load	50	N	
eh	400	mm	(Horizzontal to original load)
ev	500	mm	(Vertical to introduced load from the base)
area			
Pi	3.141		

A

1	Direct stress		
	Stress	0	N/sq.mm

1	Direct stress		
	Stress	0	N/sq.mm

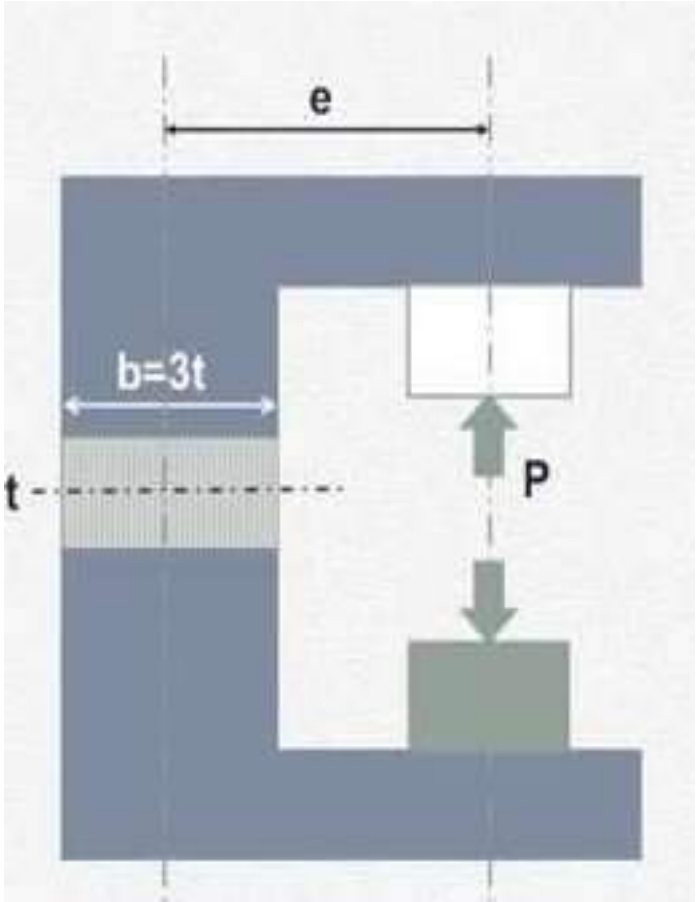
2	Bending						
	σb=	M/Z					
	M	Load x e					
		25000	N-mm				
	z =	I/y					
		I=	π*d^4/64				
		Y=	d/2				
		I =	306738	mm^4			
		y =	25	mm			
	therfore	z =	12269.5	mm^3			

Now	σb=	2.03757	N/sq.mm		

3	Shear stress		T x16/(pi x d*d*d)	refer Pg.No 7.1/DDB
		T=	20000	N mm
			(eccentrical distance is vertical distance from base)	
	Shear stress		0.81503	N/sq.mm

4	Combined stress				
	σcom=	shear stress+ σb	max	2.85	Tensile
		or			
		σb-shear stress	min	1.22	Tensile

Problem for solving



Step1

$$\sigma_d = \frac{P}{A} = \frac{P}{3t^2}$$

Step 2

$$\sigma_b = \frac{M}{Z} = \frac{Pe}{\frac{3}{2}t^3}$$

Step 3:

$$\therefore \sigma_R = \sigma_d + \sigma_b$$

$$\sigma_R = \frac{P}{3t^2} + \frac{Pe}{\frac{3}{2}t^3}$$

$$\therefore \sigma_R = \frac{P}{3t^2} + \frac{2Pe}{3t^3}$$

$$\therefore \sigma_R = \frac{Pt + 2Pe}{3t^3}$$

$$\therefore 3t^3 \times \sigma_R = Pt + 2Pe$$

Factor of safety

$$n = \frac{allowable\ stress}{working\ stress}$$

Allowable stress = design stress

Design stress → $[\sigma]$

Soft material $[\sigma_y]$, y = yield

Brittle material $[\sigma_u]$, u = ultimate

Directional assignment of stresses under combined load

$\sigma_d = \sigma_x$ Direct/ axial load

$\sigma_b = \sigma_y$ Transverse load

$\tau_{xy} = \tau_{yx} = \tau$ Shear stress/load

The directional assignments of stresses will be useful to find the ‘ principal stresses’

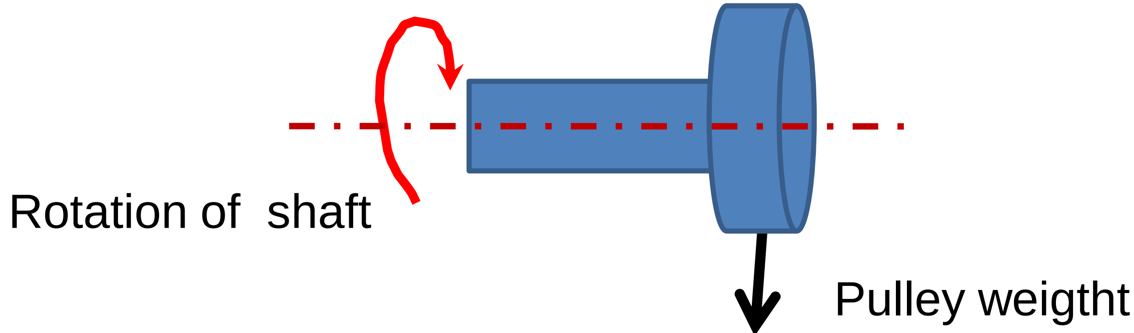
- **Principal stresses are nothing but stresses acting on principal plane**

- **Quite applicable for finding the stress intensities in the form of**

- **‘σ1 ‘ maximum and ‘σ2 minimum**

- **When the members subjected to combined stresses and simple shear stress**

Example : belt drive system – shafts carries pulley-



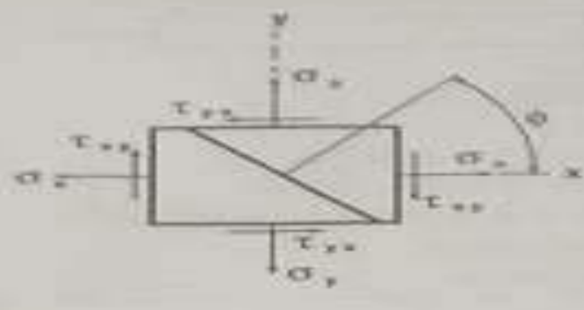
In general, Under strained material all the stress will be mutually perpendicular in three planes xy,xz,yz.

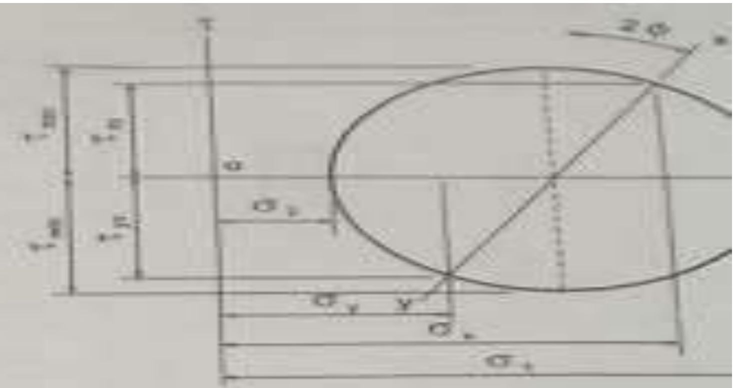
Out of the three one is max and the other is minimum

Now the combined stress problem is advancing with approach of principal stresses

Now refer the data book page : 7.2 two dimensional stresses

σ_1 , σ_2 and τ





Two Dimensional Stresses		
$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$	$\sigma_1, \sigma_2, \sigma_3$	Pri… Str…
$$\tau = \pm\frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}$$	$\sigma_x, \sigma_y, \sigma_z$	Str… Co…
Three Dimensional Stresses	$\tau_{xy}, \tau_{yz}, \tau_{zx}$	Sh… co…
$$\begin{vmatrix} (\sigma_x - \sigma) & \tau_{xy} & \tau_{xz} \\ \tau_{yx} & (\sigma_y - \sigma) & \tau_{yz} \\ \tau_{zx} & \tau_{zy} & (\sigma_z - \sigma) \end{vmatrix} = 0$$	σ_y	Yi…
	ψ	P…

The roots of this cubic equation in σ are the principal stresses

Now The objectives of solving problems

$$n = \frac{allowable\ stress}{working\ stress}$$

Finding dimension of the member

Checking design for safety

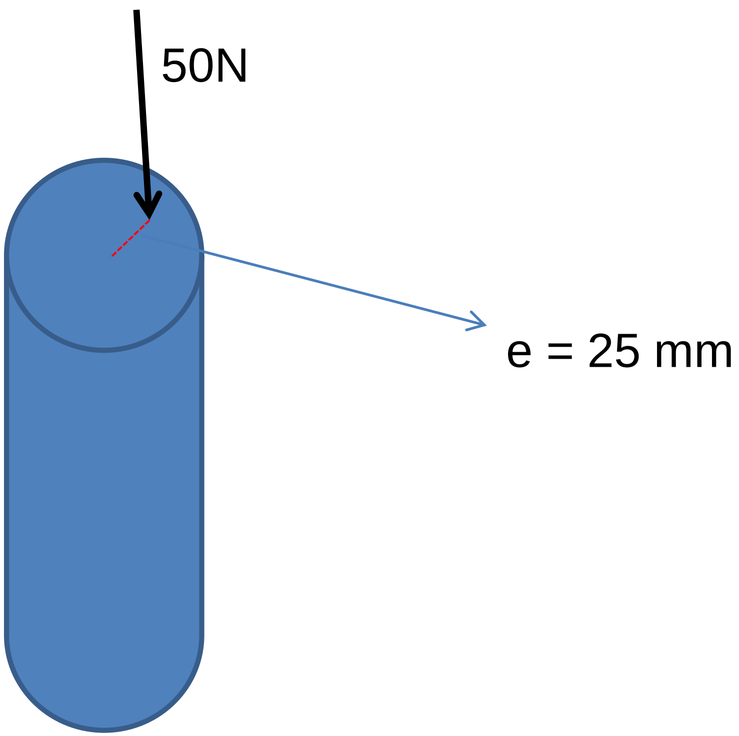
Induced stress < Design Stress

$\sigma_i < [\sigma]$,

$\tau < [\tau]$

Problem 3

A circular member of diameter 40 mm is subject to a load of 50 N, eccentrically 25 mm from the axis as shown in figure. Determine the stresses induced in the member. If the allowable stress for the member material is 2 N/mm² Also determine the factor of safety



Solving the problem

Step1: **Introduce imaginary loads equal and opposite at axis(equal to external load value)**

Step2: **Prepare the equivalent diagram , to have combined stress**

Step3: **Calculate Direct stress (by axial load)**

$$\sigma d = \text{load/area}$$

Step4: **calculate bending Stress (By combination of Original load and introduced load)**

$$\sigma_b = M/z$$

where M = moment due to external load= w x e

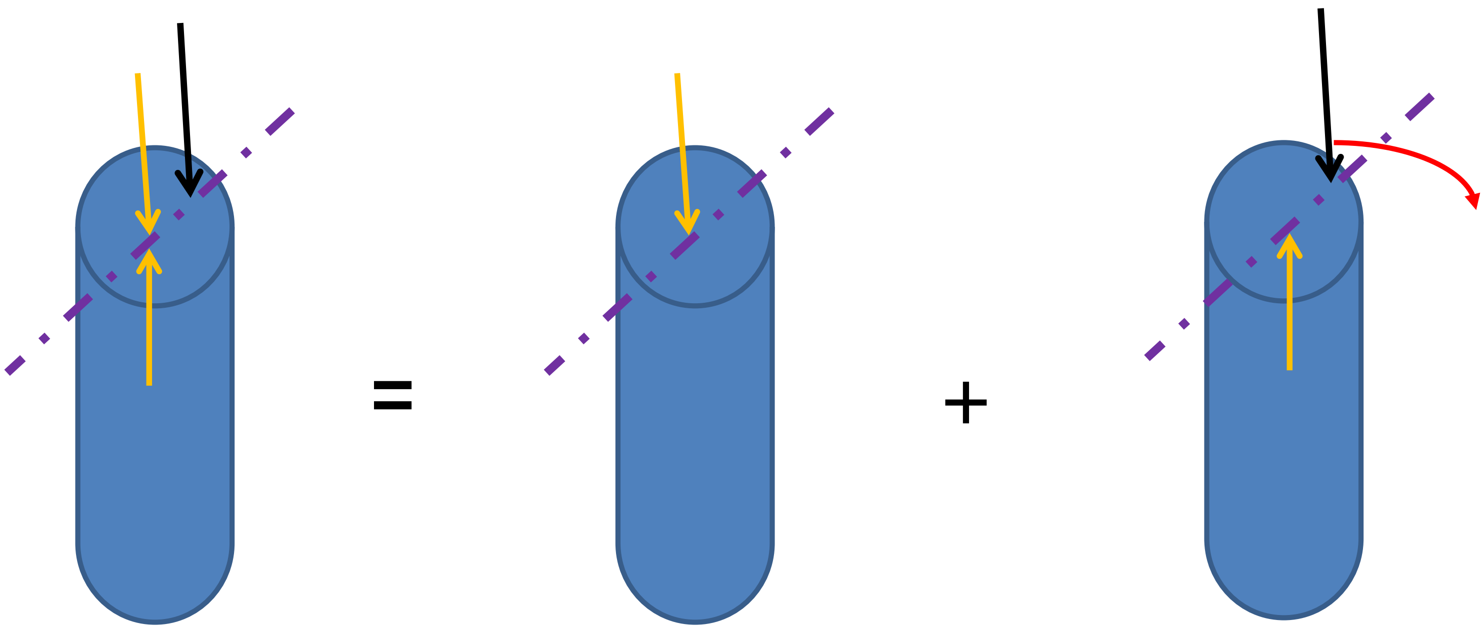
z= section modulus

Step 5 Calculate Shear stess, T = π/16 x Ʈ x d^3

Step5: **Calculate ~~combined stress~~ ' ~~σ_{com}~~ ' Principal stresses**

$$\sigma 1 \ \& \ \sigma 2 \ , \ \tau$$

How to g et resolution for the effect



σ_{com} = σ_{d} = Direct load/area + $\sigma_b = M/z$

Step3: **Calculate Direct stress (by axial load)**

$$\sigma_d = \text{load/area}$$

$$\sigma_d = \sigma_x = ???$$

Step4: calculate bending Stress (By combination of Original load and introduced load)

$$\sigma_b = M/z$$

where M = moment due to external load= w x e

z= section modulus

$$\sigma_b = \sigma_y = ???$$

Step3: **Calculate Direct stress (by axial load**)

$$\sigma_{d} = \text{load/area}$$

$$\sigma_{d} = \sigma_{x} = 0.04 \text{ N/sq.mm}$$

Step4: calculate bending Stress (By combination of Original load and introduced load)

$$\sigma_{b} = M/z$$

where M = moment due to external load= w x e

z= section modulus

$$\sigma_{b} = \sigma_{y} = 0.198 \text{ N/Sq.mm}$$

Step5: **Calculate combined stress ‘ σ_{com} ‘**

$$\sigma_{com} = \sigma_d \pm \sigma_b$$

Step6: **Calculate** σ_1 and σ_2

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_x^2}\right]$$

In this problem , shear stress is zero, $\tau = 0$

	Data			
	diameter	40	mm	
	Load	50	N	
	e	25	mm	
	area			
	Pi	3.141		
	A	1256.4	sq.mm	

1	Direct stress		
	Stress	0.0398	N/sq.mm
	σx	0.0398	

2	Bending		
	σb=	M/Z	
	M	Load x e	
		1250	N-mm
	z =	I/y	
		I=	π*d^4/64

	Y=	d/2	
	I =	125640	mm^4
	y =	20	mm
therfore	z =	6282	mm^3
Now	σb=	0.1989812	N/sq.mm
	σy =	0.198	

3 principal stress		
σ_1, σ_2=	??	
$\sigma x + \sigma y$	0.2378	**N/sq.mm**
$\sigma x - \sigma y$	-0.1582	**N/sq.mm**
$(\sigma x - \sigma y)^2$	0.0250	
$(\sigma x + \sigma y)/2$	0.1189	
$(\surd\,(\sigma x - \sigma y)2 + 0)/2$	0.0791	

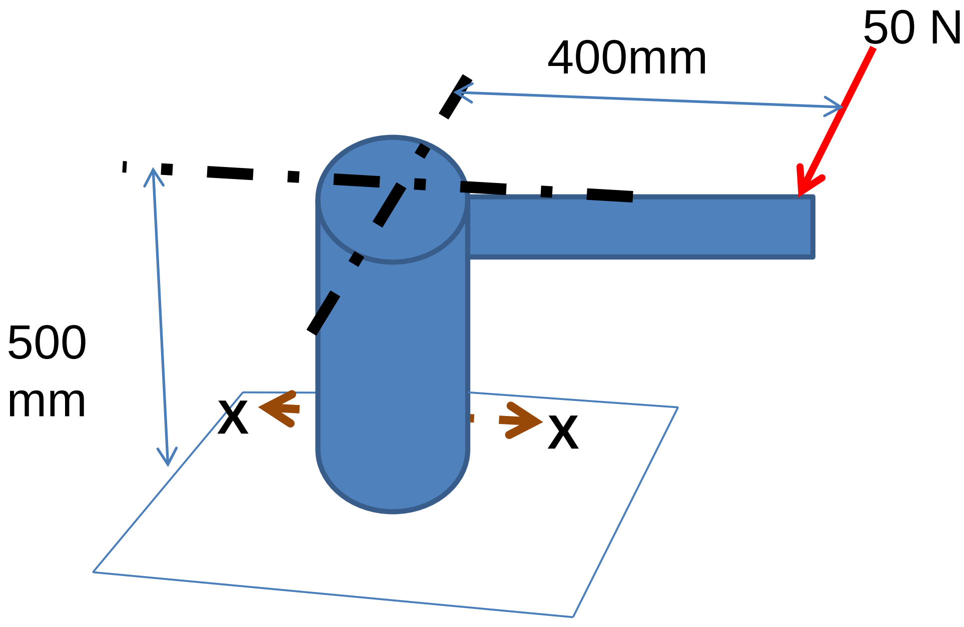
$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$\left(\frac{\sigma x + \sigma y}{2}\right) \pm 1/2\sqrt{(\sigma x - \sigma y)^2 + 4\text{T}^2}$$

$\sigma 1$ =		0.198	**N/sq.mm**
$\sigma 2$ =		0.0398	**N/sq.mm**
factor of saftey			
n=	**10.1**		

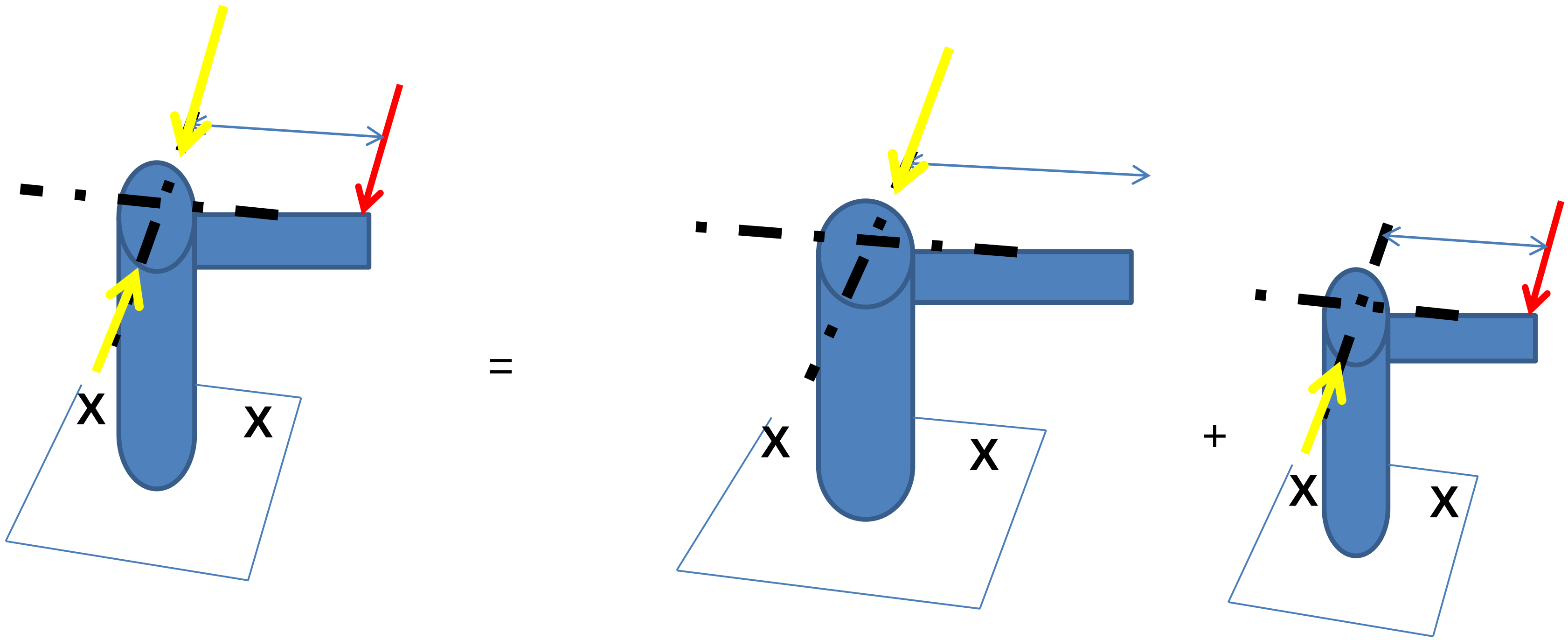
Maximum stress to be taken
As working stress.

Problem 2



Determine the principal stresses induced at Section XX, for the member shown in the figure.. Take the diameter of the member is 50 mm.

Take allowable stress is 5 N/ sq. Mm
Also find the factor of safety.



Combined stress

Bending stress

Shear stress

Data

diameter	**50mm**	
Load	**50N**	
eh	**400mm**	**(Horizontal to original load)**
ev	**500mm**	**(Vertical to introduced load from the base)**
area		
Pi	**3.141**	
A	**1963.125sq.mm**	

1 **Direct stress**

Stress	**0**	**N/sq.mm**
σx	**0**	

2 Bending			
σb=	M/Z		
M	Load x e		
	25000	N-mm	
z =	I/y		
	I=	π*d^4/64	
	Y=	d/2	
	I =	306738	mm^4
	y =	25	mm
therfore	z =	12269.5	mm^3
Now	σb=	2.03757	N/sq.mm
	σy	2.037	

3	Shear stress		T x16/(pi x d*d*d)			refer Pg.No 7.1/		
								DDB
		T=	20000	N mm				
			(eccentrical distance is horizontal distar					
	τ Shear stress		0.81503	N/sq.mm				
		τ	0.815					

Principal stress

$\sigma 1$, $\sigma 2$= ??

$\sigma x + \sigma y$	2.037N/sq.mm
$\sigma x - \sigma y$	-2.037N/sq.mm
$(\sigma x - \sigma y)^2$	4.1494
$(\sigma x + \sigma y)/2$	1.0185
τ	0.815
τ^2	0.664
(√ (σx – σy)2 +4ʈ2/2	1.304

$\sigma 1$ =	2.322941356 N/sq.mm
$\sigma 2$ =	-0.285941356 N/sq.mm

$\sigma_i < [\sigma]$,

2.32 <[5]   design is safe

factor of saftey

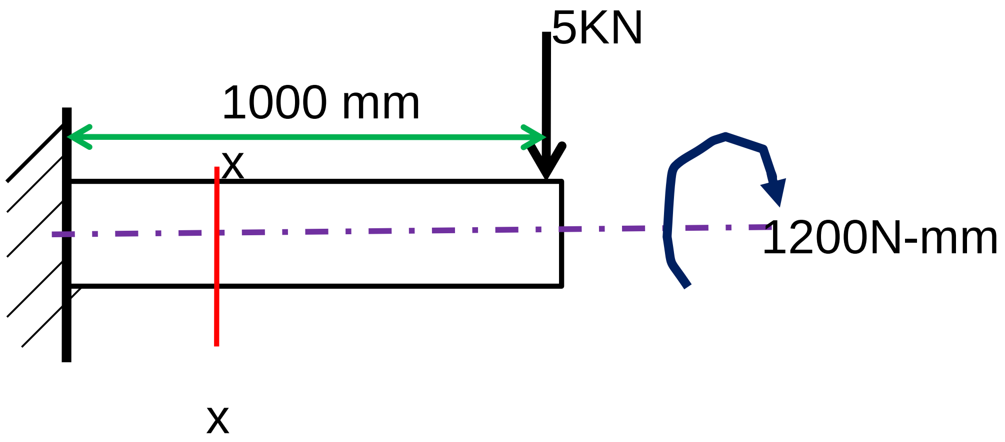
n= 2.15

AS per shear Stress Theory

$$\tau = \pm \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}$$

τ	+	1.304 N/sq.mm
τ	-	1.304 N/sq.mm

Problem 3



A cantilever circular member is subject to loads as shown in figure. Determine the diameter of it .If The allowable stress is 90 N/mm^2

STEP1. Indentify the loads types and its effect on it

Step2: calculate the individual stresses 2 or 3

Step3 : Take into the Principal stress equations

Note: If the allowable stress given in the problem, then

That is design stress $[\sigma] = [\sigma_1]$,

Find the dimension of the member

ref. pg.No.7.2,/DDB

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4\tau xy^2\right)}$$

Step1

Calculation of direct stress - $\sigma_{d} = \sigma_{x}$

$\sigma_{d} =$ **load/area= NO**

Step2

Calculation of bending stress $\sigma_{b} = \sigma_{y}$

$$\sigma_b = M/z$$

where M = moment due to external load= w x e

z= section modulus

Step3: Calculate Shear stess,

T = π/16 x Ʈ x d^3

Step 4 Take into the Principal stress equations

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4Ʈxy^2\right)}$$

$$[\sigma] = [\sigma_1],$$

$$[\sigma_1] = [90] \text{ N/mm}^2$$

Step3

Using principal stress equation, Find diameter of The member

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4\tau xy^2\right)}$$

σ_1 **is to be tken as allowable stress**

Data			
diameter	x	mm	
bendingLoad	5000	N	
Distance for bending lo	1000	mm	
Turning moment	1200	mm	
area			
Pi	3.141		
[σ1]	90	N/sq.mm	
A	0.78525	d^2	sq.mm

1	**Direct stress**		
	Stress	**no axial load**	**N/sq.mm**
	σx	**0**	

2	**Bending**				
	σb=	**M/Z**			
	M	**Load x e**			
		5000000	**N-mm**		
	z =	**I/y**			
		I=	**π*d^4/64**		
		Y=	**d/2**	**0.5*d**	
		I =	**0.049**	*** d^4**	**mm^4**

therfore	z =	0.098156 * d^3	mm^3
Now	σb=	50939191 *(1/d^3)	N/sq.mm
	σy	50939191 *(1/d^3)	

3	Shear stress		T x16/(pi x d*d*d)		
		T=	1200	N mm	
	τ Shear stress		6112.703 *(1/d^3)		N/sq. mm

Principal stress

σ1, σ2=	??		
σx + σy	50939191.34	*(1/d^3)	N/sq.mm
σx - σy	**-50939191.34**	*(1/d^3)	**N/sq.mm**
$(\sigma x - \sigma y)^2$	2594801214407510	*(1/d^3)^2	
(σx + σy)/2	25469595.67	***(1/d^3)**	

τ	6112.703	*(1/d^3)
τ^2	37365137.487	*(1/d^3)^2
(√ (σx – σy)2 +4τ2/2	25469596.404	*(1/d^3)

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4\tau xy^2\right)}$$

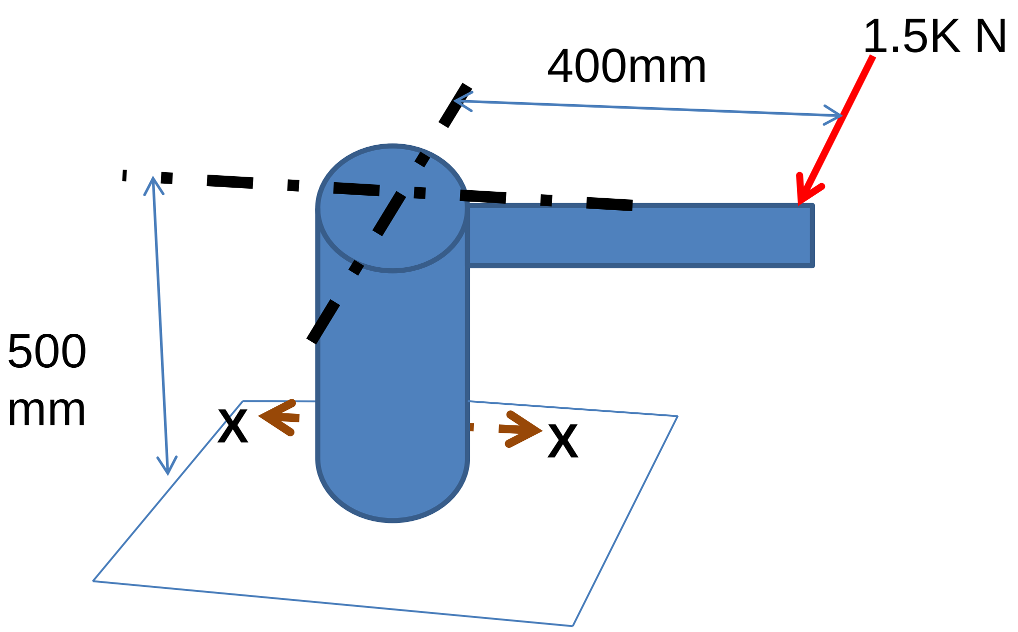
Asper principal stress

[90]=	50939192.074	*(1/d^3)
90	50939192.074	*(1/d^3)

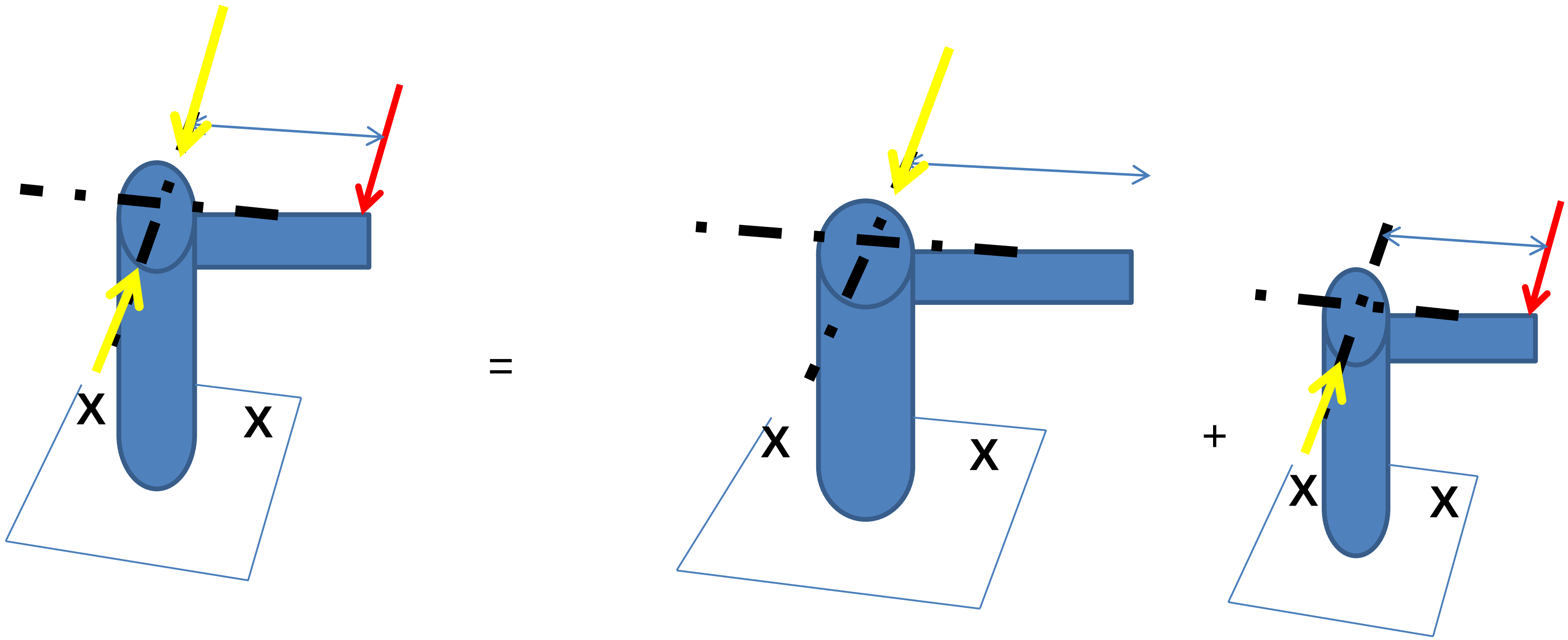
d^3= 565991.023

d 82.72 mm

Problem 4



Determine the diameter of the member at Section XX, for the member shown in the figure. Take the allowable stress of the member is 120 N/sq. mm



Combined stress

Bending stress

Shear stress

STEP1. Indentify the loads types and its effect on it

Step2: calculate the individual stresses 2 or 3

Step3 : Take into the Principal stress equations

Note: If the allowable stress given in the problem, then

That is design stress $[\sigma] = [\sigma_1]$,

Find the dimension of the member

ref. pg.No.7.2,/DDB

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4\tau xy^2\right)}$$

Step1

Calculation of direct stress - $\sigma_d = \sigma_x$

$\sigma_d =$ load/area= NO

Step2

Calculation of bending stress $\sigma_b = \sigma_y$

$\sigma_b = M/z$

where M = moment due to external load= w x e

z= section modulus

Step3: Calculate Shear stess,

T = π/16 x Ʈ x d^3

Step 4 Take into the Principal stress equations

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4Ʈxy^2\right)}$$

$$[\sigma] = [\sigma_1],$$

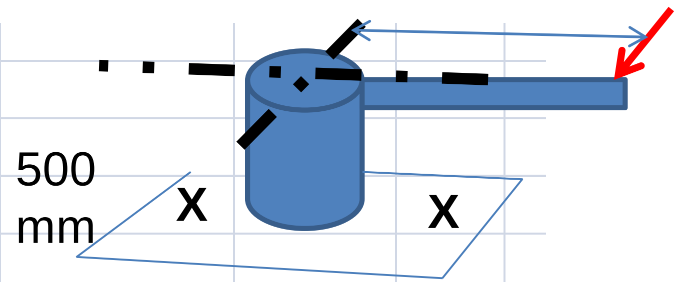
$$[\sigma_1] = [120] \text{ N/mm}^2$$

Data			
diameter	d	mm	
Load	1500	N	(1.5 x 1000)
eh- turning effect	400	mm	(Horizontal to original load)
ev- bending	500	mm	(Vertical to introduced load from the base
area			
Pi	3.141		

1	Direct stress		
	Stress	0	N/sq.mm
	σx	0	

Due to no axial load

2	Bending				
	σb=	M/Z			
	M	Load x e			
		750000	N-mm		
	z =	I/y			
		I=	π*d^4/64		
		Y=	d/2		
		I =	0.049	*d^4	mm^4
		y =	0.5	d	mm
	therfore	z =	0.0981563	mm^3	
	Now	σb=	7640878.7	*(1/d^3)	N/sq.mm
		σy	7640878.7	*(1/d^3)	



3	Shear stress		T x16/(pi x d*d*d)	refer Pg.No 7.1/DDB
		T=	600000 N mm	
			(eccentrical distance is horizontal distance from axis)	
	τ Shear stress		τ= 3056351.5 *(1/d^3)	N/sq.mm

4	Principal stress		
	σ1, σ2=	??	
	σx + σy	7640878.701	**N/sq.mm**
	σx - σy	**-7640878.701**	**N/sq.mm**
	$(\sigma x - \sigma y)^2$	58383027324169.0000	***(1/d^3)^2**
	(σx + σy)/2	3820439.351	***(1/d^3)**

τ	3056351.48	*(1/d^3)
τ^2	9341284371867.040	*(1/d^3)^2
(√ (σx – σy)2 +4Ʈ2/2	4892549.561	*(1/d^3)

$$\sigma_{1,2} = \frac{\sigma x + \sigma y}{2} \pm \frac{1}{2}\sqrt{\left((\sigma x - \sigma y)^2 + 4\tau xy^2\right)}$$

Asper principal stress

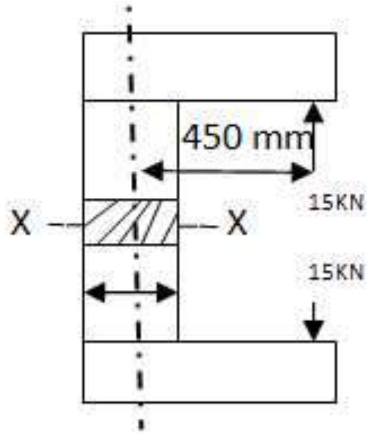
[90]=	8712988.911	*(1/d^3)
120 90	8712988.911	*(1/d^3)

d^3= 96810.9879

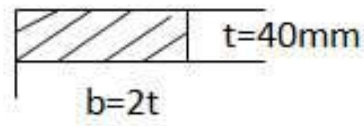
d 45.92 **mm**

41.72

Tutorial 1 3-7-2020

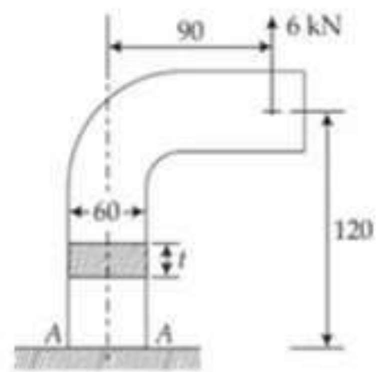


At X-X,



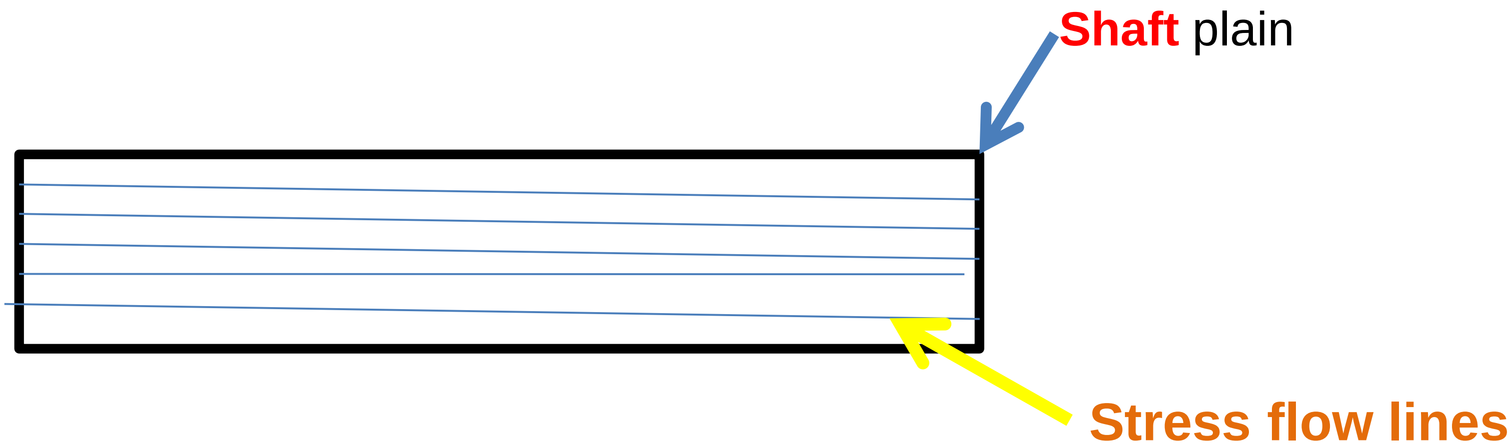
- Determine the principal stresses in C – clamp at XX section resembles rectangular.

- Determine the thickness of the steel bracket loaded as shown in fig. taking allowable stress as 90 Mpa



STRESS CONCENTRATION FACTOR

STRESS CONCENTRATION FACTOR- K_t



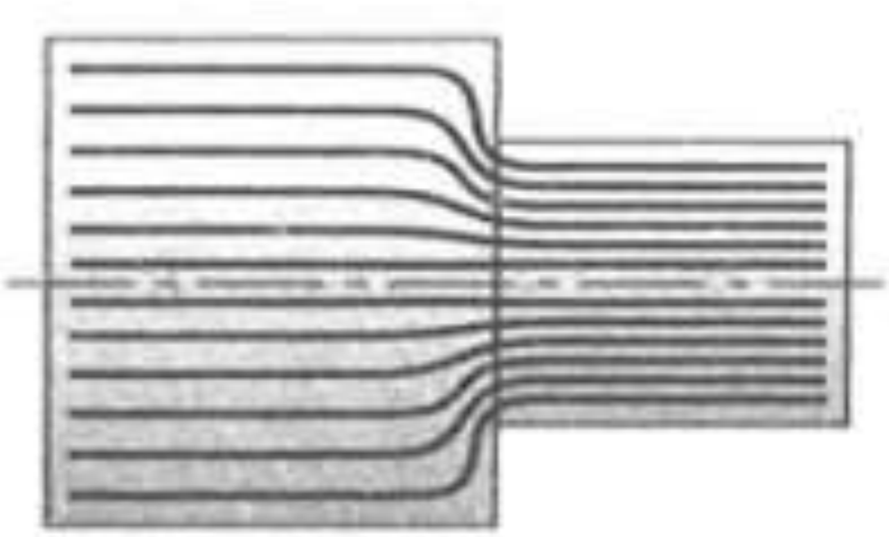
Def:

The stress concentration factor, , is the ratio of the highest stress to a nominal stress of the gross cross-section

$$K_t = \sigma_{Max} / \sigma_{nominal}$$

where σ_{Max} **is** allowable or Design Stress

$\sigma_{nominal}$ **is the stress with respect to area of cross section of the member**



Force flow lines in an abrupt changes in the cross section of the member

Methods of Reducing Stress Concentrations

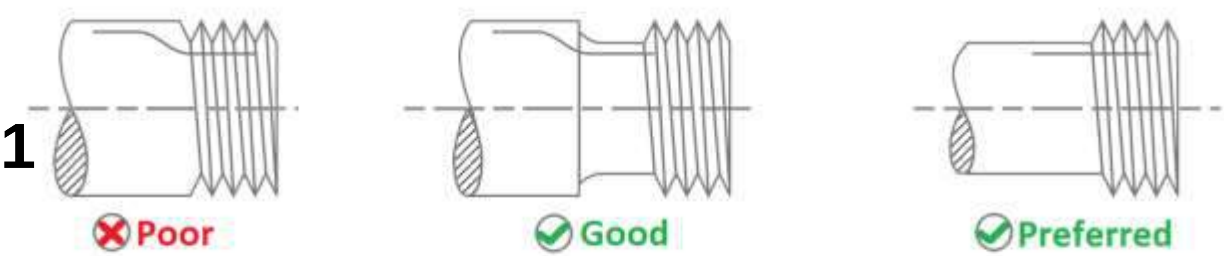
- ❑**Avoiding sharp** corners and only **using rounded** corners with maximum radii.

- ❑**Sanding and polishing** surfaces to remove any notches or defects that occur during forming and processing.

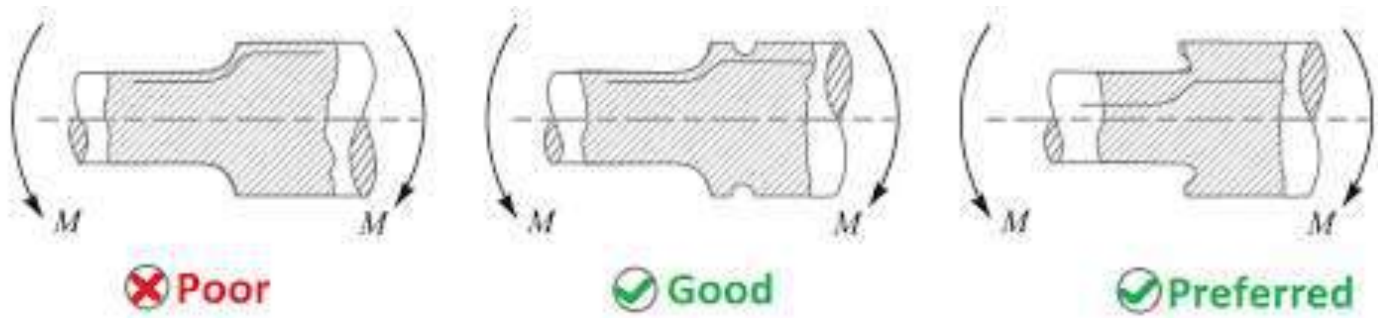
- ❑Lowering the stiffness of straight load-bearing segments.

- ❑**Placing notches** and threads **in low-stress areas.**

1


Poor
Good
Preferred

2


M
M
M
M
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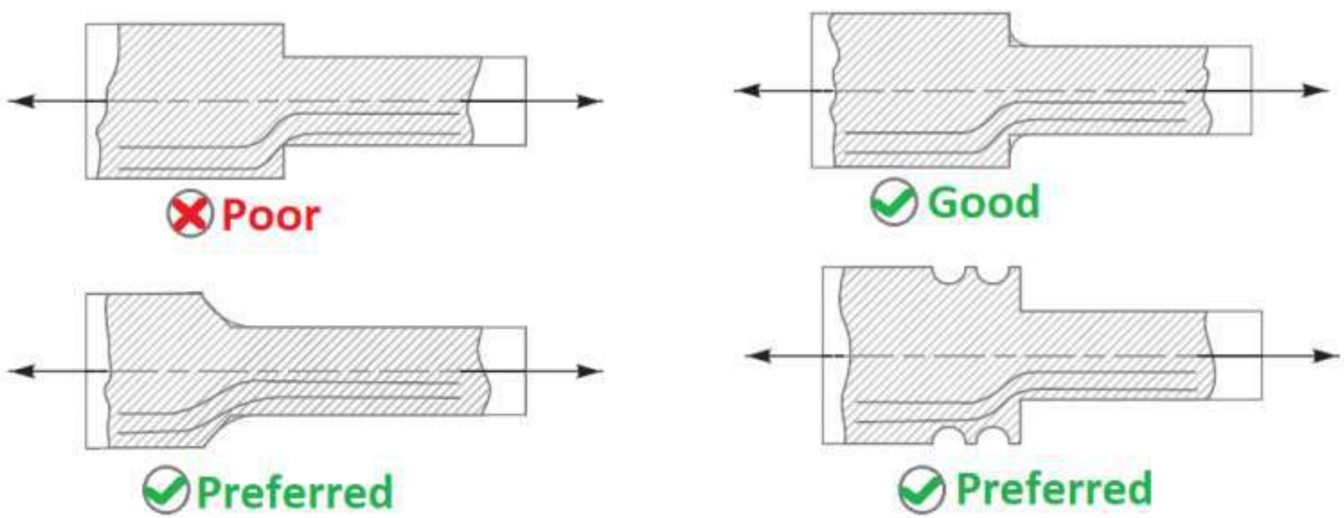
M
Poor
Good
Preferred

3


Poor
Good
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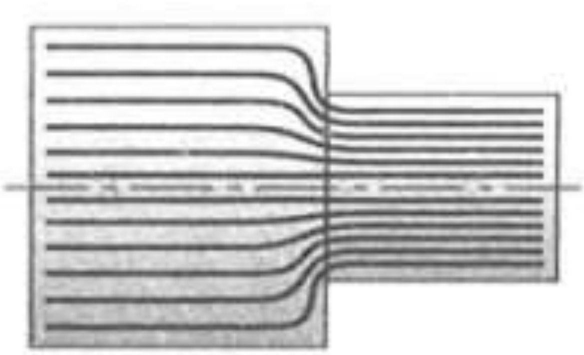
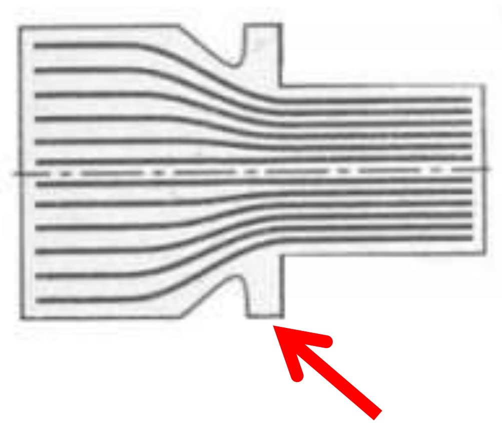


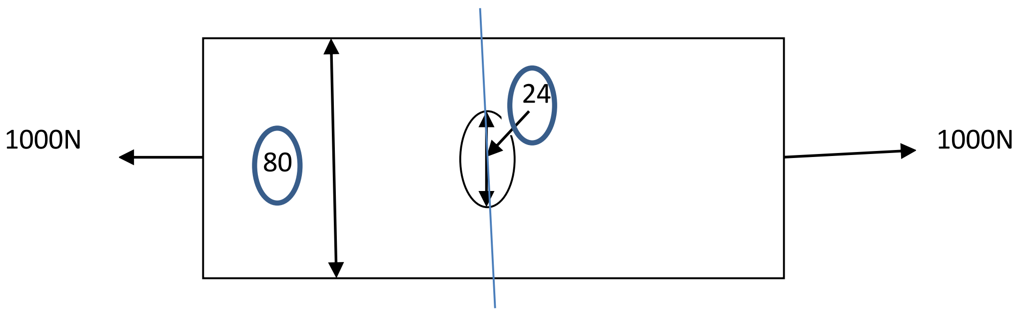


Stress relieving groove

To solve problems

refer design data book pages 7.8 to 7.16 pages
for variou types of members and their loading condition.

P5. Find the maximum stress induced in the plate as shown in figure. All sizes are in mm T=15MM



$$Kt = \frac{\sigma max}{\sigma nominal/working}$$

Identify the Notations as per data book Refer pg.No 7.10/DDB

W= 80 mm & a = 24 mm

Stress concentration Factor					
DATA					
Load	p	1000	N		
Width	W	80	mm		
hole di:	a	24	mm		
Thuckness		15	mm		
Area		(w-a)t			
		840	mm^2		
σmax		???			

1σnominal L/A

1.19 N/mm^2

2 Kt ?

To be obtained rom DDB 7.10

Ratio a/w

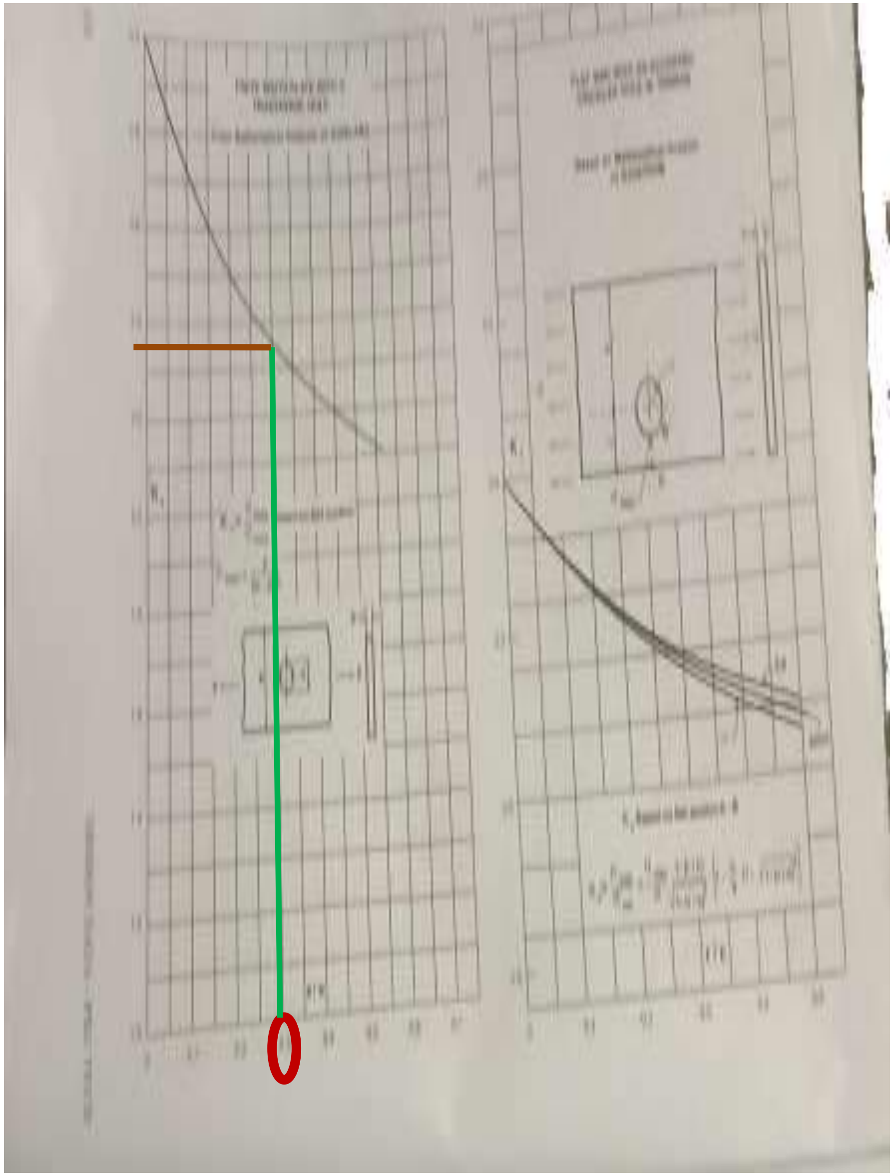
0.3

Kt 2.4

3 σmax

σmax = Kt x σnom

2.8 N/mm^2



P6. Find the maximum stress induced in the object as shown in figure. All sizes are in mm

Load is 1KN, Bigger dia is 80 mm and stepped dia is 40 mm. The radius of the fillet is 8 mm

Stress concentration Factor

DATA

Load	**p**	**1000**	**N**
Dia(D)	**D**	**80**	**mm**
smalldia(d)	**d**	**40**	**mm**
radius of fillet r		**8**	**mm**
Area			
pi/4*(d^2)		**1256.4**	**N/mm^2**
σmax		**???**	

Take small area

1 σnom L/A

0.795924865 N/mm^2

2 Kt ?

To be obtained from DDB 7.11

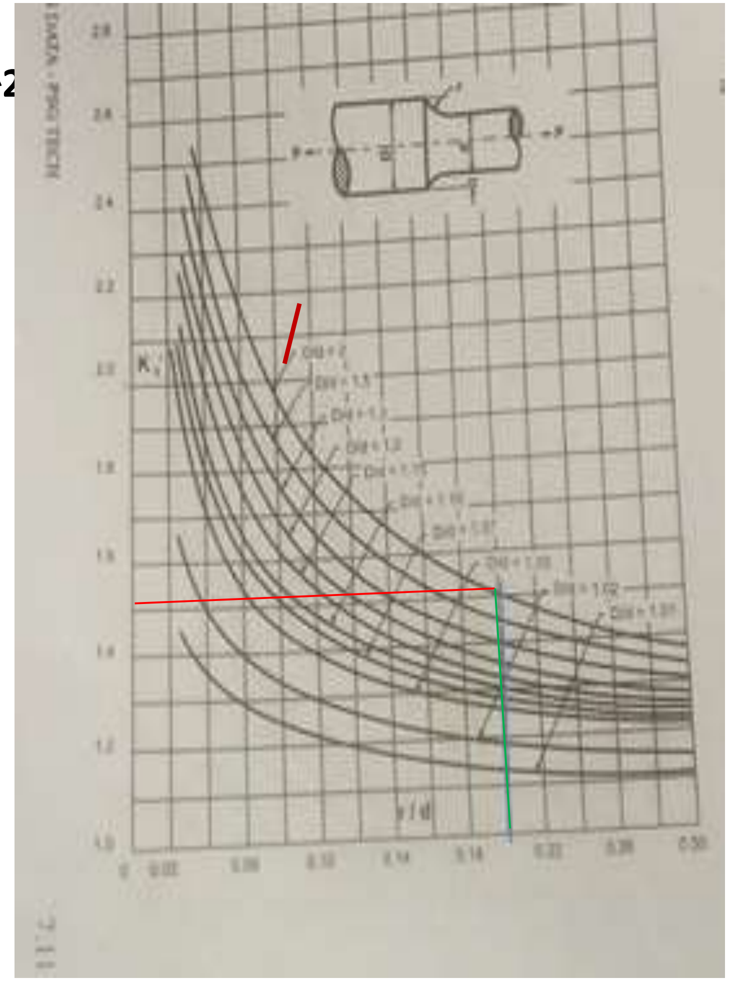
Ratio r/d

0.2

Ratio D/d

2

Kt 1.5

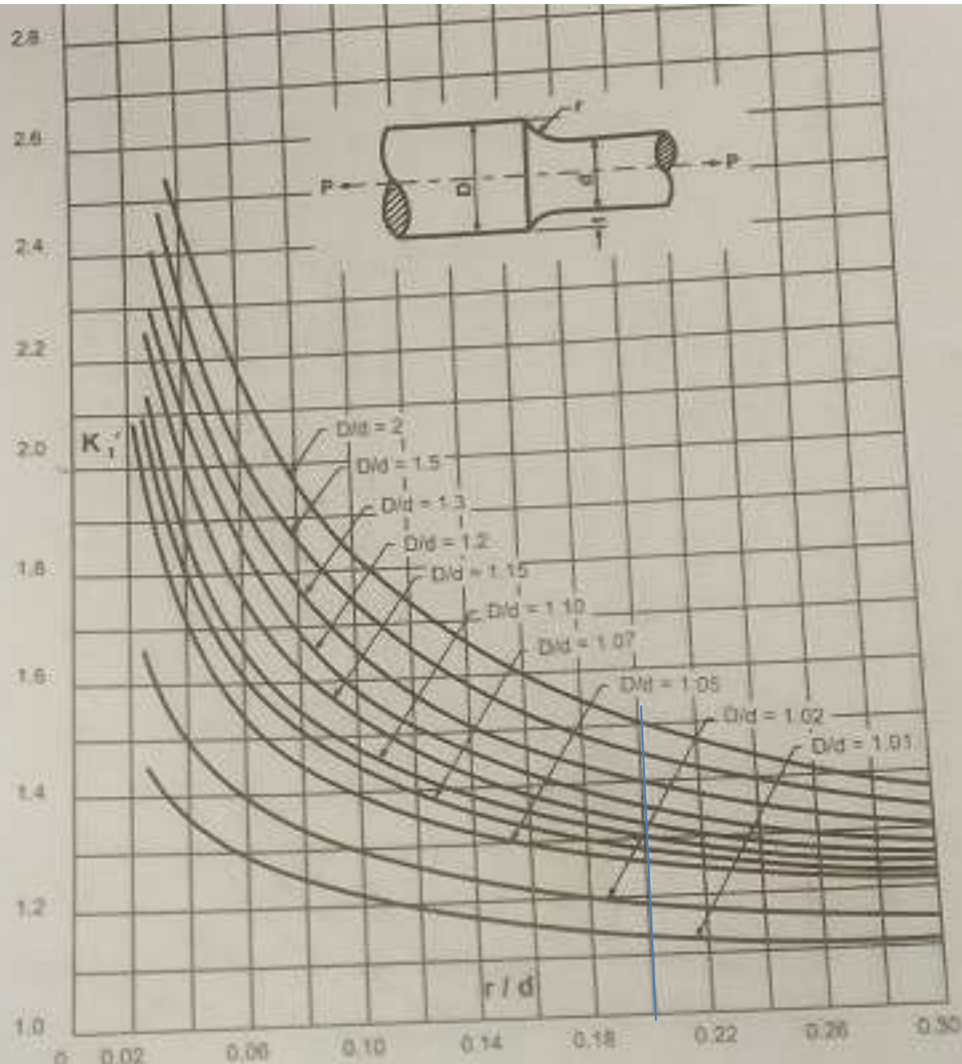


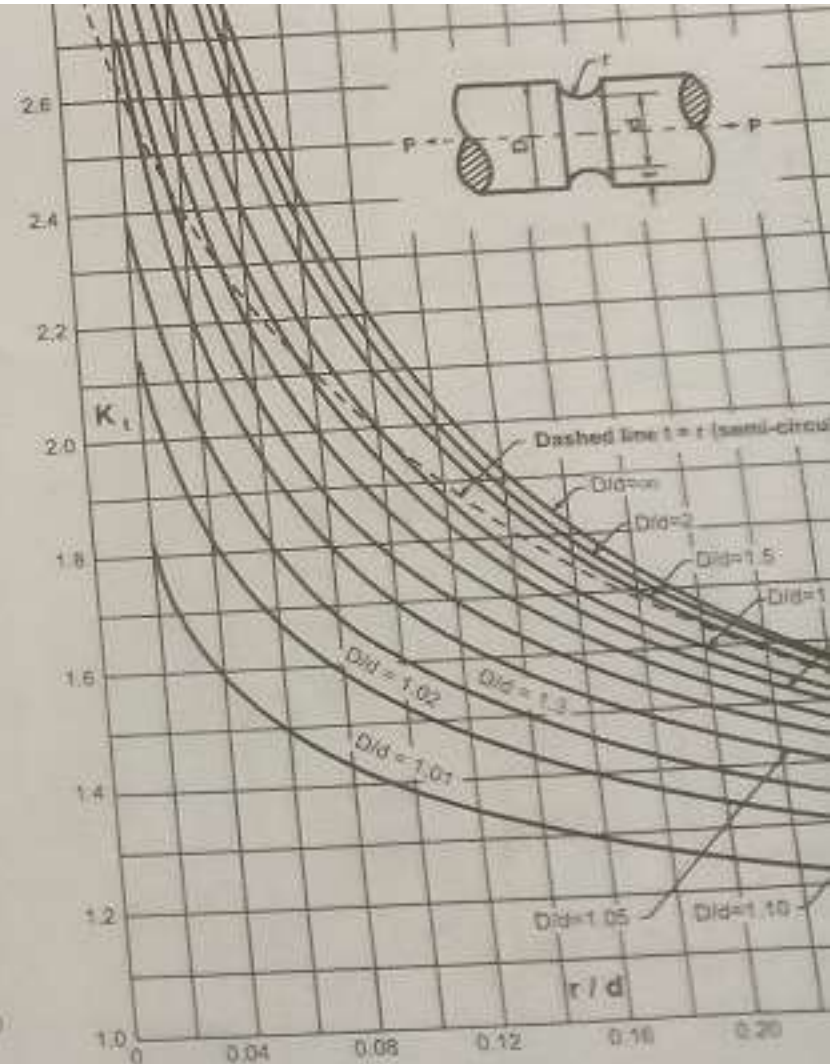
3 σ_{max}

σ_{max} = Kt x σ_{nom}

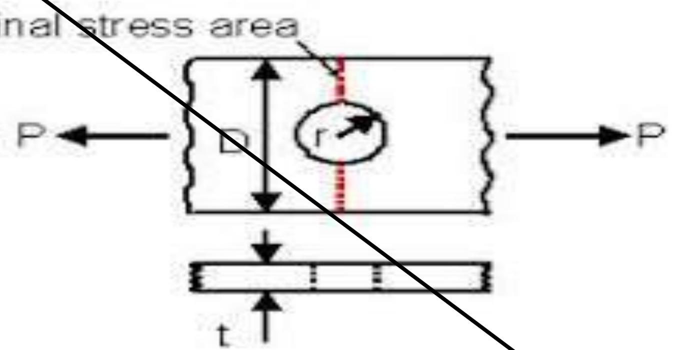
1.194 N/mm^2

Practice by yourself various components in stress concentration


K't
D/d = 2
D/d = 1.5
D/d = 1.3
D/d = 1.2
D/d = 1.15
D/d = 1.10
D/d = 1.07
D/d = 1.05
D/d = 1.02
D/d = 1.01
r / d
P
D
d
r
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8
0
0.02
0.06
0.10
0.14
0.18
0.22
0.26
0.30


Kt
D/d=∞
D/d=2
D/d=1.5
D/d = 1.02
D/d = 1.2
D/d = 1.01
D/d=1.05
D/d=1.10
r / d
P
D
t
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
0
0.04
0.08
0.12
0.16
0.20

1.Determine the stress concentration factor for the plate ubjected to load of 9KN As In figure.



$$\sigma_{nom} := \frac{Force}{Area}$$

Flat Bar Dimensions

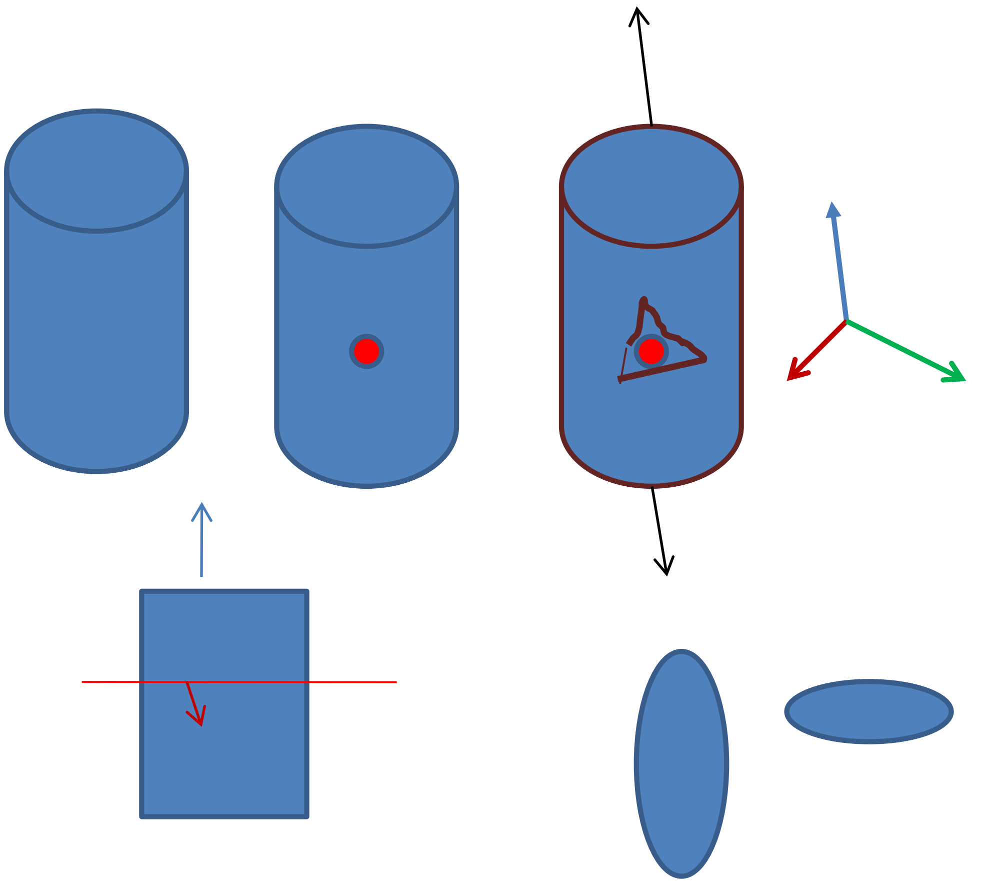
Radius of hole:	$r = 10 \cdot mm$
Width of section:	$D = 100 \cdot mm$
Thickness of plate:	$t = 10 \cdot mm$
Applied load:	$P = 9000 \cdot N$

Stress concentration factor $\quad k_t := 3.00 - 3.13 \cdot \left(\frac{2 \cdot r}{D}\right) + 3.66 \cdot \left(\frac{2 \cdot r}{D}\right)^2 - 1.53 \cdot \left(\frac{2 \cdot r}{D}\right)^3$

$$k_t = 2.508$$

Nominal stress $\quad \sigma_{nom} := \frac{P}{t \cdot (D - 2 \cdot r)} \qquad \sigma_{nom} = 11.25\,MPa$

Peak stress $\quad \sigma_{max} := k_t \cdot \sigma_{nom} \qquad \sigma_{max} = 28.217\,MPa$



THEORIES OF FAILURES

Why ? Need to study

Under combined stress , it is difficult to predict by which stress the failure of member occurred.

Hence the stresses are accounted in terms of principal stresses

This involves the assessment of stresses in biaxial or tri axial stresses.

It is by nature all the engineering components subjcted to those systems stresses

TYPES OF THEORIES OF FAILURES

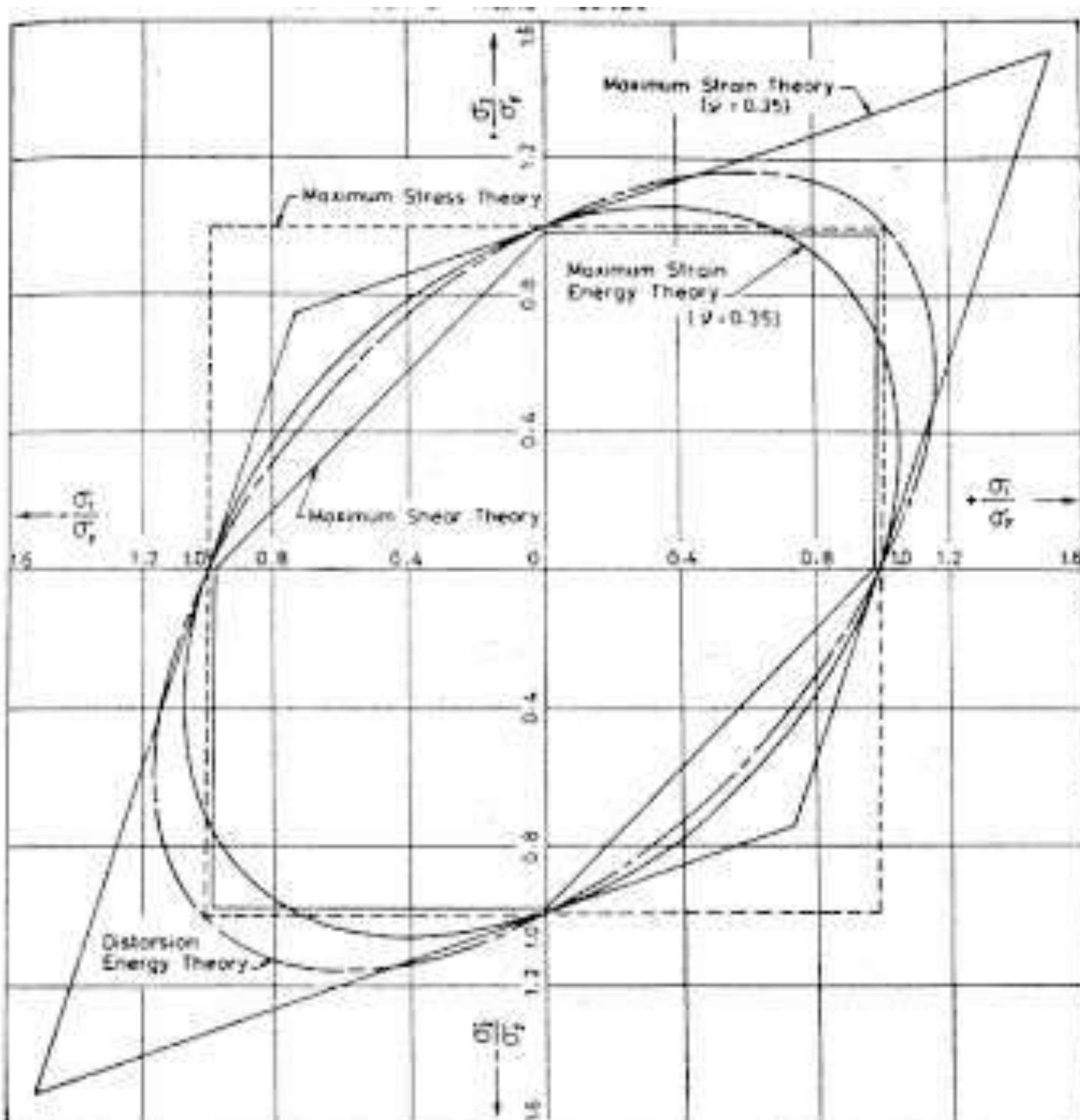
1. **Maximum Principal Stress theory (RANKINE'S THEORY)—(σnormal)**

2. **Maximum Shear Stress theory (GUEST AND TRESCA'S THEORY)--- (τ_{max})**

3. **Maximum Principal Strain theory (St. VENANT'S THEORY)---(e_{max})--- No reliable results- Not followed**

4. **Total Strain Energy theory (HAIGH'S THEORY)**

5. **Maximum Distortion Energy theory (VONMISES AND HENCKY'S THEORY)**



Maximum Strain Theory (ν = 0.35)
Maximum Stress Theory
Maximum Strain Energy Theory (ν = 0.35)
Maximum Shear Theory
Distorsion Energy Theory
σ_1/σ_y
σ_2/σ_y
1.6
1.2
1.0
0.8
0.4
0

How ? The Theories of failure to be defined

Member's **σy** value under
The biaxial or Tri axial systems stresses

=

Standard Tension Test value for ["**σy**"]

Condition is for same material

1.Max. Normal or principal stress Theory

Def:

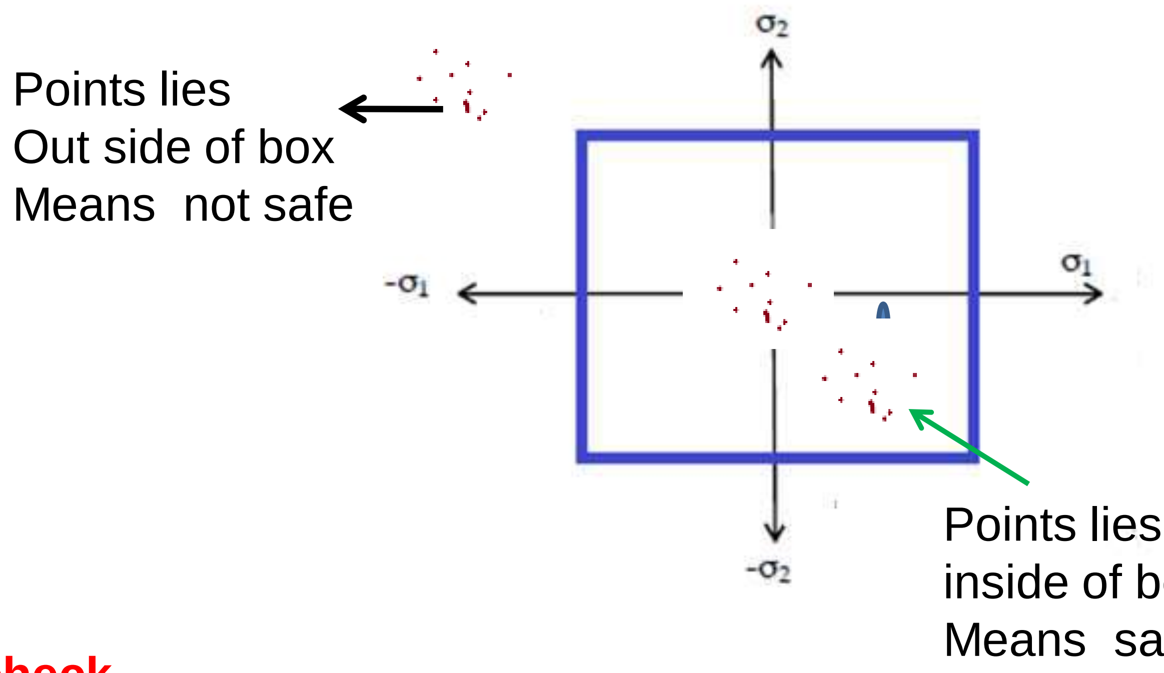
{ Member's **σy** value under The biaxial or Tri axial systems stresses } = { Standard Tension Test value for ["**σy**"] }

Formula

$\sigma 1/ \sigma 2/ \sigma 3 \leq \sigma y$,
Chose max

$\sigma 1/ \sigma 2/ \sigma 3 \leq \sigma y/n$
Chose max

Graphical study



Design check

'σ1/ σy" ratio to be calculated and it lies with in square region

Then the design is safe,
if it fall out side then the design is not safe, then rework to be done

2.Max. Shear stress Theory --Ʈmax

Def:

{ Member's **Ʈy** value under The biaxial or Tri axial systems stresses } = { Standard Tension Test value for ["**Ʈy**"] }

Formula

$$(\sigma 1- \sigma 2) /(\sigma 2- \sigma 3) /(\sigma 3- \sigma 1) \leq \sigma y/2 ,$$

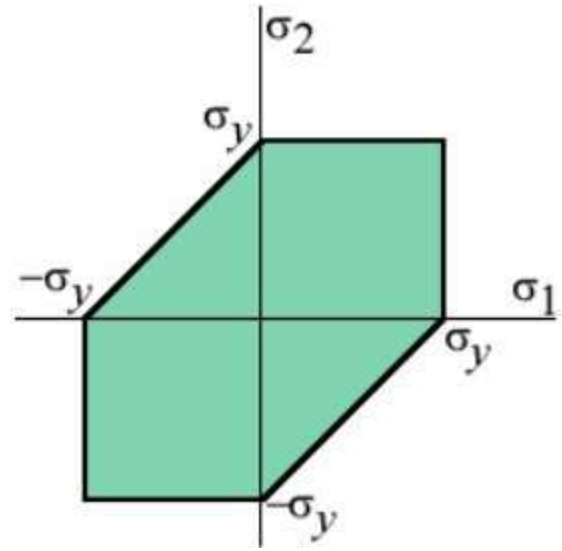
Chose max

$$(\sigma 1- \sigma 2) /(\sigma 2- \sigma 3) /(\sigma 3- \sigma 1) \leq \sigma y/2n$$

Chose max

As per max shear stress theory

$$\tau_{max,simp} = \frac{\sigma_y}{2}$$



Where **"σy"** is the material yield value

It is suitable for ductile material and shaft design always use it

Design check

- **'σ1/ σy" ratio to be calculated and it lies with in shaded/coloured region**
- **Then the design is safe,**
- **if it fall out side then the design is not safe, then rework to be done**

3.Max. Strain Theory – e_{max}

Def:

$$\left\{ \begin{array}{l} \text{Member's } e_y \text{ value under} \\ \text{The biaxial or Tri axial} \\ \text{systems stresses} \end{array} \right\} = \left\{ \begin{array}{l} \text{Standard Tension Test value} \\ \text{for [“}e_y\text{”]} \end{array} \right\}$$

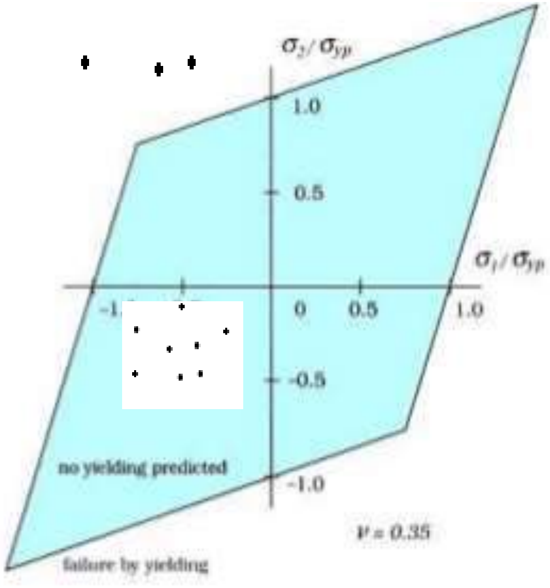
Formula

Maximum strain theory (St. Venant's)

$$\left. \begin{array}{r} \sigma_1 - \nu\,(\sigma_2+\sigma_3) \text{ or } \sigma_2 - \nu\,(\sigma_3+\sigma_1) \\ \text{or } \sigma_3 - \nu\,(\sigma_1+\sigma_2) \text{ whichever is maximum} \end{array} \right\} = \sigma_Y$$

As this theory is not providing
Reliable results,

Not being/ recommended used in
the design



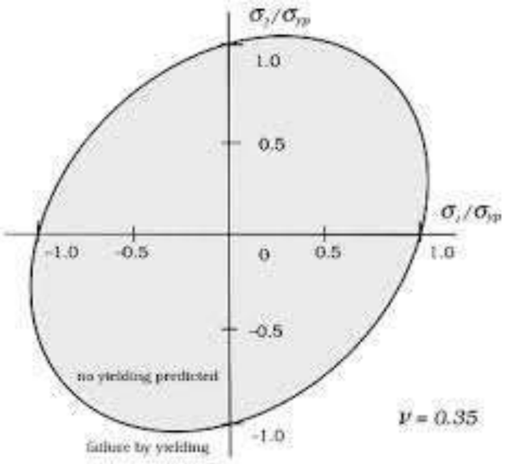
4.Max. Strain EnergyTheory

Def:

{ Member's energy stored/unit volume value (U) under the biaxial or Tri axial systems stresses } = { Standard Tension Test value for limiting strain energy "U_y"] }

Formula

Poisson ratio



It holds good results for ductile materials

Design check

'σ1/ σy" ratio to be calculated and it lies with in square region

Then the design is safe,
if it fall out side then the design is not safe, then rework to be done

5.Max. Distortion Strain Energy Theory

Def:

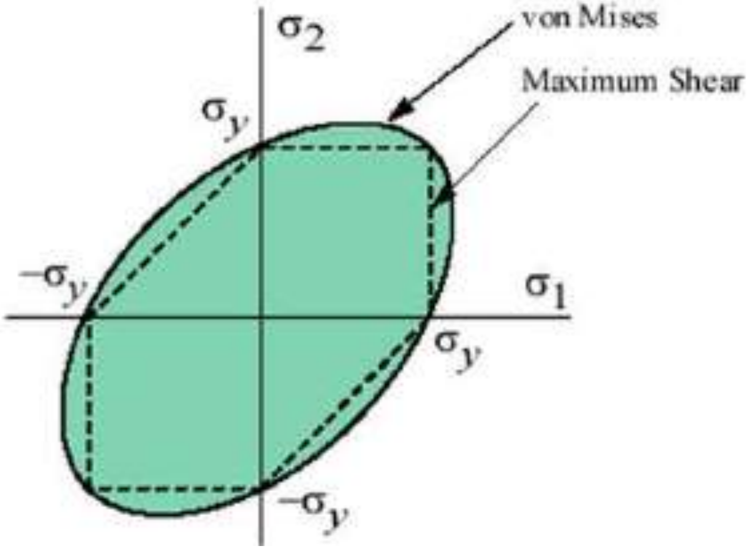
{ Member's energy stored/unit volume value under the biaxial or Tri axial systems stresses (U) } = { Standard Tension Test value for limiting strain energy ["**U y**"] }

Formula

Maximum strain energy theory

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

It holds good results for ductile materials



Dist. Strain energy=
strain energy- strain energy by stress

Design check

‘σ1/ σy” ratio to be calculated and it lies with in region

Then the design is safe,
if it fall out side then the design is not safe, then rework to be done

Problems Objectives

1. **Find the size of the member**
2. **Find the factor of safety**

A member is having its principal stresses as follows σ_1= 190 Mpa, σ_2= 90Mpa and σ_3 =0 find the factor of safety for the member using theories of failures. Take material of C45 has 360 Mpa,

σ_1 =	**190**
σ_2 =	**90**
σ_y	**360**
$\sigma_1-\sigma_2$	**100**
$\sigma_2-\sigma_3$	**90**
$\sigma_3-\sigma_1$	**-190**
1n	**1.894**
2n	**1.8**

3n	2.27	Max. Strain Theory
4n^2	4.021098	Max. Strain energy Theory
n	2.005	
5n^2	4.782288	Max. Distoriton strain energy Theory
n	2.19	

STEPS to solve for theories of failure

Step1: Identify the stresses available

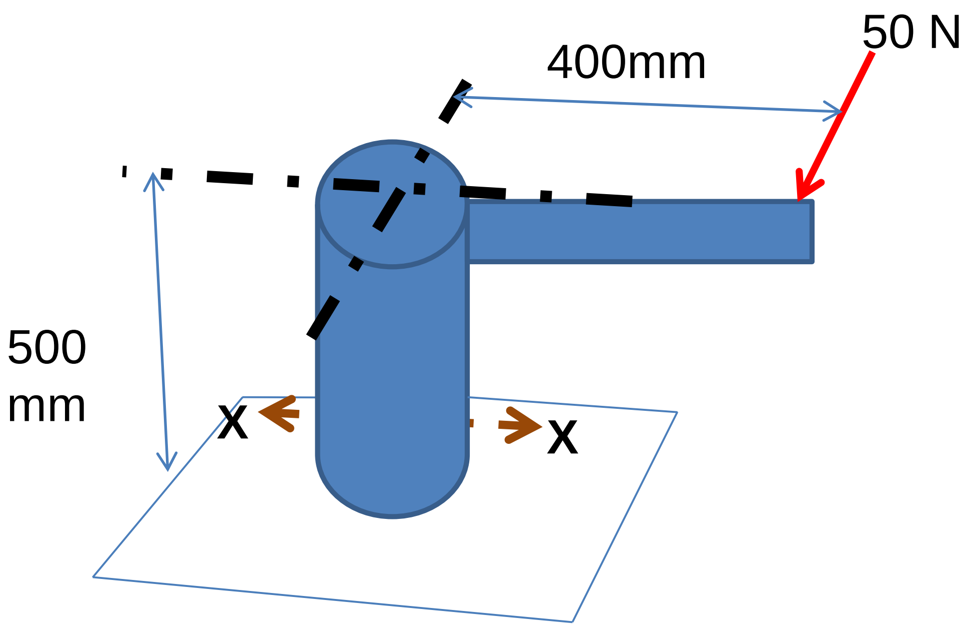
Step2: Directional assignments the stresses available

Step3 : Calculate principal stresses

Step4: Applying theories of failures

& find the objective

Problem 2

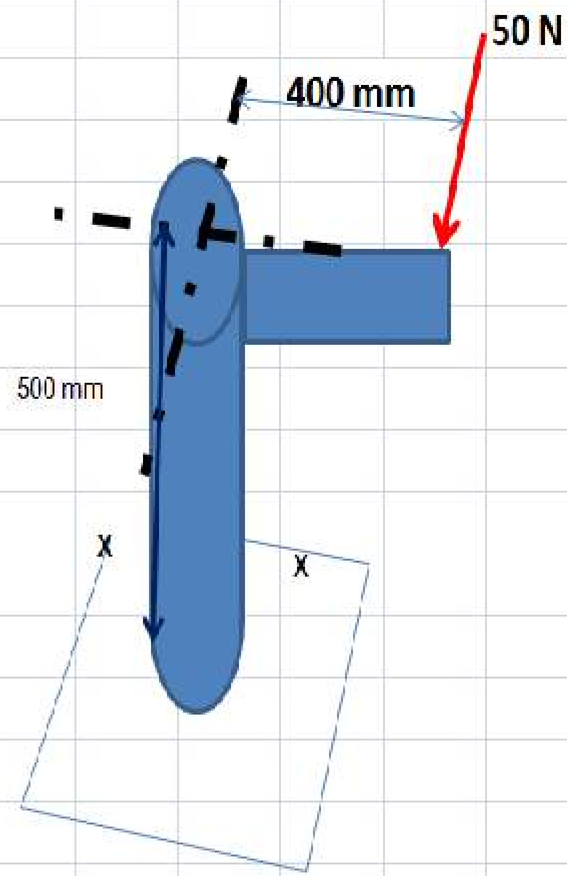


Determine diameter at Section XX, for the member shown in the figure. Find the diameter of the member Using theories of failure. Take factor of safety n=2.5,

2

DATA

D	???
twisting moment distance	400 mm
Bending Moment	500 mm
n- factor of safety	2.5



First

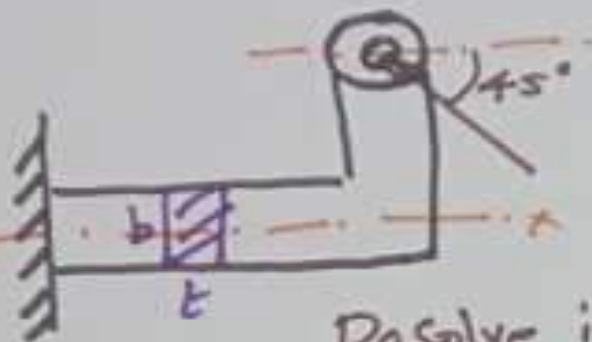
calculate σ1, σ2 (It should be done bt directional assignment of bending = σy, Ţ, but σd = 0)

Apply to theories of failure

1	Caculation of σ1 & σ2		
	Data		
	diameter	x	mm	
	Load	50	N	
	eh	400	mm	(Horizontal to original load)
	ev	500	mm	(Vertical to introduced load from the base)
	area			
	Pi	3.141		
	A	0.78525	d^2	sq.mm
	Direct stress			
1	Stress	0	N/sq.mm	
	σx	0		

2	Bending				
	σb=	M/Z			
	M	Load x e			
		25000	N-mm		
	z =	I/y			
		I=	π*d^4/64		
		Y=	d/2		
		I =	0.0491	d^4	mm^4
		y =	0.5	d	mm
	therfore	z =	0.09815625	d^3	mm^3
	Now	σb=	2.547E+05	*(1/d^3)	N/sq.mm
		σy	2.547E+05	*(1/d^3)	

3	Shear stress		T x16/(pi x d*d*d)			refer Pg.No 7.1/DDB
		T=	20000	N mm		
			(eccentrical distance is horizontal distance from axis)			
	τ Shear stress	τ	101878.3827	*(1/d^3)	N/sq.mm	

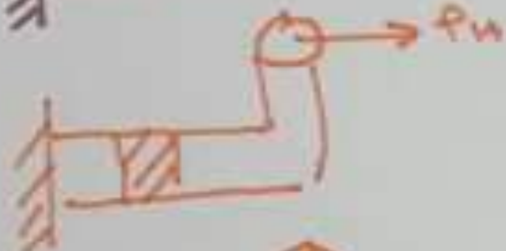


With respect to x–x axis.

Resolve it.

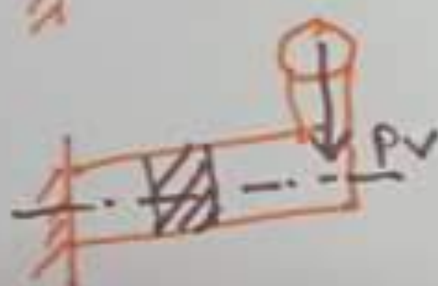
$P_H = \cos 45° \cdot P$

$P_V = \sin 45° \cdot P$



Only Bending

$\sigma_{b_H} = \frac{M_H}{Z}$ (1)

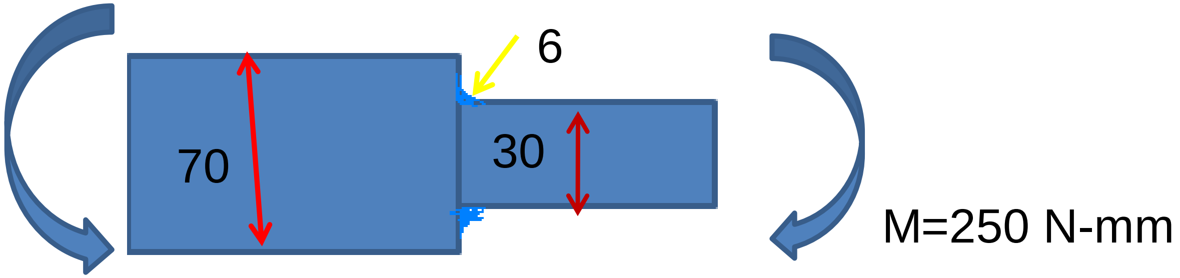


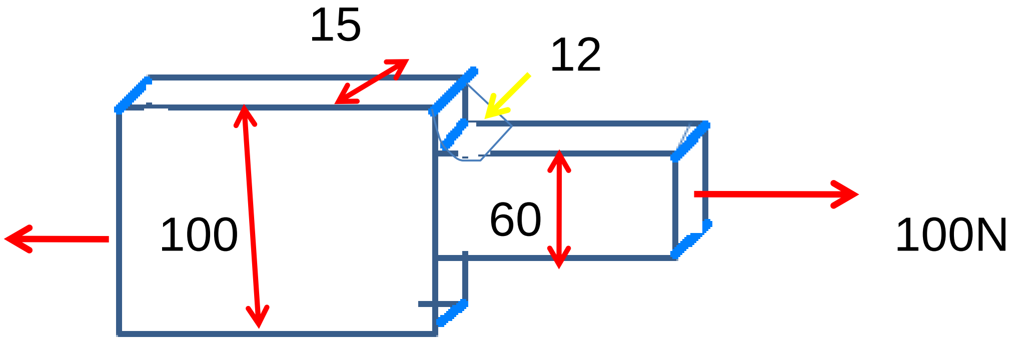
– $\sigma_d = \frac{P_v}{A}$ ①

→ $\sigma_{b_v} = \frac{M_v}{Z}$ ②

$\sigma_{com} = \sigma_{b_H} + \sigma_{b_V} + \sigma_d$

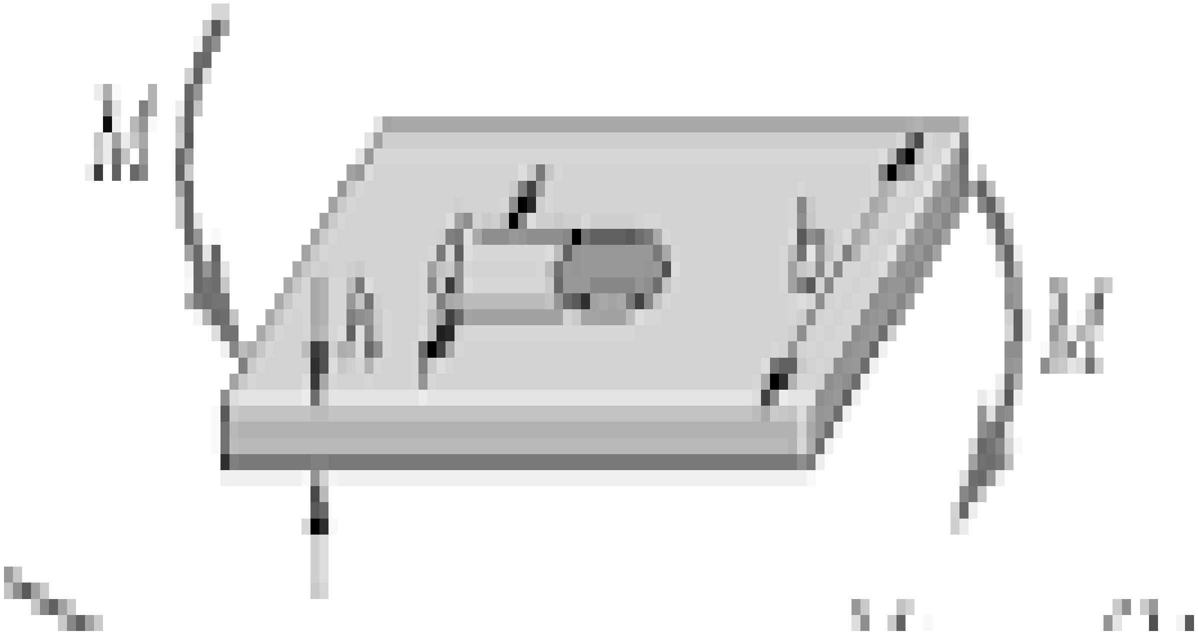
Tutorial 2 week2 20 marks credit





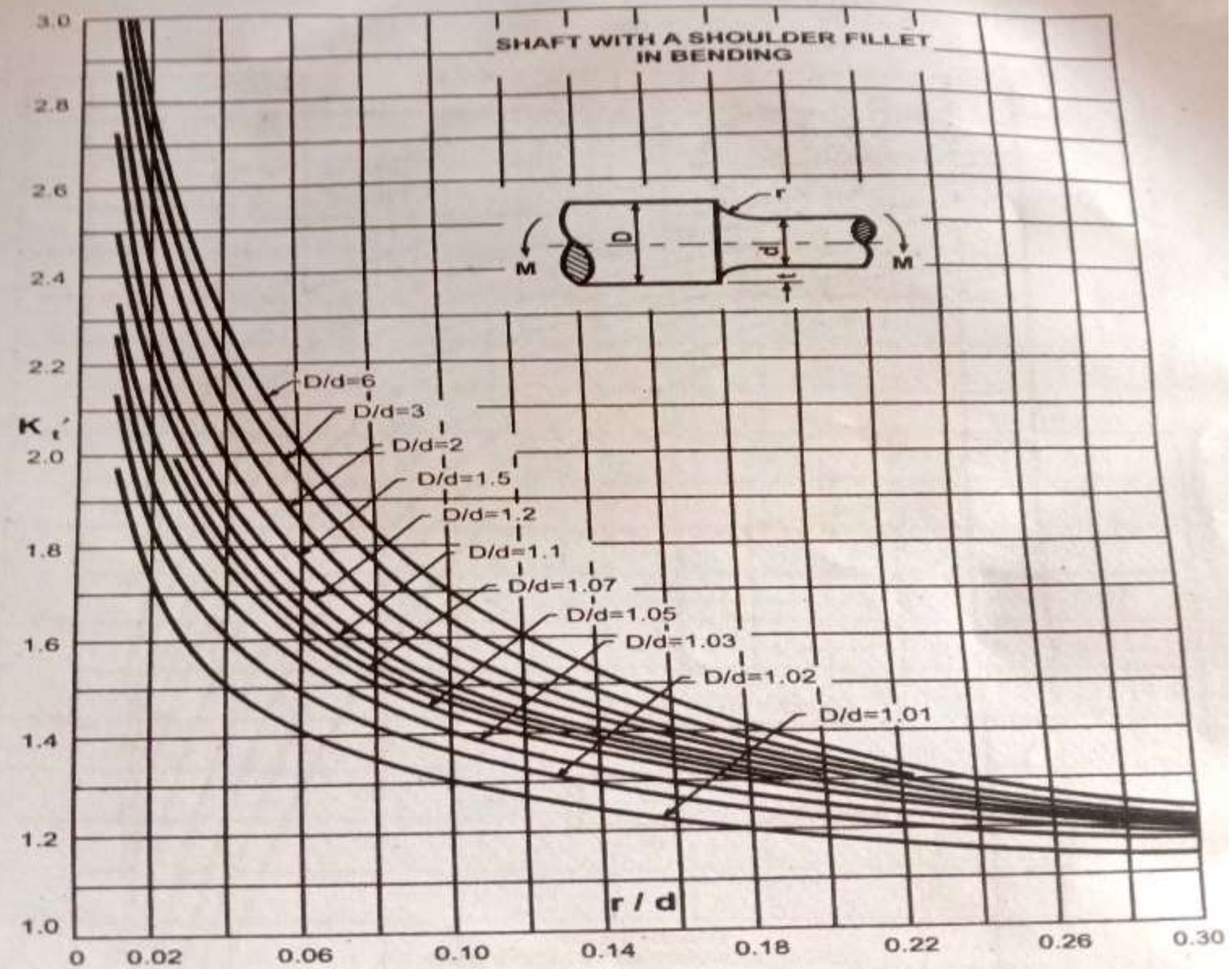
Determine the maximum stress induced in the members, all dimensions are in mm

Tutorial 2 week2 20 marks credit

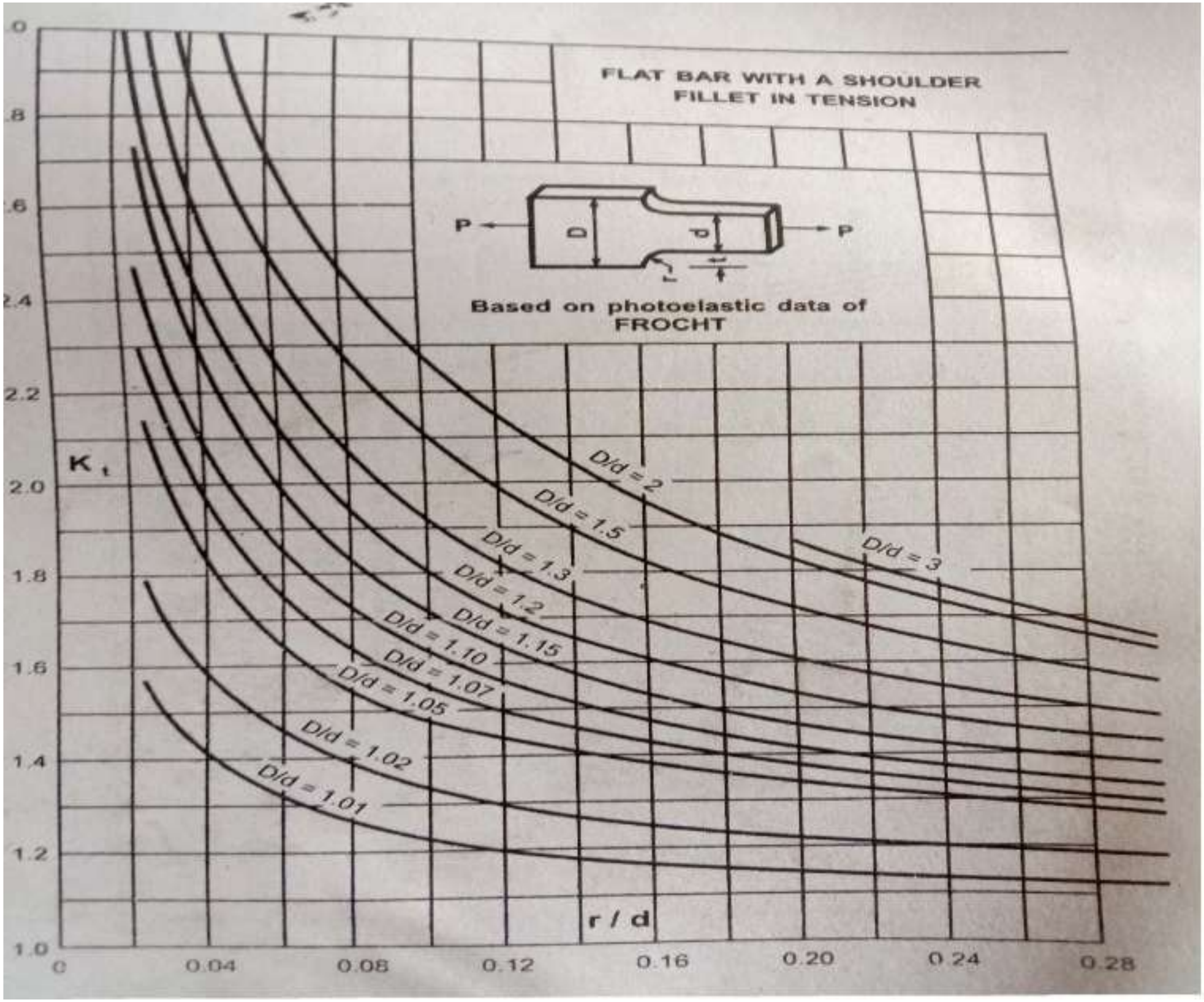


M=150 N-mm

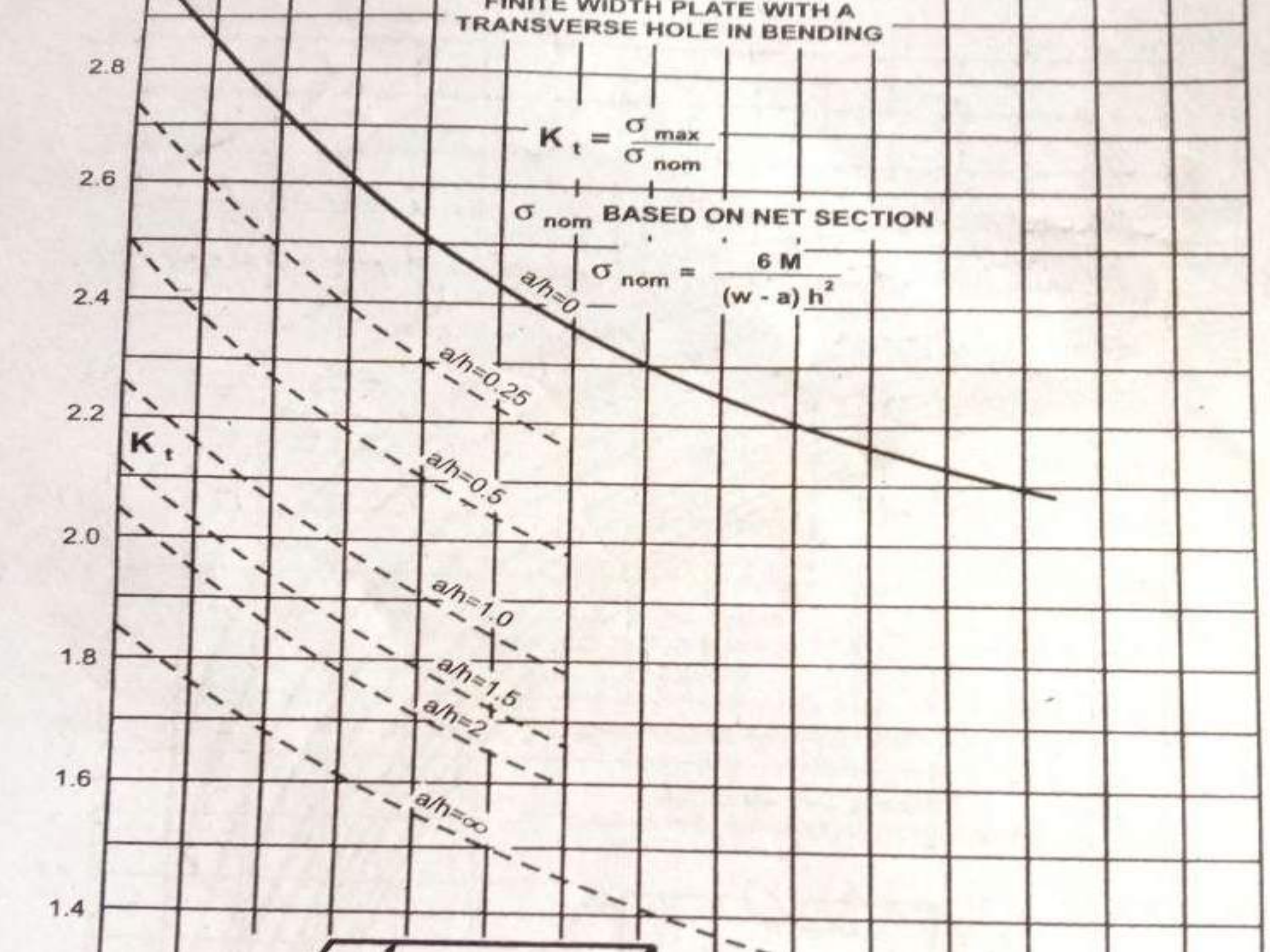
Width = 100, a = 40 , t = 20 in mm



SHAFT WITH A SHOULDER FILLET
IN BENDING
r
D
d
t
M
M
D/d=6
D/d=3
D/d=2
D/d=1.5
D/d=1.2
D/d=1.1
D/d=1.07
D/d=1.05
D/d=1.03
D/d=1.02
D/d=1.01
K_t'
3.0
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0
0.02
0.06
0.10
0.14
0.18
0.22
0.26
0.30
r / d



FLAT BAR WITH A SHOULDER FILLET IN TENSION
Based on photoelastic data of FROCHT
P
D
d
r
P
D/d = 2
D/d = 1.5
D/d = 1.3
D/d = 1.2
D/d = 1.15
D/d = 1.10
D/d = 1.07
D/d = 1.05
D/d = 1.02
D/d = 1.01
D/d = 3
K t
r / d
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
.6
.8
.0
0
0.04
0.08
0.12
0.16
0.20
0.24
0.28


FINITE WIDTH PLATE WITH A
TRANSVERSE HOLE IN BENDING
$K_t = \frac{\sigma_{max}}{\sigma_{nom}}$
σ_{nom} BASED ON NET SECTION
$\sigma_{nom} = \frac{6M}{(w-a)h^2}$
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
K_t
a/h=0
a/h=0.25
a/h=0.5
a/h=1.0
a/h=1.5
a/h=2
a/h=∞

Principal stress		
σ1, σ2=	??	
σx + σy	2.547E+05	*(1/d^3)N/sq.mm
σx - σy	-2.547E+05	*(1/d^3)N/sq.mm
$(\sigma x - \sigma y)^2$	64870030360.2	*(1/d^3)^2N/sq.mm
(σx + σy)/2	1.273E+05	*(1/d^3)N/sq.mm
Ʈ	101878.3827	*(1/d^3)
$Ʈ^2$	10379204857.6	*(1/d^3)^2
(√ (σx – σy)2 +4Ʈ2/	163084.985	*(1/d^3)

Now		
σ1 =	2.9043296E+05	*(1/d^3)N/sq.mm
σ2 =	-35737.007	*(1/d^3)N/sq.mm

Apply the theories of failure

1. Max.Normal Stess Theory

Ref. Pg.7.3/DDB

$$\sigma 1 / \sigma 2 / \sigma 3 \leq \sigma y/n$$

(σ3 struck through, marked 0)

chose max value

$\sigma 1 = \sigma y/n$ (1)

n	2.5	
Select material C45 ref. Pg.1.9/DDB		(ASSUMED)
σy= 360		N/mm^2
Use the equation 1		
2.9043296E+05	360/2.5	
2.9043296E+05	144	
2.0168956E+03	,=	d^3
d ,=12.63mm		

Now to check the design safe

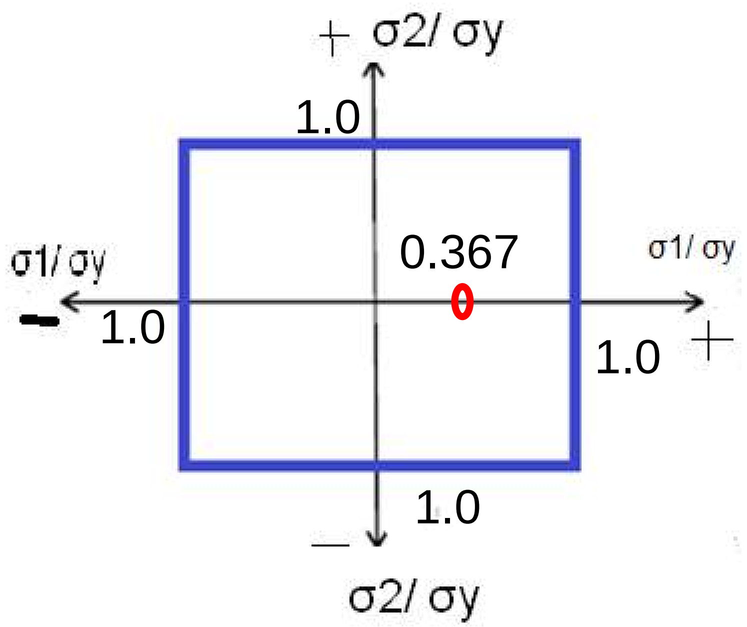
Theory used is Max .Normal Stress Theory

Check for safe

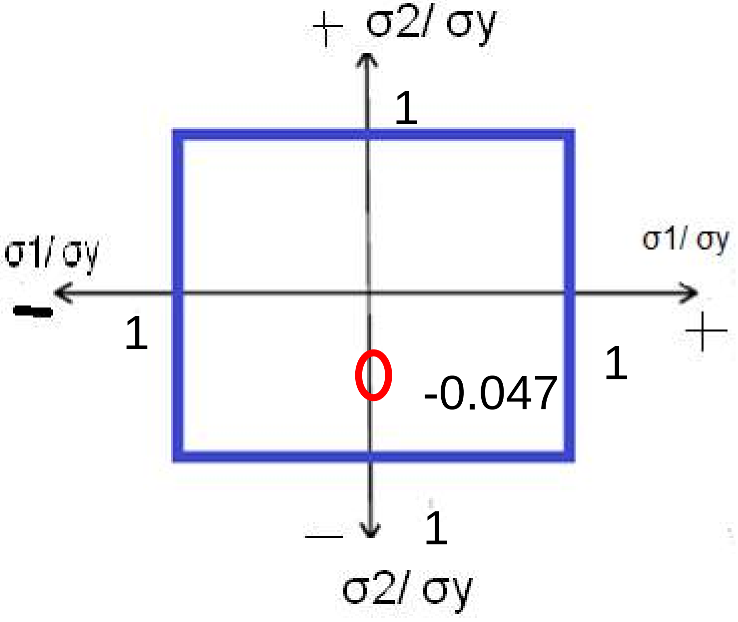
σ1 =2.9043296E+05 *(1/d^3)N/sq.mm

substitute d= 13 mm (rounded off)

σ1 =	**1.3219525E+02**	**N /Sq.mm**
σy=	**360**	**N/Sq.mm**
σ1/σy	**3.7E-01**	**(0.367)**



As the ratio point lie in the region, **Design is safe** σ2/σy = -0.047



2Max Shear Stress Theory

Ref pg.7.3/DDB

σ1-σ2/σ2-σ3/σ3-σ1

= σy/2n

σy= 360, n=2.5

σ1-σ2=	**3.2616997E+05*(1/d^3)**
σ2-σ3=	**-35737.007*(1/d^3)**
σ3-σ1=	**-2.9043296E+05*(1/d^3)**

Chose max, σ1-σ2

3.261699x 10^5/d^3=	**72**
d^3	**4.5301385E+03**
d=	**16.54mm**

Similar way find the diameter by the other theories also check for the design safe

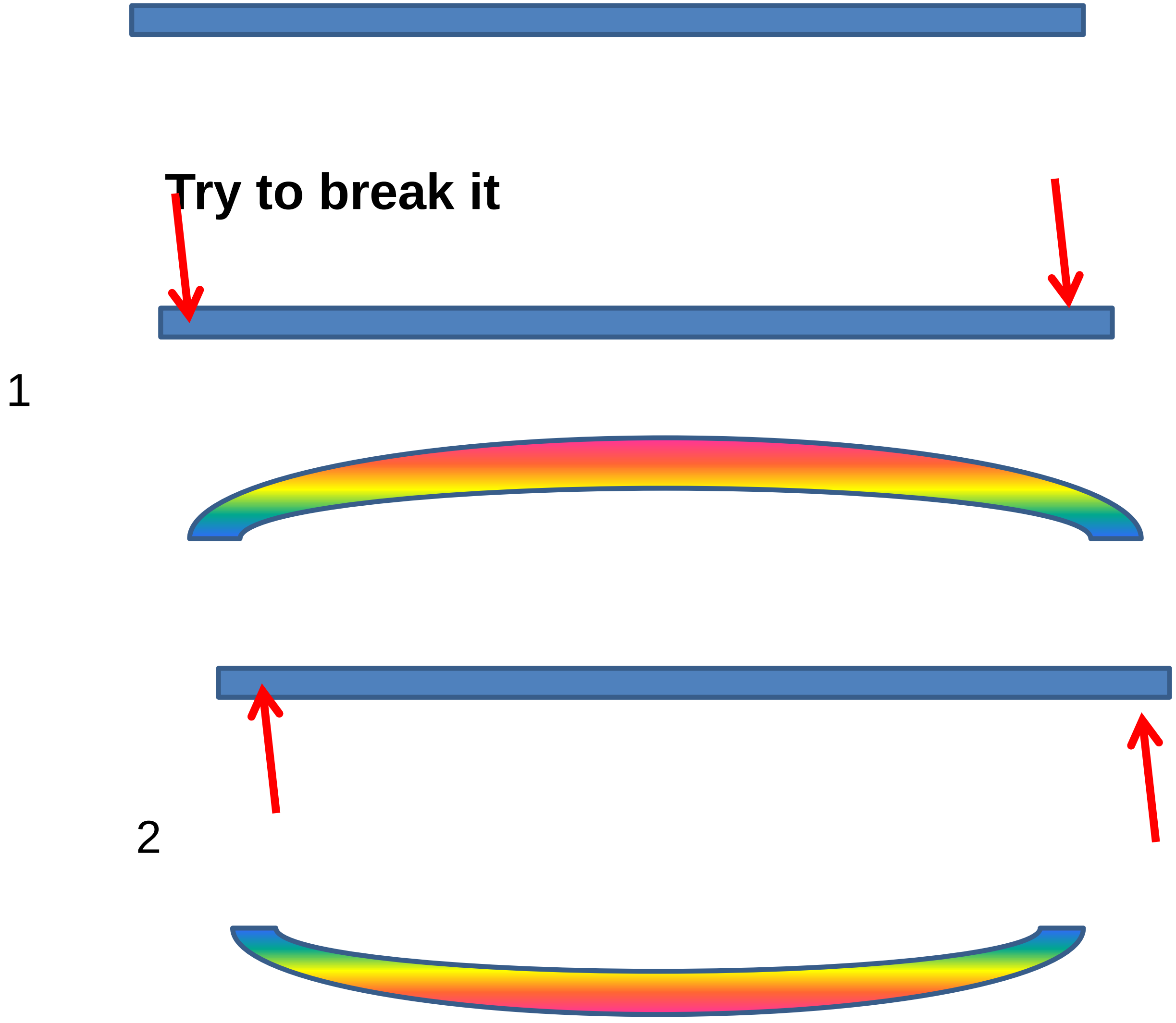
Dynamic Design

Fatigue & Endurance strength
SN – Curve

Theories
Gerber equation
Goodman equation
Soderberg Equation

Solving the problem

To understand cyclic stress



Cyclic Stresses

General terms

ref.Pg.7.6/DDB

FATIGUE

Cyclic Stresses

Parameters used to characterize the fluctuating stress cycle:

1. Mean Stress (σ_m):

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

2. Range of Stress (σ_r) :

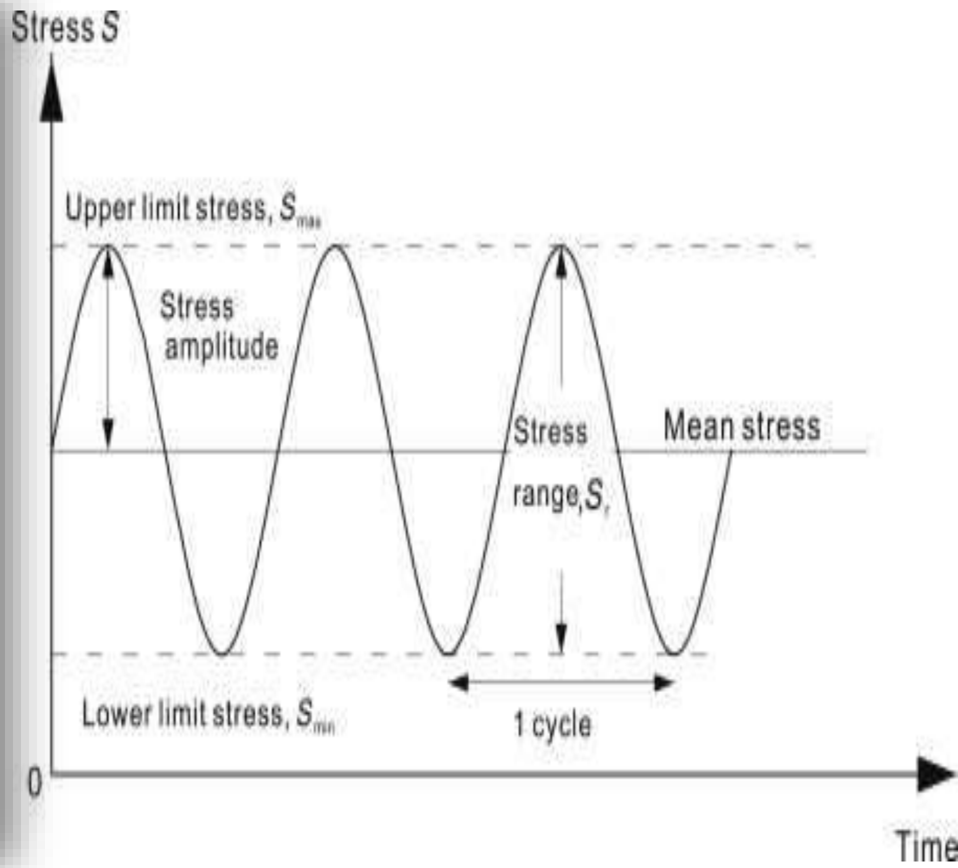
$$\sigma_r = \sigma_{max} - \sigma_{min}$$

3. Stress Amplitude (σ_a) :

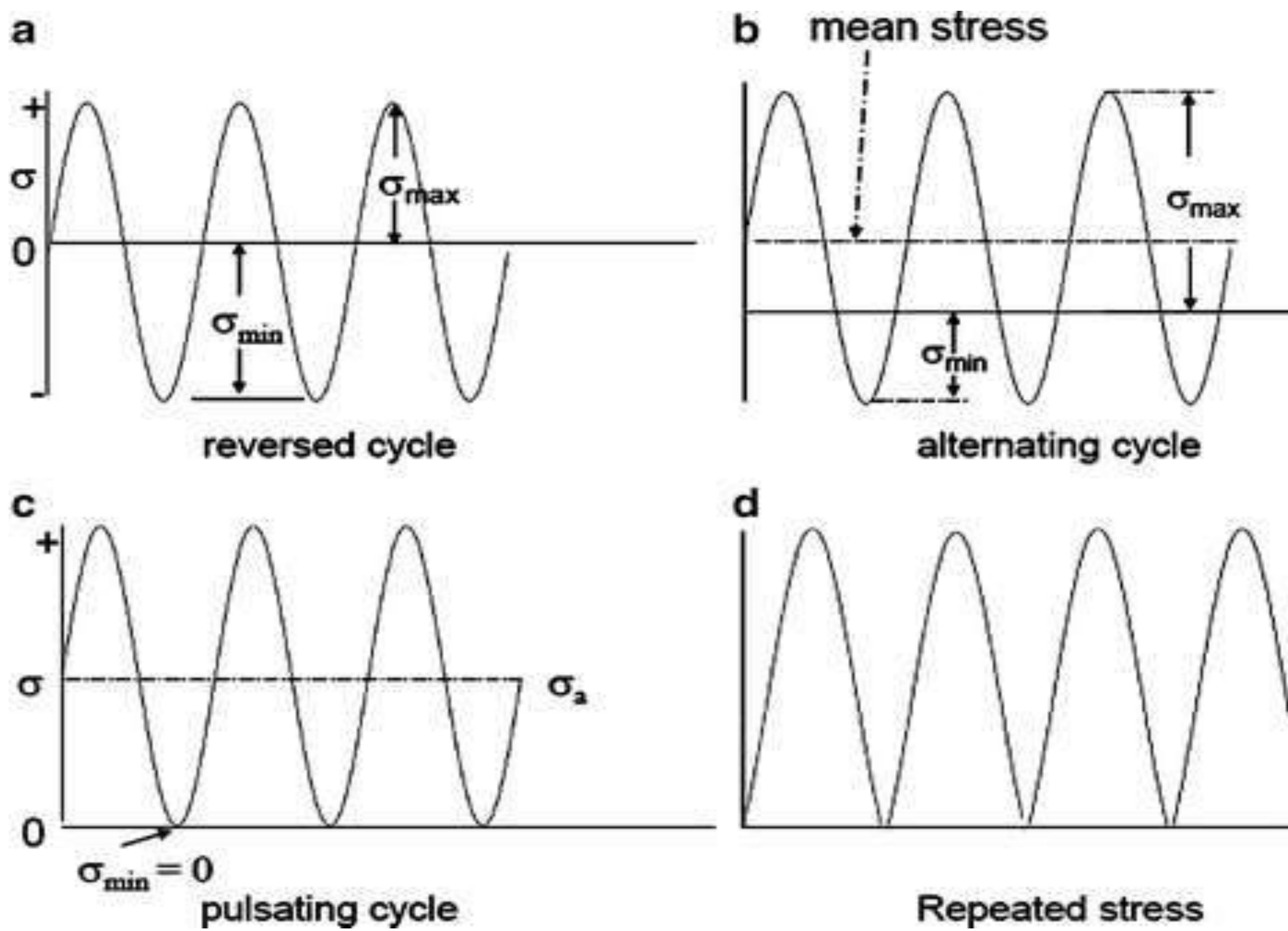
$$\sigma_a = \frac{\sigma_r}{2} = \frac{\sigma_{max} - \sigma_{min}}{2}$$

4. Stress Ratio (R) :

$$R = \frac{\sigma_{min}}{\sigma_{max}}$$



Cyclic stresses



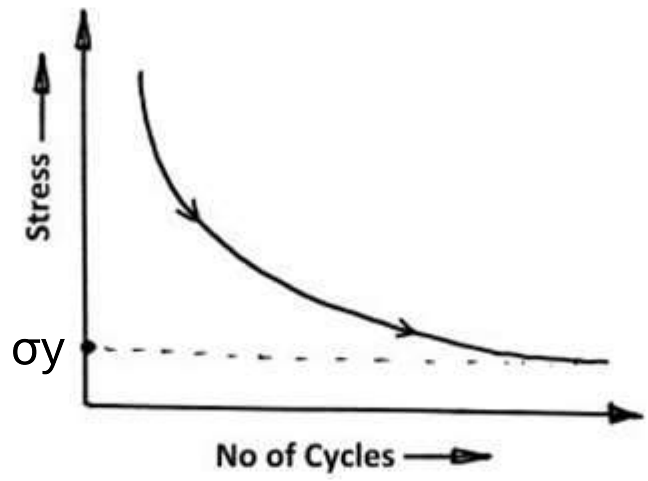
What is fatigue?

When a member subject to cyclic stresses , it fails below its yield stress value known as “fatigue”.

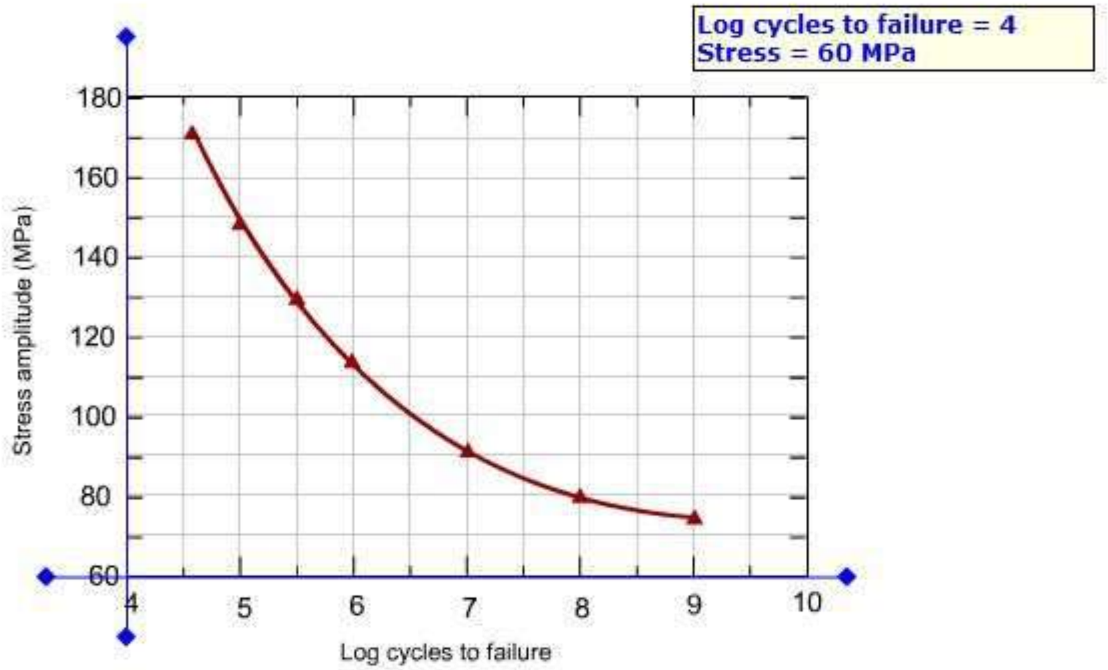
It is unpredictable, - its so dangerous. It occurs due to flaws in the member(under micro study), which initiates the crack formation, then propagation and finally fracture.

Endurance or fatigue limit?

A member will undergo n numbers of cycles. With out failure/fracture at a particular stress value. This stress is known as Endurance or fatigue limit.



S-N diagram – example to predict the stress values With respect to cycles



Fatigue stress concentration factor K_f

Def:

Experimental definition only available

Fatigue stress concentration factor,

$$K_f = \frac{\text{Endurance limit without stress concentration}}{\text{Endurance limit with stress concentration}}$$

Notch sensitivity "q "

Def:

It is defined as the degree of attaining the Kt- theoretical stress concentration. q is estimated with some experimental curves. No extensive data available.

Ref. Pg.no:7.8/DDB

Notch sensitivity q index *is defined by*

$$q = \frac{K_f - 1}{K_t - 1} \quad \text{or} \quad q_{shear} = \frac{K_{fs} - 1}{K_{ts} - 1}$$

→ Kf=1+q(kt-1)

STRESS CONCENTRATION FACTORS

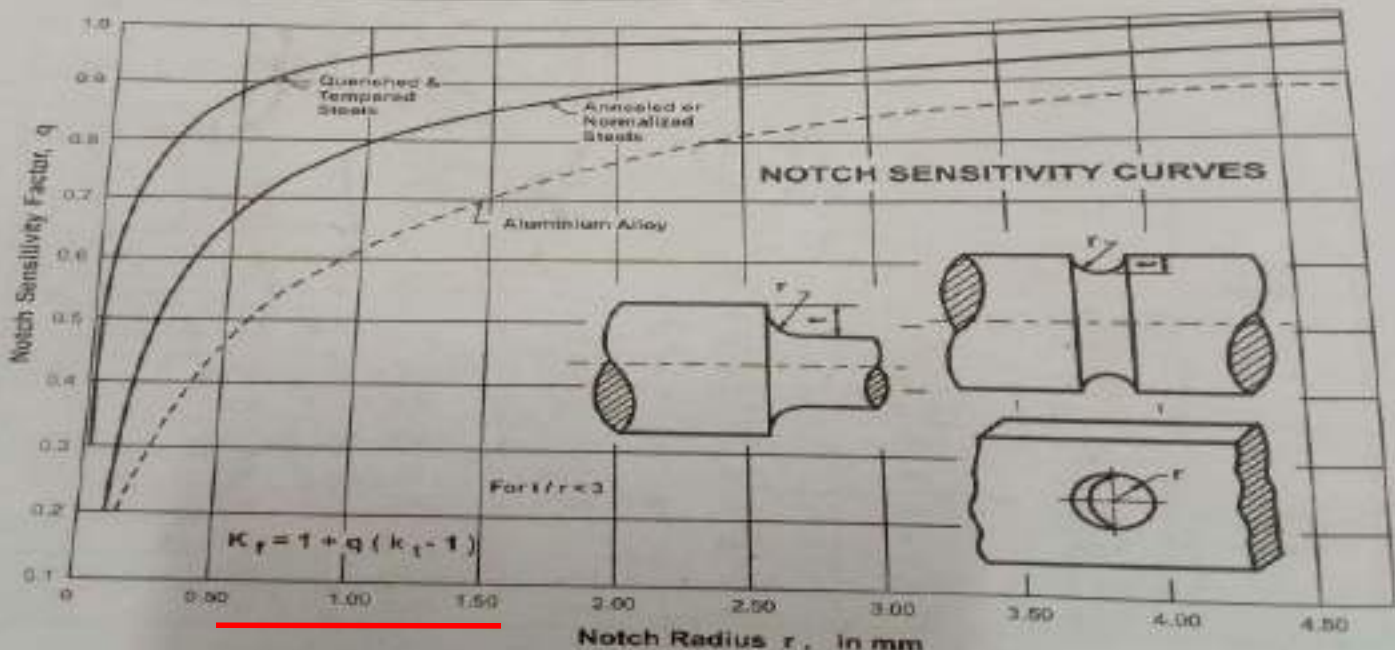
Stress concentration factor } K_t = $\frac{\text{maximum stress}}{\text{nominal stress at the net section}}$

K_t' = combined stress concentration factor, taking into account biaxial or triaxial stresses and Von Mises - Hencky theory.

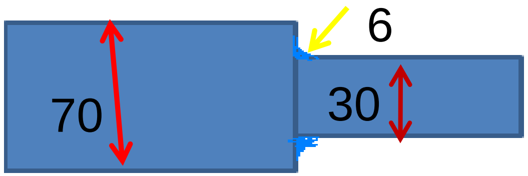
K_t is always greater than K_t', therefore where information is lacking or doubtful, K_t should be used, to be on the safe side.

VALUES OF K_t FOR KEYWAYS

KIND OF KEYWAY	Annealed		Hardened	
	Bending	Torsion	Bending	Torsion
PROFILE	1.6	1.3	2.0	1.6
SLED RUNNER	1.3	1.3	1.6	1.6



Problem for finding "Kf"



Find the Kf of the member which is made of steel and annealed.

SOLUTION

The equation,

$$Kf=1+q(kt-1)$$

Ref. Pg.no:7.8/DDB

1. Kt =?
2. q = ?

Ref. Pg.no:7.11/DDB

r/d=0.2,
D/d = 2.33(2.0)

Kt=1.5

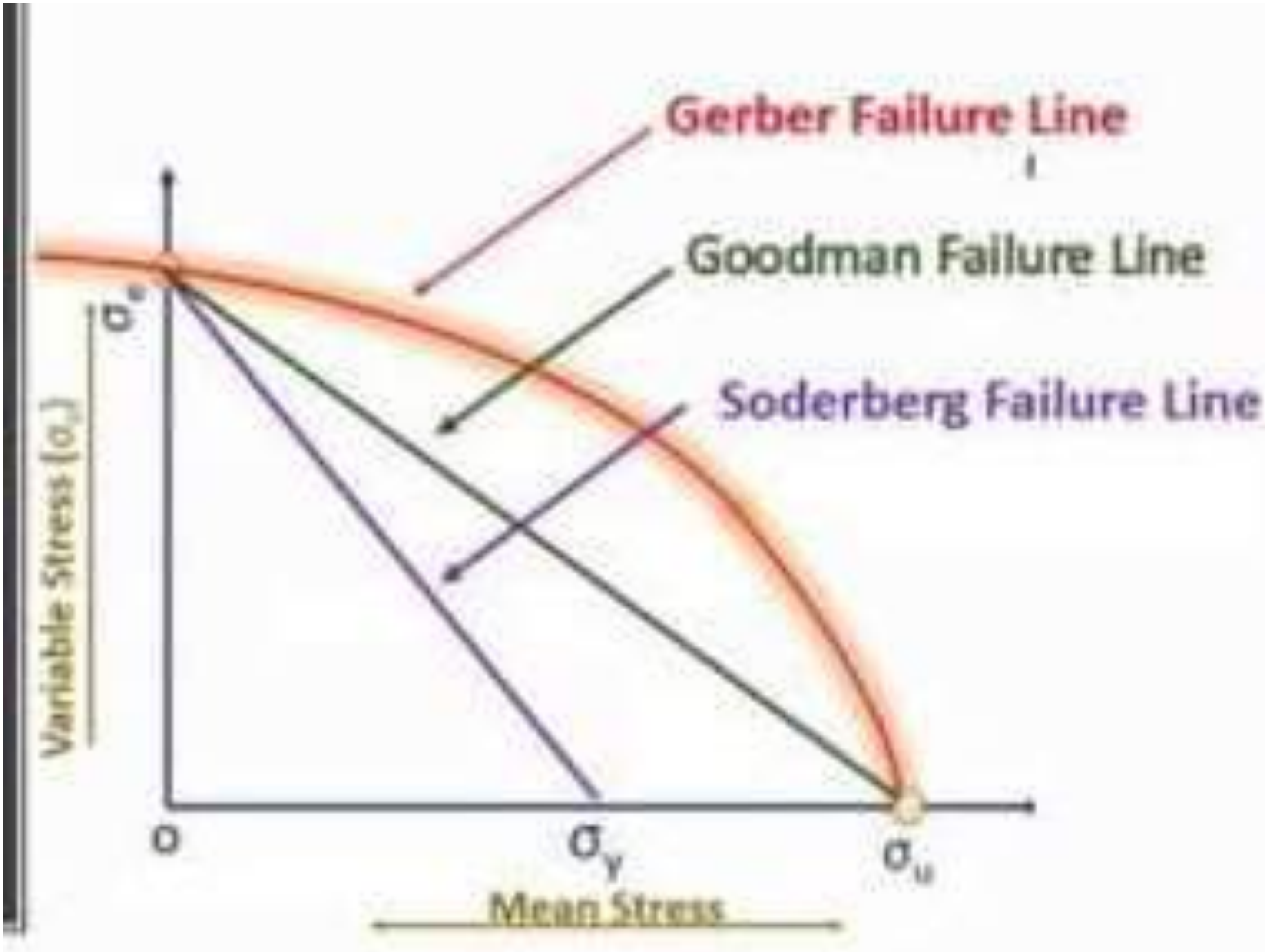
Ref. Pg.no:7.8/DDB

r = 6 (4.5), material steel & annealed
q = 0.95

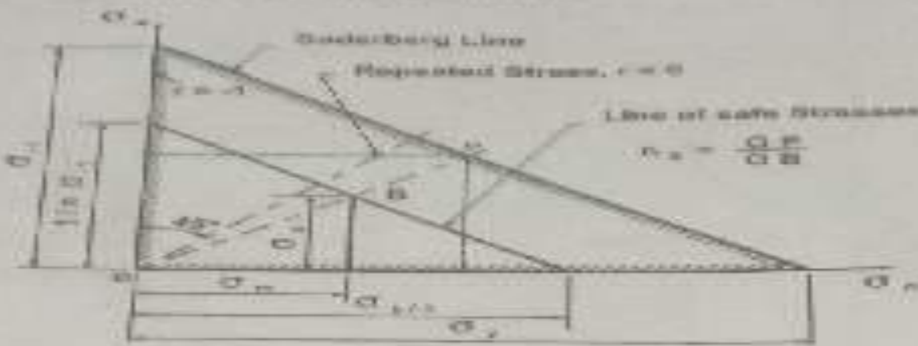
Now **Kf= 1.475**

Theories for Varying stresses

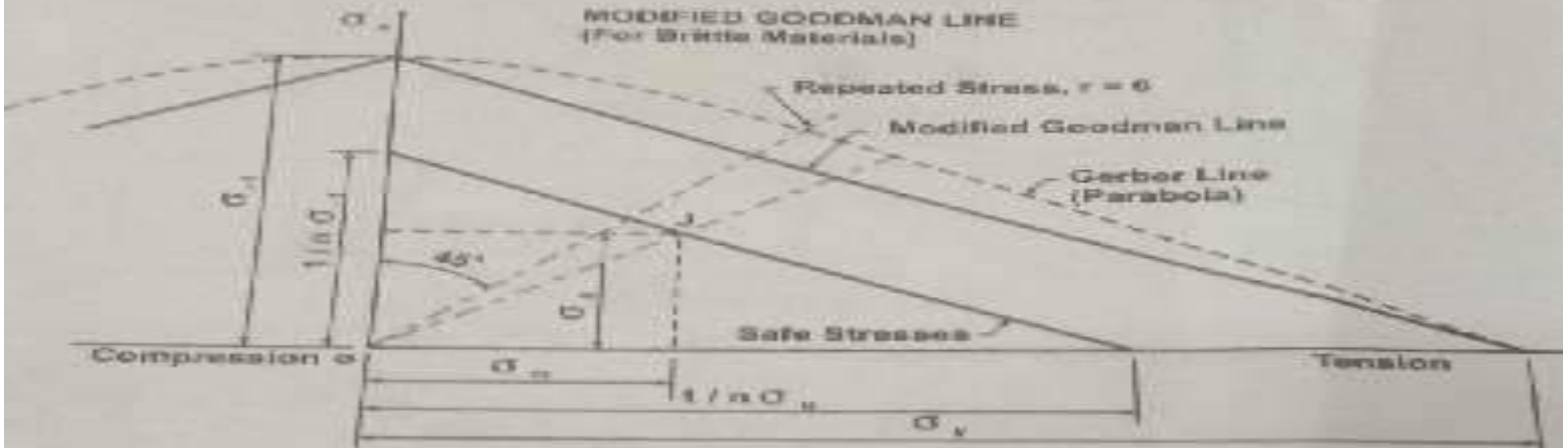
Refer pg.No.7.4 & 7.6 /DDB



SODERBERG LINE (Recommended for ductile materials only)



MODIFIED GOODMAN LINE
(For Brittle Materials)



Soderberg Equations	
$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_{-1}}$	$\frac{1}{n} = \frac{\tau_m}{\tau_y} + \frac{\tau_a}{\tau_{-1}}$

Goodman Equations	
$\frac{1}{n} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}}$	$\frac{1}{n} = \frac{\tau_m}{\tau_u} + \frac{\tau_a}{\tau_{-1}}$

SODERBERG Equation

$$\frac{1}{n} = \frac{\sigma m}{\sigma y} + \frac{\sigma a}{\sigma - 1}$$

$$\frac{1}{n} = \frac{\tau m}{\tau y} + \frac{\tau a}{\sigma - 1}$$

Goodman equation

$$\frac{1}{n} = \frac{\sigma m}{\sigma u} + \frac{\sigma a}{\sigma - 1}$$

$$\frac{1}{n} = \frac{\tau m}{\tau u} + \frac{\tau a}{\sigma - 1}$$

Problem Objectives

1. Find factor of safety “n”

2. Find size of the member

Steps to solve the problem

Step1 Find the mean and amplitude loads

(Wmax+Wmin)/2= Wmean (Wm)

(Wmax-Wmin)/2= Wampl or Wa

(Amplitude or varying Load)

Step2 Find the mean and amplitude Stresses

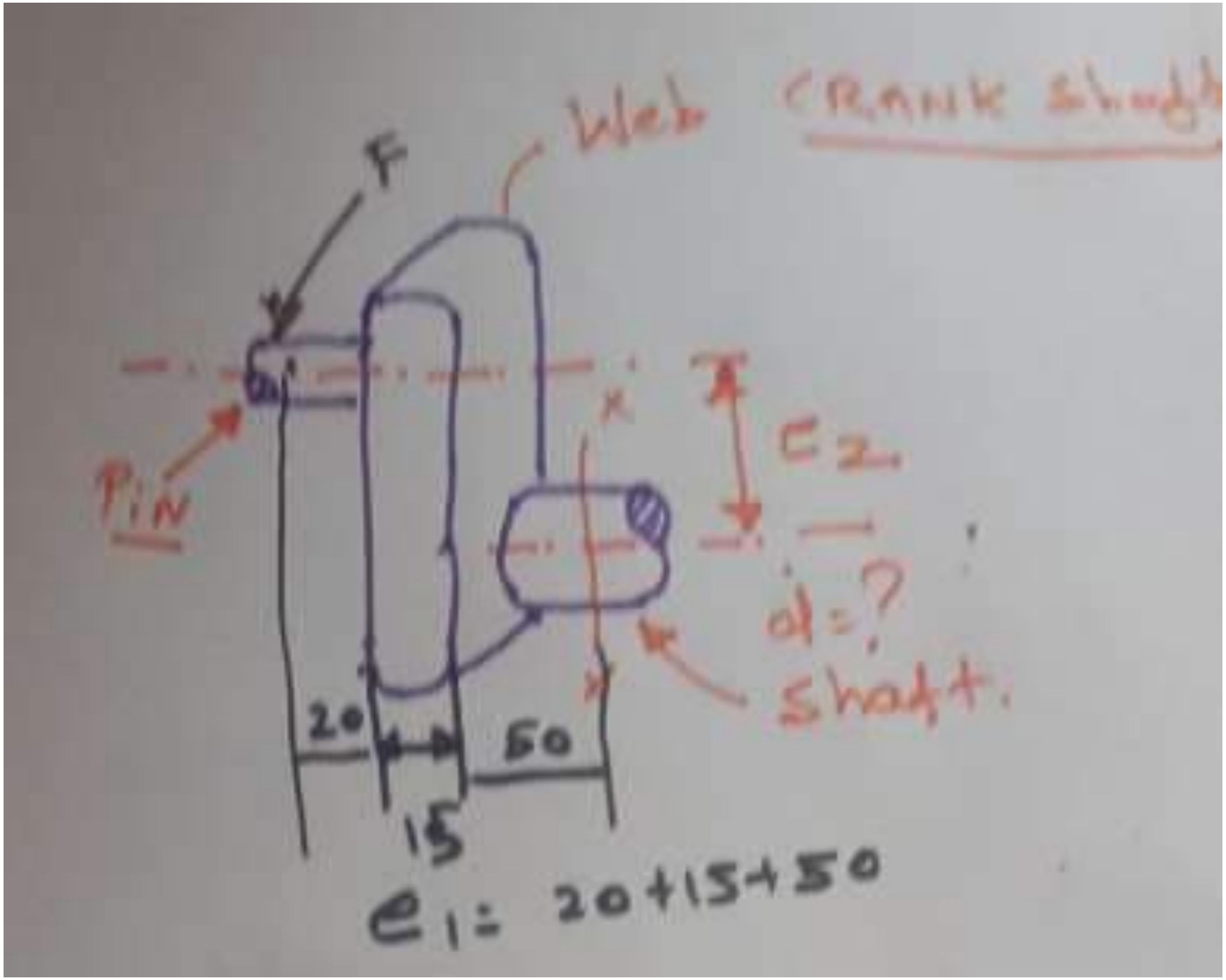
σmean= Wmean/ Area ,

σaor σv= Wa or Wv/ Area

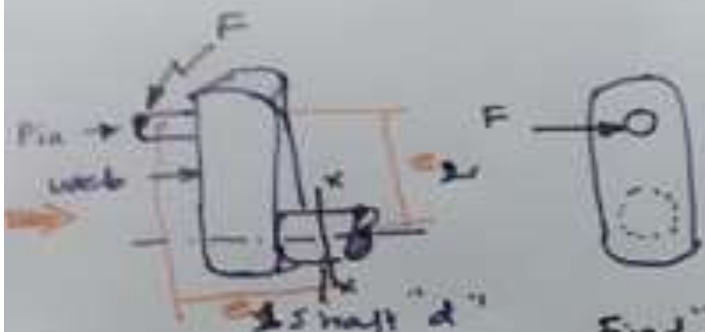
Step3 Use the theories for varying stress

Find the size or factor of safety

which ever is applicable with respect to the problem



Web
CRANK Shaft
F
Pin
C2
d=?
Shaft.
20
15
50
e1= 20+15+50

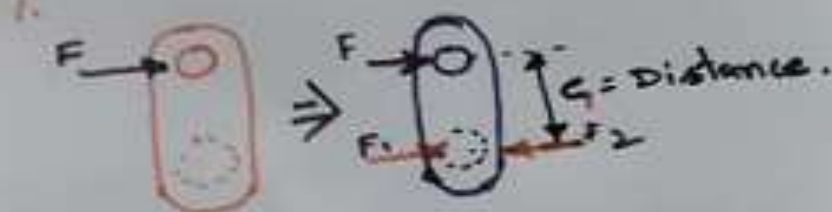


Find "d" at x–x?

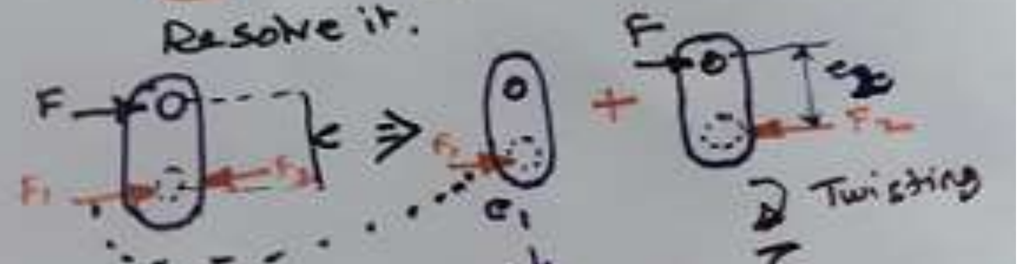
<u>Solution</u>

- Eccentric Load concept.

1.



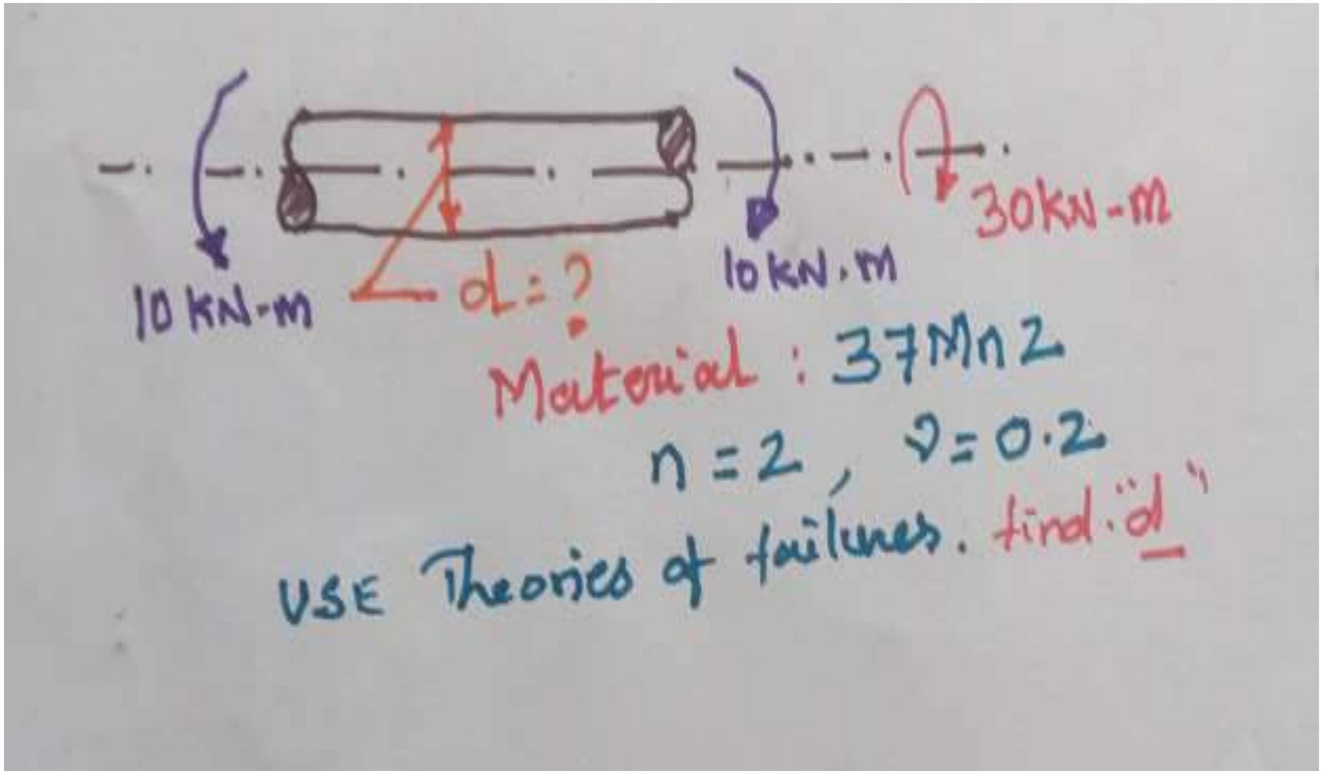
Resolve it.

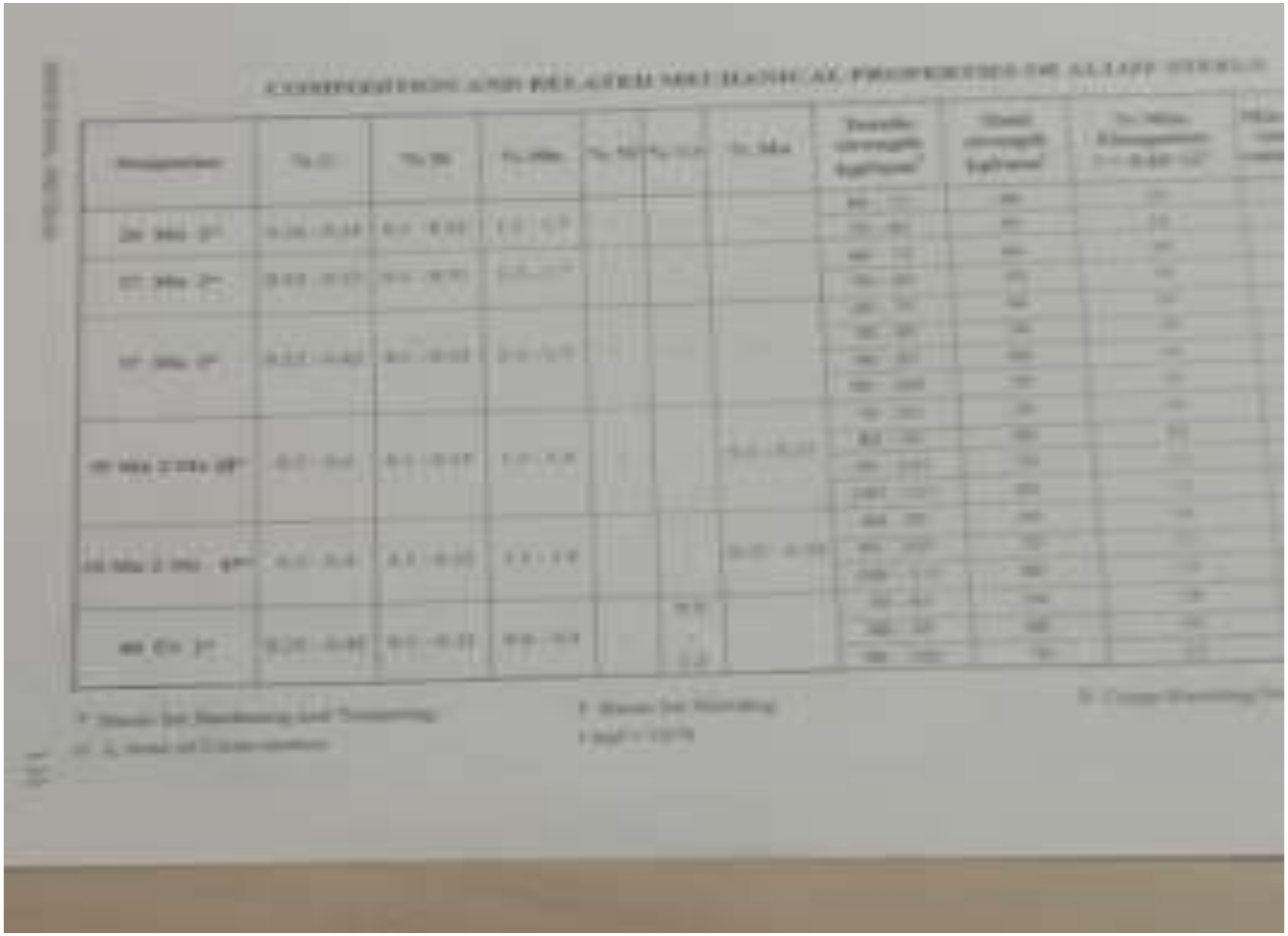


σ_{com} = σ_b + τ

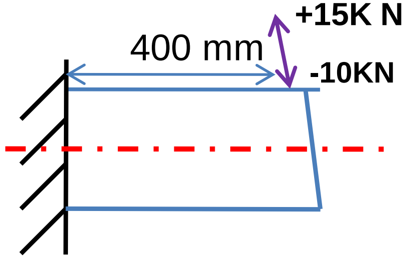
$G_{1,2}$ $\quad$ σ_y + $\tau_{xy} = \tau_{yx} = \tau$

Tutorial 3 - Topic theories of failure





Pg. 1.13 /DDB, alloyed steel



Determine the factor of safety for the member having diameter 50 mm which is subjected to fluctuating load from -10 to +15 KN. The ultimate and yield stress values are 60 N/mm^2 and 40N/mm^2 respectively. Use the Good man and Soderberg theories

Data:

n= ?

D= 50 mm

Ultimate = σu= 60 N/mm^2

Yield = σy = 40 N/mm^2

Max .load= 15000N

Min.load = -10000 N

Soderberg

$$\frac{1}{n} = \frac{\sigma m}{\sigma y} + \frac{\sigma a}{\sigma - 1}$$

Goodman

$$\frac{1}{n} = \frac{\sigma m}{\sigma u} + \frac{\sigma a}{\sigma - 1}$$

Steps to solve the problem

Step1 Find the mean and amplitude loads

(Wmax+Wmin)/2= Wmean (Wm)

(Wmax-Wmin)/2= Wampl or Wa
(Amplitude or varying Load)

Wmax = 15000 N

Wmin = -10000 N

Wm	2500N	(15000-10000)/2= Wmean (Wm)
Wa	12500N	(15000+10000)/2= Wampl or Wa

Step2 Find the mean and amplitude Stresses

σmean= Wmean/ Area

Area= 1963.1mm^2

σm = 1.3N/mm^2

σaor σv= Wa or Wv/ Area

σv = 6.4N/mm^2

Step3 Use the theories for varying stress

$1/n = (1.27348/40) + (6.3674/\sigma_1)$

Now to find σ_1

Refer pg.1.42/DDB

Take reversed cycle

Bending Load $\sigma_1 = 0.46\sigma_u$

$\sigma_1 = 27.6 N/mm^2$

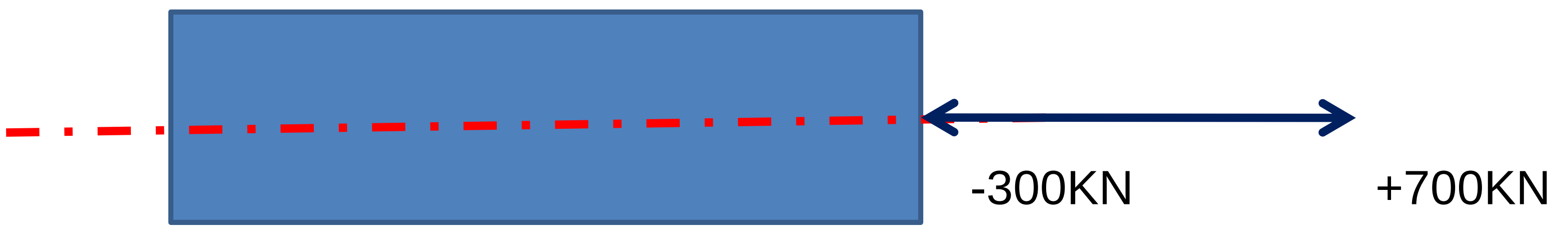
$1/n = 0.031836995 + 0.230703$

$1/n = 0.262539854$

$n \quad 3.808945518$

G&S Probm2

Determine the diameter of circular shaft subjected to axial load of range from - 300KN to 700 KN, Use factor of safety as 2.0.



NO yield, No Ultimate stress given, hence select suitable material
Then find the both stress values

Material : C50, refer pg.1.9, σy= 380 N/mm^2 & σu = 730 N/mm^2

G&S Prob2

DATA

n = 2

Wmax 700000N
Wmin -300000N

σu 730N/mm^2

σy 380N/mm^2

σ_1 ?

Soderberg

$$\frac{1}{n} = \frac{\sigma m}{\sigma y} + \frac{\sigma a}{\sigma - 1}$$

Goodman

$$\frac{1}{n} = \frac{\sigma m}{\sigma u} + \frac{\sigma a}{\sigma - 1}$$

Step 1 Caculating Wm(mean) and amplitude Wa

Wm	200000N	(Wmax+Wmin)/2= Wmean (Wm)
Wa	500000N	Wmax-Wmin)/2= Wampl or Wa

Step2 Find the mean and amplitude Stresses

σmean= Wmean/ Area

Area= 0.78525 d^2 mm^2

σm = 254695.96/ d^2 N/mm^2

σaor σv= Wa or Wv/ Area

σv = 636739.89/ d^2 N/mm^2

Step3 Use the theories for varying stress

$$1/2.0 = [254695.96/(d^2*380) + (636739.89/(d^2*\sigma_1)]$$

Now to find σ_1

Refer pg.1.42/DDB

Take reversed cycle

Load: Tension- compression

$\sigma_1 = 0.36\sigma_u$

262.8

σ_1 = **335.8** N/mm^2

1/2.0 = 670.2525176 *(1/d^2) + 1896*(1/d^2)

1/2.0 = 2566.440403/d^2

d^2 5132.880805

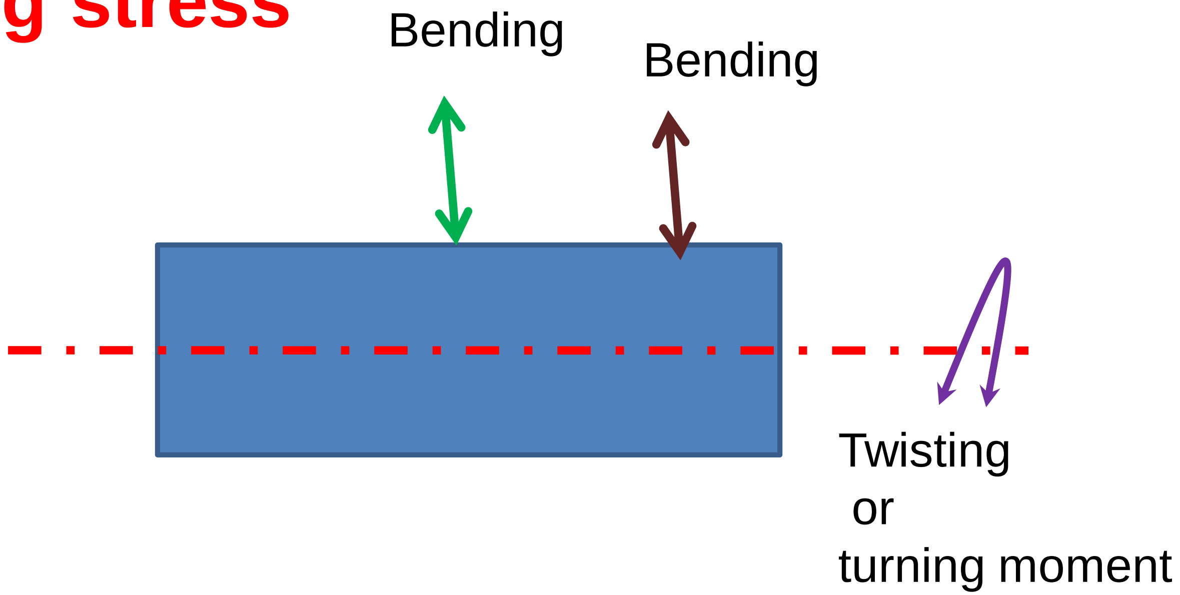
d 71.6mm

72 mm (78.65)
mm

Right answer: 78.65 mm = 79 mm

Combined varying stress

Refer Pg.7.6/DDB



$$1/n = \left[\left(\frac{\sigma eq}{\sigma y} \right)^2 + \left(\frac{\tau eq}{\tau y} \right)^2 \right]^{1/2}$$

Required Task , to calculate σeq & τeq,

Again refer Pg.7.6/DDB

Soderberg Equations	
$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + K_f \frac{\sigma_a}{\sigma_{-1}}$	$\frac{1}{n} = \frac{\tau_m}{\tau_y} + K_f \frac{\tau_a}{\tau_{-1}}$
For ductile materials	
Goodman (modified) equation	
$\frac{1}{n} = K_t \left[\frac{\sigma_m}{\sigma_u} + \frac{\sigma_a}{\sigma_{-1}}\right]$	$\frac{1}{n} = K_t \left[\frac{\tau_m}{\tau_u} + \frac{\tau_a}{\tau_{-1}}\right]$
For brittle materials (notch sensitivity negligible)	
Combined Stresses	
$\sigma_{eq} = \frac{\sigma_y}{n} = \sigma_m + K_f \frac{\sigma_a \sigma_y}{\sigma_{-1}}$	$\tau_{eq} = \frac{\tau_y}{n} = \tau_m + K_f \frac{\tau_a \tau_y}{\tau_{-1}}$
$\frac{1}{n} = \left[\left(\frac{\sigma_{eq}}{\sigma_y}\right)^2 + \left(\frac{\tau_{eq}}{\tau_y}\right)^2\right]^{1/2}$	**Max. shear theory** : $\tau_y = \sigma_y / 2$ **Octahedral shear theory** : $\tau_y = \sigma_y / \sqrt{3}$

Steps to solve the problem

Step1 Find the mean and amplitude Moments (bending & twisting)

$(M_{max}+M_{min})/2 = M_{mean}$ (M_m)

$(M_{max}-M_{min})/2 = M_{ampl}$ or M_a
(Amplitude or varying Load)

Step2 Find the mean and amplitude Stresses (bending & twisting)

$\sigma b_{mean} = M_{mean}/Z$, $\tau_{mean} = 16 \times T_m/(\pi \cdot d^3)$

σb_a or $\sigma v = M_a$ or M_v/Z $\tau_a = 16 \times T_a/(\pi \cdot d^3)$

Step3 Calculating equivalent σ_e & τ_{eq} varying stress

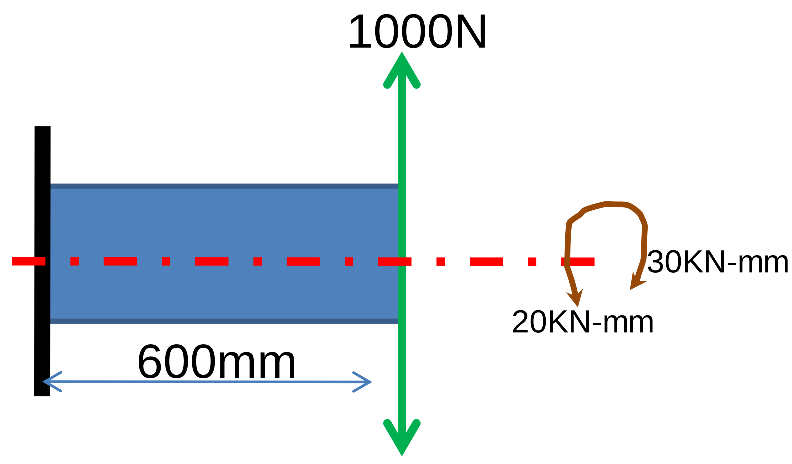
$$\sigma_{eq} = \frac{\sigma_y}{n} = \sigma_m + K_f \frac{\sigma_a \sigma_y}{\sigma_{-1}}$$

$$\tau_{eq} = \frac{\tau_y}{n} = \tau_m + K_f \frac{\tau_a \tau_y}{\tau_{-1}}$$

Step4 : Use of Soderberg equation, **Find the size or factor of safety**

$$\frac{1}{n} = \left[\left(\frac{\sigma_{eq}}{\sigma_y}\right)^2 + \left(\frac{\tau_{eq}}{\tau_y}\right)^2\right]^{1/2}$$

Combined stress Problem1



Determine the diameter of the circular member which is subjected to loads as shown in figure. Take n=2.0

DATA

d = ?

n=2.0

Bending load = +1000N to – 700N

Twisting Moment = + 30KN-mm to -20KN-mm

Use combined varying stress equation to solve

Other works

Material Assumption

List yield & ultimate Stresses

σ_1 & τ_1 – Ref. DDB_Pg.1.42

APPROXIMATE VALUES OF ENDURANCE LIMIT FOR STEELS

Loading	Reversed cycle	Repeated cycle
Tension - compression	$\sigma_{-1t} = 0.36\ \sigma_u$	$\sigma_{0t} = 0.5 \quad \sigma_u \le \sigma_y$
Bending	$\sigma_{-1b} = 0.46\ \sigma_u$	$\sigma_{0b} = 0.6 \quad \sigma_u \le \sigma_y$
Torsion	$\tau_{-1} = 0.22\ \sigma_u$	$\tau_0 = 0.3 \quad \sigma_u \le \sigma_y$

APPROXIMATE RELATIONSHIP BETWEEN ENDURANCE LIMITS FOR DIFFERENT MATERIALS

Material	Relationship
Steels (Generally)	$\sigma_{-1t} = (0.7 \text{ to } 0.8)\ \sigma_{-1b}$; $\tau_{-1} = (1.55 \text{ to } 0.58)\ \sigma_{-1b}$
Carbon Steels	$\sigma_{0b} = 1.5\ \sigma_{-1b}$; $\sigma_{0t} = 1.6\ \sigma_{-1b}$; $\tau_0 = (1.8 \text{ to } 2)\ \tau_{-1}$
Alloy Steels	$\sigma_{-1t} = 0.95\ \sigma_{-1b}$; $\sigma_{0b} = 1.6\ \sigma_{-1b}$; $\tau_0 = (1.8 \text{ to } 2)\ \tau_{-1}$ $\sigma_{0t} = (1.5 \text{ to } 1.6)\ \sigma_{-1t}$
Copper Alloys	$\tau_{-1} = 0.58\ \sigma_{-1b}$; $\tau_0 = (1.4 \text{ to } 2)\ \tau_{-1}$
Aluminum Alloys	$\sigma_{0b} = 1.8\ \sigma_{-1b}$; $\sigma_{-1t} = 0.7\ \sigma_{-1b}$; $\tau_{-1} = (0.55 \text{ to } 0.58)\sigma_{-1b}$ $\tau_0 = (1.4 \text{ to } 2)\ \tau_{-1}$
Grey Cast Iron	$\sigma_{-1t} = (0.6 \text{ to } 0.7)\ \sigma_{-1b}$; $\sigma_{0b} = (1.2 \text{ to } 1.5)\ \sigma_{-1bl}$ $\tau_{-1} = (0.75 \text{ to } 0.9)\ \sigma_{-1b}$; $\tau_0 \doteq (1.2 \text{ to } 1.3)\ \tau_{-1}$ $\quad \dfrac{\sigma_{-1b}}{\sigma_u} \approx 0.38$

Endurance strength for finite life $\sigma'_{-1} = \sigma_{-1} \left(\frac{10^6}{N}\right)^{0.09}$ where N is the required life in cycles

DATA

D	**???**
twisting moment	**30000N-mm**
twisting moment(-)	**-20000N-mm**
Bending load	**1000N**
Bending load	**-700N**
Bending load Distance	**600mm**
n- factor of safety	**2**

Step1: Cal. Mean & amplitude for bending Moment & Twisting Moment

BM = F x Distance

Max.M		600000N-mm
Min.M		-420000N-mm
	90000	
MMean		250000N-mm
Ma		950000N-mm
	510000	

Twisting Moment directly given

	5000	
T mean		20000N-mm
Tamp--- 25000		40000N-mm

Step2 Cal. σb and Ţ (mean and amplitude)

σb=	M/Z		
σbmean=	Mmean/Z		
Z=	I/y		
I=	π*d^4/64		
Y=	d/2		
I=	0.0491	d^4	mm^4
Y=	0.5	d	mm
z =	0.09816	d^3	mm^3
	90000		
M mean =	390000	N-mm	
σbmean=	3973256.92	*(1/d^3)	N/mm^2
	916496.945		5193482.688
σbampl=	8252149	*(1/d^3)	N/mm^2

Now , find Shear Stress

Ʈ Shear stress	T x16/(pi x d*d*d)	N/mm^2
Ʈ mean	25469.59567 *(1/d^3)	(Tmean)
Ʈ ampl	127347.9784 *(1/d^3)	(Tamp)

Step3: Cal. σeq and τeq

$$\sigma_{eq} = \frac{\sigma_y}{n} = \sigma_m + K_f \frac{\sigma_a \sigma_y}{\sigma_{-1}}$$

Kf=1 (assumed)
n=2
σ_1 = ?
σy=?

To find σy, σ_1

Material	C45	refer pg.1.9/DDB	
σy	360	N/mm^2	
σu (630- 710)	670	N/mm^2	(assumed)

Now refer pg.1.42/DDB , Get σ_1

Load	Bending	
reverse cycle		
	σ_1 =0.46σu	
σ_1 =	308.2	N/mm^2

σeq=σm+ kf*σa*σy/σ_1			
RHS	916496.945		
σm	**3973256.925**	*(1/d^3)	
	1869653768		
kf*σa*σy	~~2970773639~~	*(1/d^3)	
σ_1 =	**308.2**		
kf*σa*σy/σ_1	~~9639109.795~~	*(1/d^3)	(LHS) of equation
	6066365.243		
LHS	RHS		
σeq =	~~13612366.72~~	*(1/d^3)	

6982862.188

Now , to find

$\tau_{eq} = \tau_m + k_f * \tau_a * \tau_y / \tau_{-1}$

Kf=1 (Assumed)

τ_y = σy/2 (refer Pg. 7.6 /DDB)

τ_y = 180N/mm^2

To find τ_{-1}

Refer Pg.1.42/DDB

Load	Torsion
Cycle: Reversed	$\tau_{-1} = 0.22\sigma_u$

τ_{-1}= 147.4N/mm^2

RHS of equation

Ʈm	**25469.59567*(1/d^3)**	**N/mm^2**
kf*Ʈa*Ʈy	**155513.135*(1/d^3)**	**N/mm^2**
	22922636.04	
kf*Ʈa*Ʈy/Ʈ_1	**1055.041622*(1/d^3)**	
	155513.135*(1/d^3)	

Therefore

Ʈeq= 26524.63729*(1/d^3)

180982.7308

STEP 4: CALCULATION OF " d" USING COMBINED VARYING STRESS EQUATION (REFR Pg.7.6 /DDB)

LHS of equation

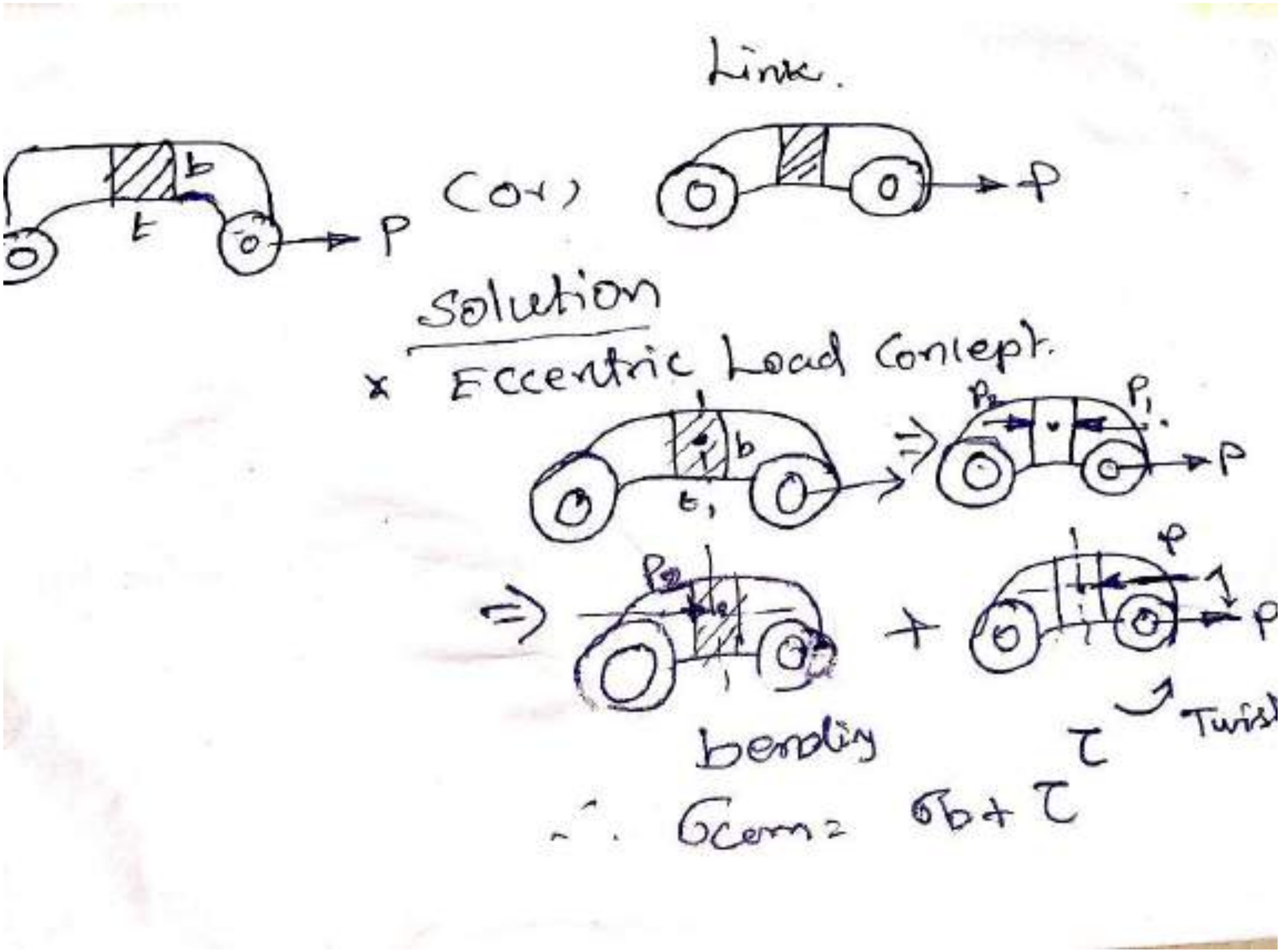
0.5

RHS of equation

(σeq/σy)^2	1429757158*((1/d^3))^2
	376237379.1
(Ʈeq/Ʈy)^2	21714.7032*((1/d^3))^2
	1010949.027

0.5 =	37812.4169*(1/d^3)	
	19422.882	
d^3	75624.8338	
	38845.763	
d	42.130mm	33.86

Some critical and important problem



Tutorial 4: Topic: combined varying stress

A hot rolled C60 steel shaft is subjected to torsion moment that varies from 330 N-m clockwise to 110N-m counter clockwise and an applied bending moment at a critical section varies from 440 N-m to -220N-m. The shaft is of uniform cross section and no keyway is present at the critical section. Determine the required diameter of shaft. Take factor of safety as 2, the size factor as 0.85 and surface finish factor as 0.62.

UNIT II

ME18503-Design Of Machine Elements

UNIT II DESIGN SHAFT, KEYS AND FITS AND TOLERANCE AND COUPLINGS 12

Preferred Numbers- Standardization Design of shafts under static and fatigue loadings, Keys – types of keys , design of keys. Design of Rigid coupling, and Flexible coupling. Fits- types of Fits and Tolenrance- hole basis system Shaft basis problems.

Objective

- **This course will make acquainted design principles on shaft, fits and tolerances. and couplings.**

Outcome

Analysing and applying the design of solid, hollow shafts keys and couplings. Also Understanding knowledge of fits and tolerance and analysing it

ROAD MAP

SHAFT - DESIGN - static and fatigue loading

KEY Design

Couplings- Rigid coupling & Flexible coupling

Fits & Tolerance

Preferred Numbers & Standardization

SHAFT

What is shaft?

An element which is usually in circular or round bar to transmit the motion from one element to other

Types of Shaft

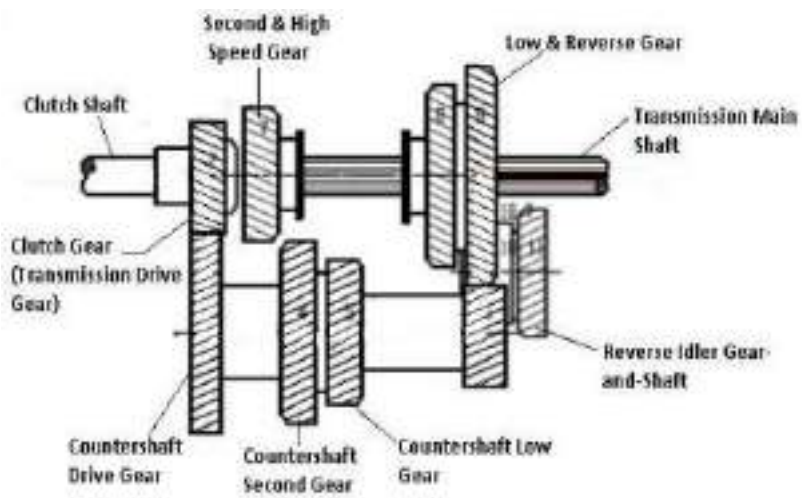
1. Shaft ---- cylindrical revolving member used in power transmission

2. Axle------ Non rotating element, acting as supporting for the rotational elements.
3. Spindle--- Short length shaft – machine tools

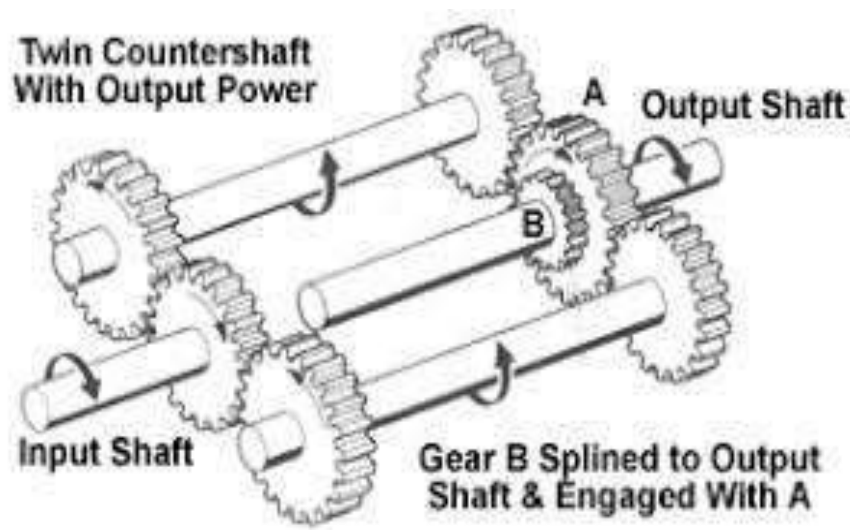
Line shaft

line shafting was used to distribute power from a large central power source to machinery

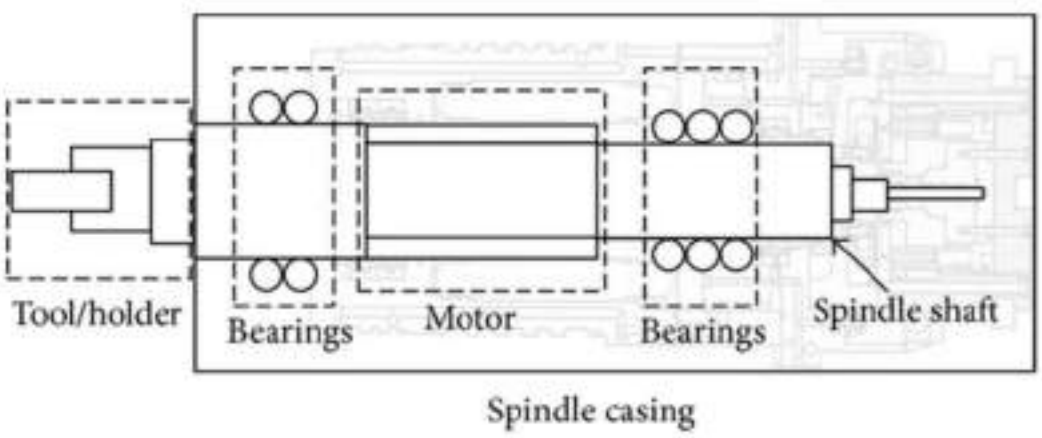
Counter shaft Or Lay shaft



Sliding Mesh Gear Box



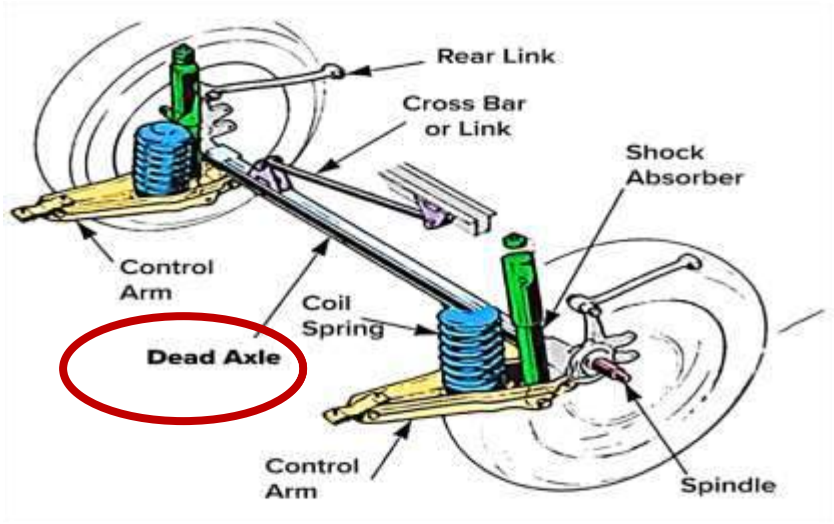
Spindle shaft



Axle

An axle is a central shaft for a rotating wheel or gear. On wheeled vehicles, the axle may be fixed to the wheels, rotating with them, or fixed to the vehicle, with the wheels rotating around the axle. ... Sometimes, especially on bicycles, bikes.

bullock cart wheels on shaft



Shaft manufacturing : hot Rolled or cold working methods

Shaft standard Size: 5 – 7 meters.

Shaft Design

1. **Strength Based :- theory of simple bending= $M/I= \sigma/Y$**

2. **Rigidity /stiffness based:- Theory of Torsion= $T/J=\tau/y$**

Shaft generally subjected to

1. **Torsion due to rotational action**
2. **Bending due to pulleys mounted on it.**
3. **Both**
4. **axial loads in some specific application**

For remembrances

types of beam:

1. Cantilever
2. Simply supported
3. Over hanging (right, Left, or Both sides)

Shafts may be solid or hollow

hollow shaft is better than solid for the same power transmission.
due to saving of material.

Shaft design always prefers the Max Shear stress Theory

Refer the eqn. Pg.7.2/DDB. $\tau_{max} = \pm\sqrt{\sigma^2 + 4.\tau^2}$

SHAFT DESIGN STEPS

Objective = Diameter finding

Step 1: Indentify the loads applied on shaft

Step2: Select material (optional) List σu & σy

Step3: calculate T torque transmission

$$P = (2x \pi x NT) / 60,$$

N- rpm, P = power, T= ?

Step 4: Find the maxim bending moment
(use of BMD- bending moment diagram)

Step5 : Apply the equation to Find " d " of the shaft –
applying the loading both T & M

use the equation in Pg. 7.21/DDB

$$d_o^3 = \frac{16}{\pi [\tau] \left\{ 1 - \left(\frac{d_i}{d_o} \right)^4 \right\}} \sqrt{\left[K_b M_b + \alpha \frac{P d_o}{8} \left(1 + \frac{d_i^2}{d_o^2} \right) \right]^2 + (K_t M_t)^2}$$

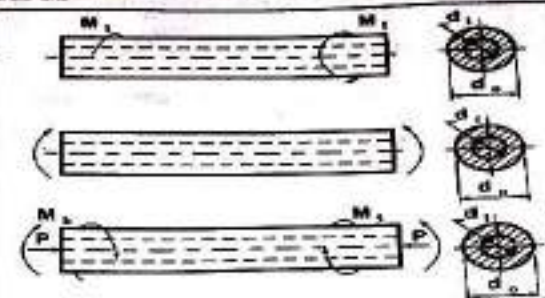
* **Hollow shaft subjected to torsion only**

$$\tau_{max} = \frac{16\ M_t\ d_o}{\pi\left(d_o^4 - d_i^4\right)}$$

* **Hollow shaft subjected to bending only**

$$\sigma_{b\,max} = \frac{32\ M_b\ d_o}{\pi\left(d_o^4 - d_i^4\right)}$$

* **Hollow shaft subjected to bending, torsion and axial load**



$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha\frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

* **For solid shaft put $d_i = 0$**   **Shaft sizes to be rounded off to R 20 series**

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

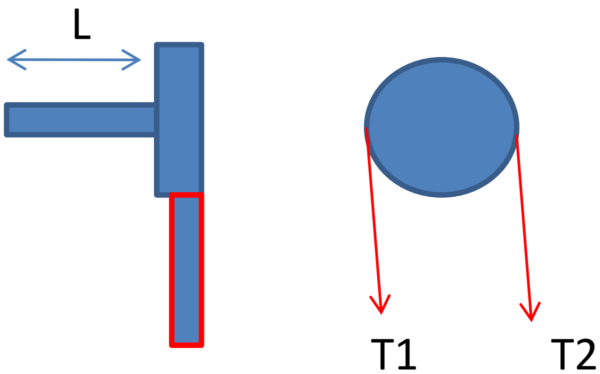
P	axial load
M_t	twisting moment
M_b	bending moment
$[\tau]$	design shear stress
K_b	combined shock and fatigue factor applied to M_b
K_t	combined shock and fatigue factor applied to M_t
τ	shear stress
σ_b	bending stress
α	column action factor
	$= 1$ for tensile load
	$= \dfrac{1}{1 - 0.0044\left(\frac{l}{r}\right)}$ for $\frac{l}{r} < 115$
	$= \dfrac{\sigma_y}{\pi^2\ n\ E}\left(\frac{l}{r}\right)^2$ for $\frac{l}{r} > 115$
E	young's modulus
σ_y	yield stress
r	radius of gyration
l	length of shaft under axial load
n	end condition coefficient

Standard Series

BASIC SERIES OF PREFERRED NUMBERS

R 5 φ = 1.6	R 10 φ = 1.25	R 20 φ = 1.12	R 40 φ = 1.06
1.00	1.00	1.00	1.00
			1.06
		1.12	1.12
			1.18
	1.25	1.25	1.25
			1.32
		1.40	1.40
			1.50
1.60	1.60	1.60	1.60
			1.70
		1.80	1.80
			1.90
	2.00	2.00	2.00
			2.12
		2.24	2.24
			2.36
2.50	2.50	2.50	2.50
			2.65
		2.80	2.80
			3.00
	3.15	3.15	3.15
			3.35
		3.55	3.55
			3.75
4.00	4.00	4.00	4.00
			4.25
		4.50	4.50
			4.75
	5.00	5.00	5.00
			5.30
		5.60	5.60
			6.00
6.30	6.30	6.30	6.30
			6.70
		7.10	7.10
			7.50
	8.00	8.00	8.00
			8.50
		9.00	9.00
			9.50
10.00	10.00	10.00	10.00

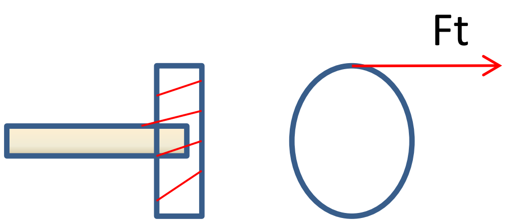
Some task for finding the bending moment



Belt pulley

$W = T1 + T2$

$BM = W \times L$



Gear drive /pulley

$W_g = F_t / \cos\alpha$

$BM = W_g \times L/4$

Ft = tangential force

α = Pressure angle(20^o)

Pg.6.4/DDB

CANTILEVER BEAMS

Bending Moment	Reaction	Deflection	Type of Loading
$M_x = -Px$ $M_{max} = -PL$	$R_B = P$	$y = -\frac{P}{6EI}\left[2L^3 - 3L^2x + x^3\right]$ $y_{max} = -\frac{PL^3}{3EI}$ at $x = 0$	
$M_x = -\frac{wx^2}{2}$ $M_{max} = -\frac{wL^2}{2}$	$R_B = wL$	$y = -\frac{w}{24EI}\left[3L^4 - 4L^3x + x^4\right]$ $y_{max} = -\frac{wL^4}{8EI}$ at $x = 0$	
$M_x = -P(x-a)$, $x > a$ $M_{max} = -P(L-a)$	$R_B = P$	$y = -\frac{P}{EI}(L-a)^2\left[\frac{(L-a)}{3} - \frac{(x-a)}{2}\right]$, $0 < x < a$ $y = -\frac{P}{6EI}\left[(x-a)^3 + (L-a)^2(2L - 3x + a)\right]$, $a < x < L$ $y_{max} = -\frac{P(L-a)^3}{3EI} - a\left[\frac{P(L-a)^2}{2EI}\right]$ at $x = 0$	
$M_x = -\frac{wx^3}{6L}$ $M_{max} = -\frac{wL^3}{6L}$	$R_B = \frac{wL}{2}$	$y = -\frac{w}{120EIL}\left[x^5 - 5L^4x + 4L^5\right]$ $y_{max} = -\frac{wL^4}{30EI}$ at $x = 0$	
$M_x = -\frac{wx^2}{6L}(3L - x)$ $M_{max} = -\frac{wL^2}{3}$	$R_B = \frac{wL}{2}$	$y = -\frac{w}{120EIL}\left[-x^5 - 15L^4x + 5Lx^4 + 11L^5\right]$ $y_{max} = -\left[\frac{11}{120EI}\right]wL^4$ at $x = 0$	
$M_x = -m$ constant	$R_B = 0$	$y = -\frac{m}{2EI}\left[L^2 - 2Lx + x^2\right]$ $y_{max} = -\frac{mL^2}{2EI}$ at $x = 0$	

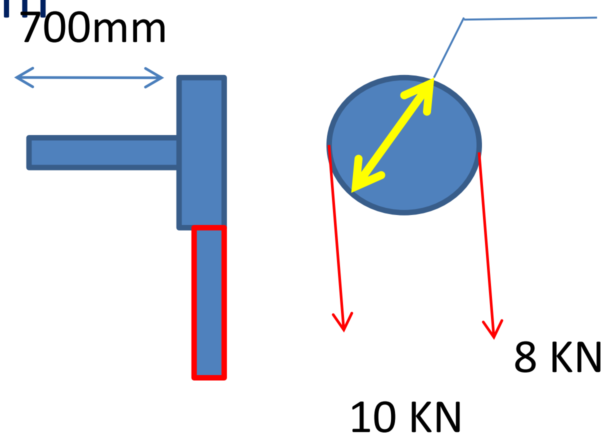
Pg.6.5/DDB

SIMPLY SUPPORTED BEAM

Bending Moment	Reaction	Deflection	Type of Loading
$M_x = \frac{P}{2}x,\ 0 < x < \frac{L}{2}$ $M_{max} = \frac{PL}{4}$	$R_A = \frac{P}{2}$ $R_B = \frac{P}{2}$	$y = \frac{P}{48EI}[3L^2x - 4x^3]$ $y_{max} = \frac{PL^3}{48EI}$ at $x = \frac{L}{2}$	
$M_x = \frac{Pb}{L}x,\ 0 < x < a$ $M_{max} = \frac{Pab}{L}$	$R_A = \frac{Pb}{L}$ $R_B = \frac{Pa}{L}$	$y = \frac{Pbx}{6EIL}[L^2 - b^2 - x^2]$ [illegible] $y_{max} = $ [illegible]	
$M_x = \frac{wLx}{2} - \frac{wx^2}{2}$ $M_{max} = \frac{wL^2}{8}$ at $x = \frac{L}{2}$	$R_A = \frac{wL}{2}$ $R_B = \frac{wL}{2}$	$y = \frac{wx}{24EI}[L^3 - 2Lx^2 + x^3]$ $y_{max} = \frac{5}{384}\left[\frac{wL^4}{EI}\right]$ at $x = \frac{L}{2}$	
$M_x = Px$ for $0 < x < a$ $M_x = Pa$ for $a < x < a + b$ $M_{max} = Pa$	$R_A = P$ $R_B = P$	$y = \frac{Px}{6EI}$ [illegible] for [illegible] $y = \frac{Pa}{6EI}$ [illegible] for [illegible] $y_{max} = \frac{Pa}{24EI}[3L^2 - 4a^2]$ at $x = \frac{L}{2}$	
$M_x = Wx\left[\frac{1}{2} - \frac{2}{3}\frac{x^2}{L^2}\right]$ for $0 < x < \frac{L}{2}$ $M_{max} = \frac{WL}{6}$ at $x = \frac{L}{2}$	$R_A = \frac{W}{2}$ $R_B = \frac{W}{2}$	$y = \frac{Wx}{12EIL^2}$ [illegible] for $0 < x < \frac{L}{2}$ $y_{max} = \frac{WL^3}{60EI}$ at $x = \frac{L}{2}$	
$M_x = \frac{Wx}{3}\left[1 - \frac{x^2}{L^2}\right]$ $M_{max} = 0.128\,WL$ at $x = 0.5774\,L$	$R_A = \frac{W}{3}$ $R_B = \frac{2W}{3}$	$y = \frac{Wx}{180EIL^2}[3x^4 - 10L^2x^2 + 7L^4]$ $y_{max} = 0.01304\left[\frac{WL^3}{EI}\right]$ at $x = 0.519L$	

6.5

SP1- Find the diameter of the cantilever shaft which carries a belt pulley at its end as shown in figure. Take allowable Shear Stress as 50 N/mm^2 p=10kW,at800 rpm



DATA

D= ?

Loads = Only belt pulley - tensions

T1= 10 KN,

T2= 8 KN,

[τ] = 50 N/mm^2

Equation to be used for the combination Torsion and bending

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha \frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

DATA

D	??
T1	10000N
T2	8000N
Diameter	mm
[τ]	50N/mm^2

Step1 Loads identification

T1 & T2
Weight of belt pulley
(If not given in problem,)

W = T1+ T2

If weight of the pulley is given

W = Wp + T1 + T2

Now

W 18000N (W = T1 + T2)

Step2: Material Selection, List σu & σy _ (optional)

Allowable shear stress is gievn, no need to select material

[τ] = 50 N/mm^2

Step3 Calculation of torque T

Step3: calculate T torque transmission

P= (2x π x NT) /60,

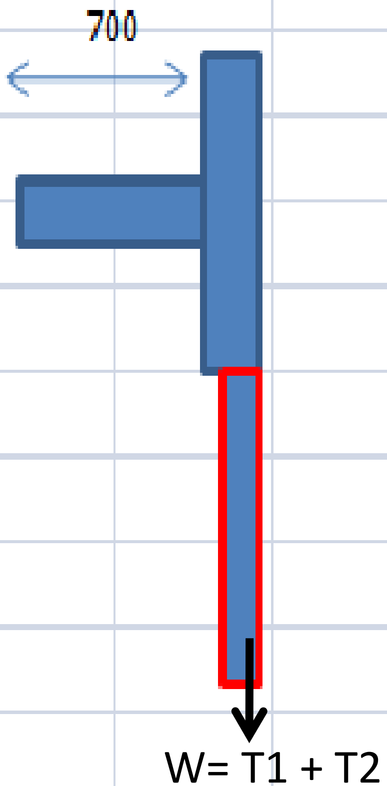
N- 500 rpm, P = 8 x 10^3, T= ?

T=Mt=152.817 N-m,

=152.817 x 10^3 N-mm

Step4 Calculation of Bending moment

Beam	cantilever	
w	T1+T2	
	18000	N
L	700	mm
M =	W X L	
M =	12600000	N-mm



Step 5 CALCulAtion of Diameter

Refer Pg. 7.21/DDB

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha\frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

0

Di = 0, for solid shaft

[Ʈ]	**50**
Mb	**12600000**
Mt	**152817**

Put P = 0 No axial load

Kb & Kt = ?
Refer Pg. 7.21/DDB

Take : revolving condition

Assumed Gradual Loading

Kb	1.5
Kt	1

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

D^3= ??

Kb x Mb	1.9E+07	3.572E+14	(Kb x Mb)^2
Kt x Mt	152817	2.335E+10	(Kt x Mt)^2

sqrt(KbMb^2+KtMt^2)= 18900617.8

16/(π x [ꞇ]) = 0.101878383

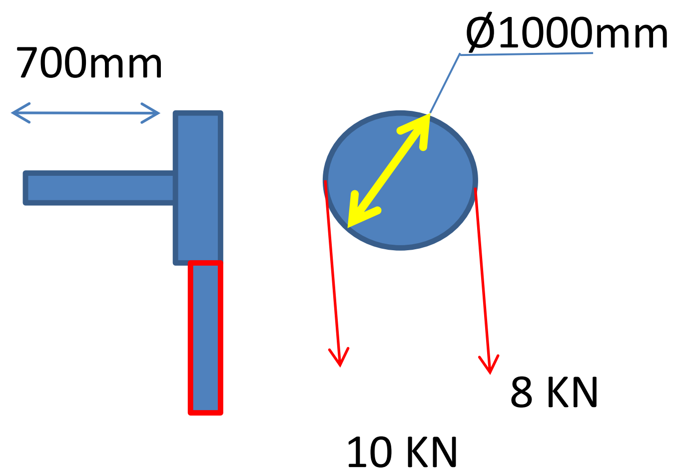
d^3 = 1925564.373

d = 123.8105999

Pg.7.20/DDB 124.409

STD R20 series 125mm

SP1a- Find the diameter of the cantilever shaft which carries a belt pulley at its end as shown in figure. Take allowable Shear Stress as 50



DATA

D= ?

Loads = Only belt pulley - tensions

T1= 10 KN,

T2= 8 KN,

[τ] = 50 N/mm^2

DATA

D	**??**
T1	**10000N**
T2	**8000N**
Diameter	**1000mm**
[τ]	**50N/mm^2**

Step1 **Loads identification**

T1 & T2
Weight of belt pulley
(If not given in prob,)

W = **T1+ T2**

If weight of the pulley is given

W = **Wp + T1 + T2**

Now

W **18000N** **(W = T1 + T2)**

Step2: Material Selection, List σu & σy _ (optional)

Allowable shear stress is gievn, no need to select material

[τ] = 50N/mm^2

Calculation of torque

Step3 T

T = F x R — R= radius of Pulley; F= net force (T1 - T2)

T = (T1-T2)*D/2

T = 1000000N-mm

Step4 Calculation of Bending moment

Beam cantilever

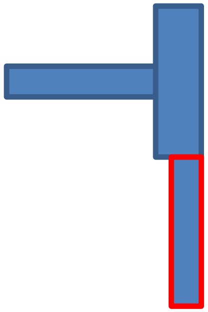
w T1+T2

18000 N

L 700mm

M = W X L

M = 12600000N-mm



Step 5 CALCulAtion of Diameter

Refer Pg. 7.21/DDB

Di = 0, for solid shaft

$[\tau]$	50
Mb	12600000
Mt	1000000

Put α = 0 No axial load

Kb & Kt = ?

Refer Pg. 7.21/DDB

Take : revolving condition

Assumed Gradual Loading

Kb	1.5
Kt	1

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

D^3=	??		
Kb x Mb	1.9E+07	3.572E+14	(Kb x Mb)^2
Kt x Mt	1000000	1.000E+12	(Kt x Mt)^2
sqrt(Kbmb^2+KtMt^2)=		18926437	
16/(π x [Ʈ]) =		0.1018784	
	d^3 =	1928194.7	
	d =	123.86689	
Refer Pg.7.20/DDB			
	STD R20 series	125	mm

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha \frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

D^3= ??

Kb x Mb	**18900000**	**(KbMb)^2 =**	**3.572E+14**
Kt x Mt	**2000000**	**(KtMt)^2 =**	**4.000E+12**

sqrt(Kbmb^2+KtMt^2)= **19005525.5**

16/(π x [Ƭ]) = **0.10187838**

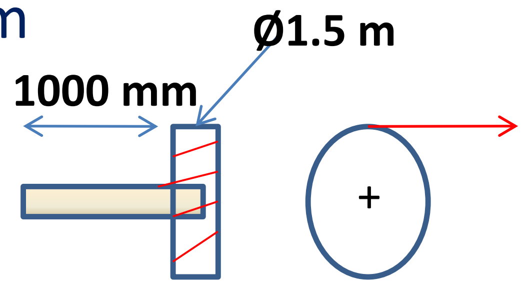
d^3 = **1936252.2**

d = **124.039018**

Pg.7.20/DDB

STD R20 series **125mm**

SP2- Find the diameter of the cantilever shaft which carries a gear pulley with pressure angle 20⁰. at its end as shown in figure. Take allowable Shear Stress as 400 N/mm^2 . Power transmitted by the drive is 8kW at 500 rpm



DATA

D= ?

Loads = Gear drive

Wgear= Ft/Cosα

α = 20⁰

[Ʈ] = 400 N/mm^2

DATA			
D	??		
P=	8 x 10^3	watts	
L	1000	mm	
Diameter	1500	mm	N= 500 rpm
[T]	400	N/mm^2	
Step1	Loads identification		
W =	Weifght of Gear Pulley		
W =	Ft/Cosα		
α =	20		
Cos20	0.4081		
T=Ft.R			
Ft =	T/R	203.7567654	N- (refer step3)
Now			
Wgear=	499.3	N	

T = P x 60/(2x π x N)

T =	152.817574N-m
	152817.574N-mm

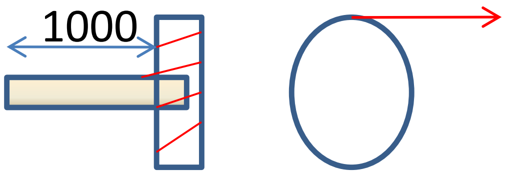
Step2: Material Selection, List σu & σy _ (optional)

Allowable shear stress is gievn, no need to select material

[τ] = 400N/mm^2

Step3	Calculation of torque T			
	P =	2x π x N x T/60		
	T =	P x 60/(2x π x N)		
	N =	500		
	P	8000	W	
	π	3.141		
	T =	152.817574	N-m	
		152817.574	N-mm	

Step4	Calculation of Bending moment			
	Beam	cantilever		
	Wgear=	499.30341		
		499.30341	N	
	L =	1000	mm	
	M =	W X L		
	M =	499303.41	N-mm	



Step 5	CALCulAtion of Diameter	
	Refer Pg. 7.21/DDB	

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha\frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

		Di = 0, for solid shaft
	[T]	400
	Mb	499303.4108
	Mt	152817.574
	Put α = 0	No axial load

Kb & Kt = ?

Refer Pg. 7.20/DDB

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

Take : revolving condition

Assumed Gradual Loading

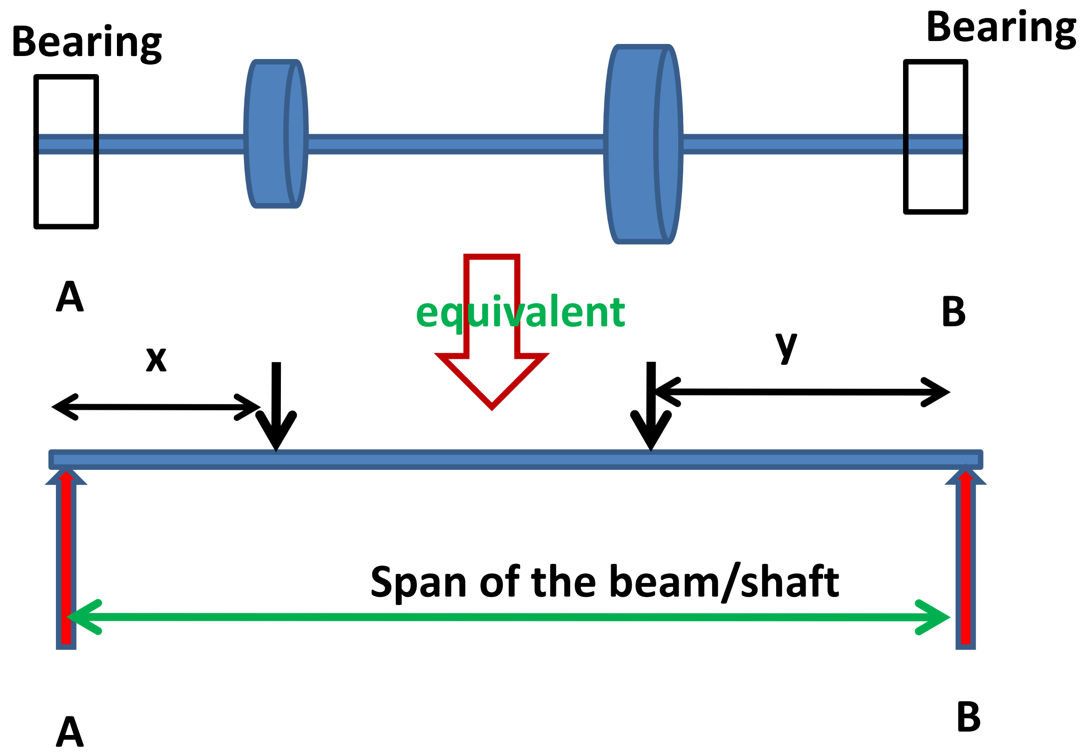
Kb	1.5
Kt	1

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha\frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

	D^3=	??			
	Kb x Mb	748955.1162		5.609E+11	=(KbMb)^2
	Kt x Mt	152817.574		2.335E+10	=(KtMt)^2
	sqrt(Kbmb^2+KtMt^2)=			764386.667	
	16/(π x [τ]) =			0.0127348	
		d^3 =		9734.30968	
		d =		21.2865666	
	Pg.7.20/DDB				
		STD R20 series		22.4	mm

Various pulleys mounted on the shaft

Develop the shaft with respect the description in the problem



SP3. A shaft is having length of 2 meter and supported at its ends. A belt driven pulley of diameter 500 mm is mounted at distance of 0.5 meter from right end bearing. The maximum and minimum belt tensions are 12 KN and 8 KN respectively. An another gear pulley of 400 mm in diameter is mounted on the shaft at 400 mm from the left end support. The power transmitted by the system is 20 KW at 1200 rpm. Shaft is made of C45 steel. Determine the diameter of the shaft?

1. Develop the shaft arrangement

$$d_o^3 = \frac{16}{\pi [\tau] \left\{1 - \left(\frac{d_i}{d_o}\right)^4\right\}} \sqrt{\left[K_b M_b + \alpha \frac{P d_o}{8}\left(1 + \frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

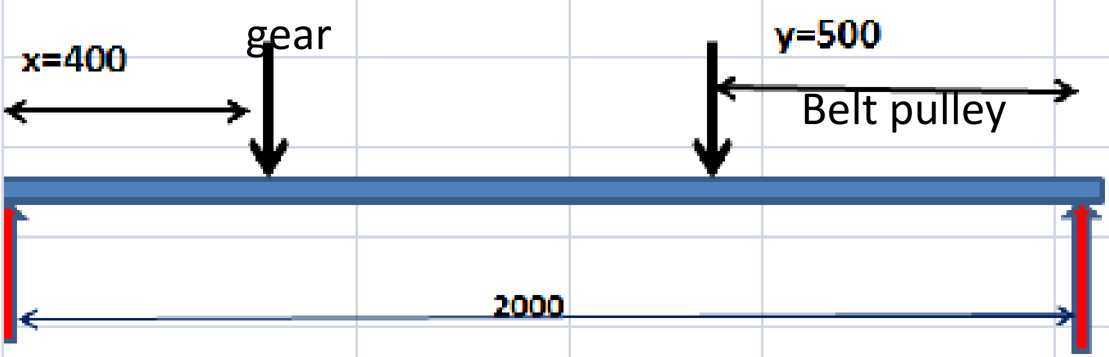
DATA

D	??	
	20 x 10^3	
P=	20000	watts
N	1200	rpm
Shaft L	2000	mm
$D_{belt\ pulley}$	500	mm
$D_{gear\ pulley}$	400	mm
[T]	??	N/mm^2
T1	12	KN
T2	8	KN

12000 N

8000 N

Shaft Diagram



Step1	**Torque calculation**		
	P =	2 πNT/60	
	N=1200rpm		
	P= 20 x 10^3		
	T =	159.1849729	N-m
		159184.9729	N-mm

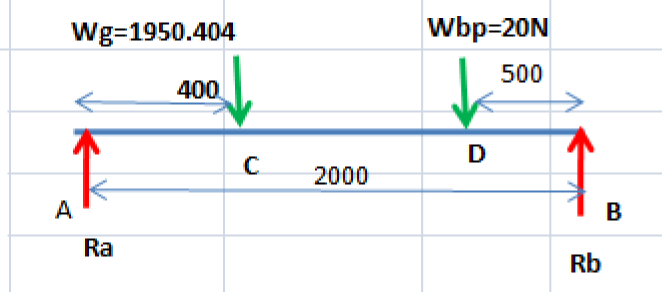
Step2	**Loads identification**			
Gear pulley				
W =	Weifght of Gear Pulley			
W =	Ft/Cosα		[α is not given]	
			[α is taken 20^o]	
α =	20			
Cos20	0.408082			
T=Ft.R				
Ft =	T/R	795.9248647	N-m	
Now				
Wgear=	1950.404	N		
Wbelt pulley				
Wbp =	T1+T2			
	20	N		

20 x 10^3

Step3: Material Selection, List σu & σy _ (optional)

	Material is given asC45			
	Allowable shear stress is not gievn			
	[τ] =	??	N/mm^2	
Refer Pg.1.9/DDB				
	C45			
	σy =	360	N/mm^2	
	σu =			
Refer Pg.7.6/DDB				
Asper τ max theory,		τ =σy/2		
		τ =	180	N/mm^2
		[τ] =	180	

Step4	Calculation of Bending moment
	Need to have line sketch of shaft
	MaxBM= ???



Ra +Rb =	Total load
Ra +Rb =	1970.4 ___ eqn1

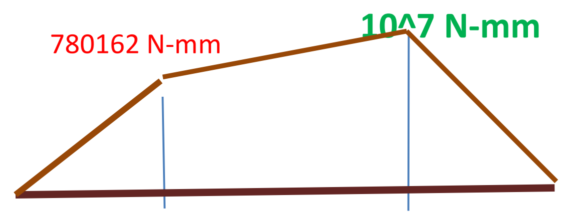
21950.404

Now bm at all points, A, C, D & B

BM at supports = 0 always

BM= F x L

Take fixed point A, moments calculated about it.

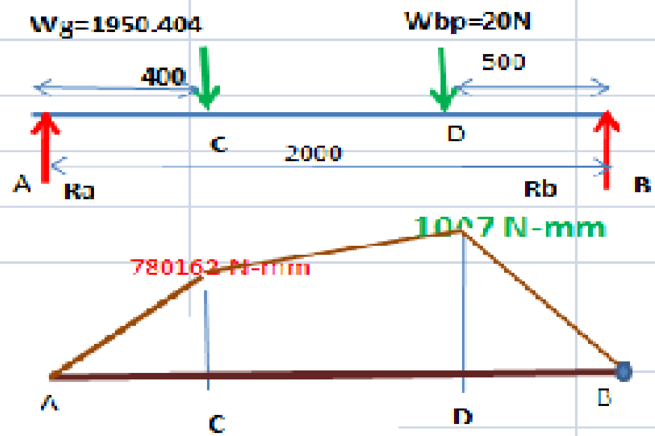


Bm at		
A	0	
C	780162	(400 x 1950.404)
D	10000	(500 x 20)
B	0	

500 x 20000 10^7

Need to have line sketch of shaft

MaxBM= ???



Ra +Rb = Total load

Ra +Rb = 21950.4 ____ eqn1

Now bm at all points, A, C, D & B

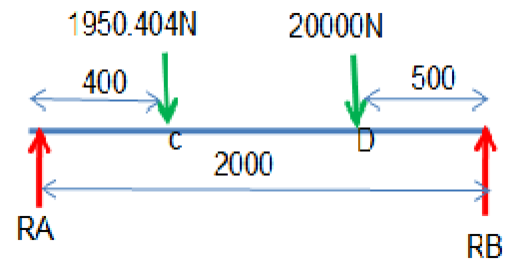
BM at supports = 0 always

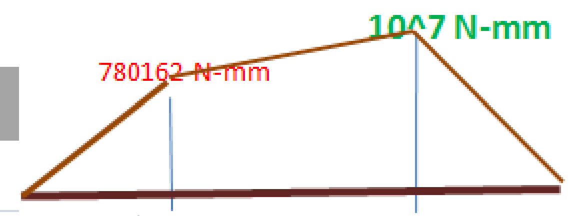
BM= F x L

Take fixed point A, moments calculated about it.

Bm at		
A	0	
C	780162	(400 x 1950.404)
D	1E+07	(500 x 20000)
B	0	

Find Ra & Rb,					
take Moment about A,					
	Clockwise= anti clock wise				
at C		at D		at B	
(400 x 1950.404)+(1500 x 20000)=				Rb x2000	
780162		30000000			
3.1E+07	=	Rb x2000			
Rb	=	15390.081	N		
Use Rb in eqn1, Ra=?					
	Ra + Rb=	21950.404			
	Ra+395.08=		21950.4		
		Ra =	6560.323	N	





Step 5	**CALCulAtion of Diameter**				
	Refer Pg. 7.21/DDB				

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}}\sqrt{\left[K_b M_b + \alpha \frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

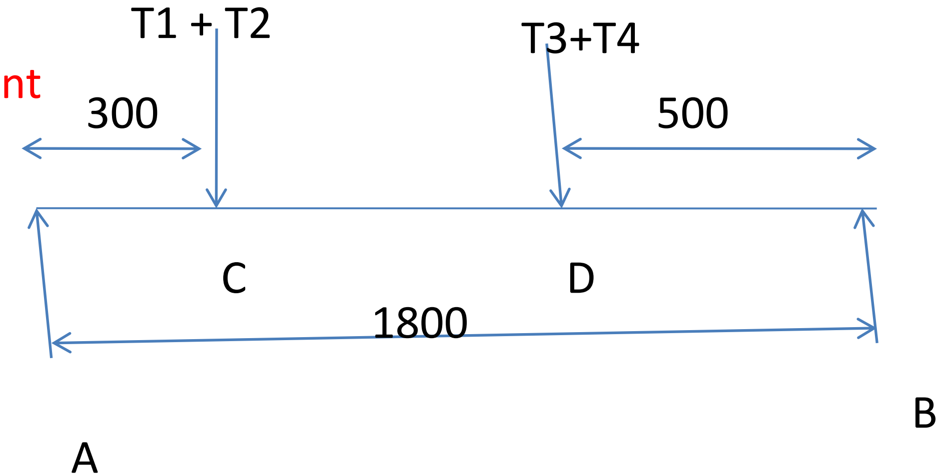
	Di = 0, for solid shaft		
[T]	**??**		
Mb	**10000000**	**N-mm**	
Mt	**159184.9729**		
Put α = 0	**No axial load**		
Kb & Kt = ?			
Refer Pg. 7.21/DDB			

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

Take : revolving condition					
Assumed	Gradual Loading				
Kb	1.5				
Kt	1				
D^3=	??				
Kb x Mb	15000000		2.250E+14	(Kb x Mb)^2)	
Kt x Mt	159184.9729		2.534E+10	(Kt x Mt)^2)	
sqrt(Kbmb^2+KtMt^2)=			15000845		
16/(π x [τ]) =			0.0282996		
	d^3 =		424517.16		
	d =		74.832304		
Pg.7.20/DDB					
	STD R20 series		80	mm	

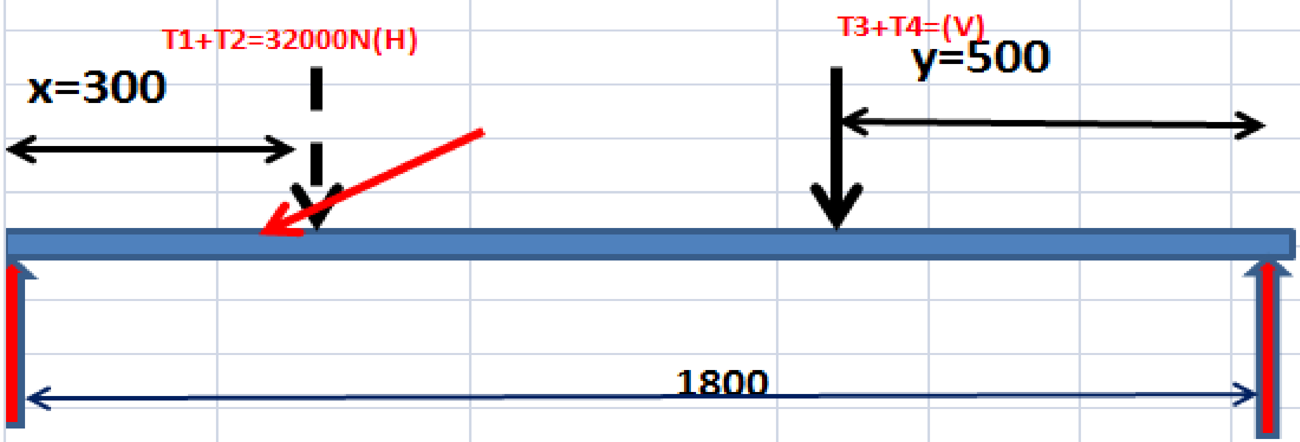
SP4. A shaft is supported at its two ends by two bearings A & B, the span between them is 1.8 meters. To the right of bearing A a belt pulley of diameter 600 mm is mounted at 300 mm takes the horizontal drive with the tensions 20KN and 12 KN. To the left of bearing B, another belt pulley of diameter of 400 mm, is located at 500mm. Also it transmits a vertical drive. Determine the diameter of the shaft when it takes minor shock loads. Take angle of contact 180deg.and μ=.25

Shaft arrangement



DATA		
D	??	
P=	?	
N	?	rpm
Shaft L	1800	mm
D1belt pulley	600	mm
D2belt pulle	400	mm
[T]	??	N/mm^2
T1	20000	N
T2	12000	N

Shaft Diagram



Step1	Torque calculation		
	P =	2 πNT/60	
	T= F X R		
	T=(T1-T2) X D1/2		
	T1-T2=	8000	
	D1/2 =	300	
	T =	2400000	N-mm

Step2 Loads identification

W1belt pulley(H)

W1bp = T1+T2

32000 N

W2Bp= T3+T4 ?

Assume the torque is same for The D2 belt Pulle

T=	T1/T2=	e^μθ	=T3/T4		
	μ =	0.3			
	θ =	170		3	
	T1/T2=	e^μθ			

T1/T2=	e^μθ
	2.1
T3/T4=	2.1

Use Torque, find T3 & T4

T=(T3-T4)xD2/2

D2/2= 200

T3= =T4*2.0993

2400000 =T4(2.0993-1) x 200

T4=	10916.04	N
T3=	22916.52	N

W2 Bp= 33832.6 N

Step3: Material Selection, List σu & σy _ (optional)

Material is given asC45

Allowable shear stress is not gievn

[Ʈ] = 0N/mm^2

Refer Pg.1.9/DDB

C45

σy = 360N/mm^2

σu =

Refer Pg.7.6/DDB

Asper Ʈ max theory, Ʈ =σy/2

Ʈ = 180N/mm^2

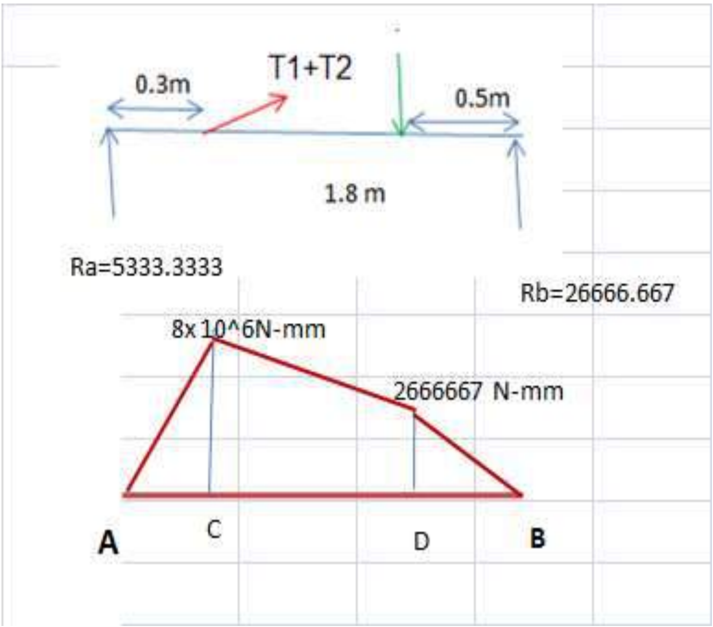
[Ʈ] = 180

Step4 Calculation of Bending moment

Need to have line sketch of shaft

MaxBM= ???

H & V



Ra +Rb =	Total load	(only Horizontal)
Ra +Rb =	32000 ____ eqn1	

Now bm at all points, A, C, D & B

BM at supports = 0 always

BM= F x L

Apply moment about A, to find Ra & Rb

1800Rb= 300 x 32000

Rb = 5333.333333 N

Ra= 26666.6667 N

Now find moment at a,b,c,d

At A & B= 0

At C

8000000(Ra x dist from c)

AT D

2666667(Rb x dist from d)

Ra +Rb = Total load **(only Vertical)**

Ra +Rb = 33832.56 ____ eqn1

Now bm at all points, A, C, D & B

BM at supports = 0 always

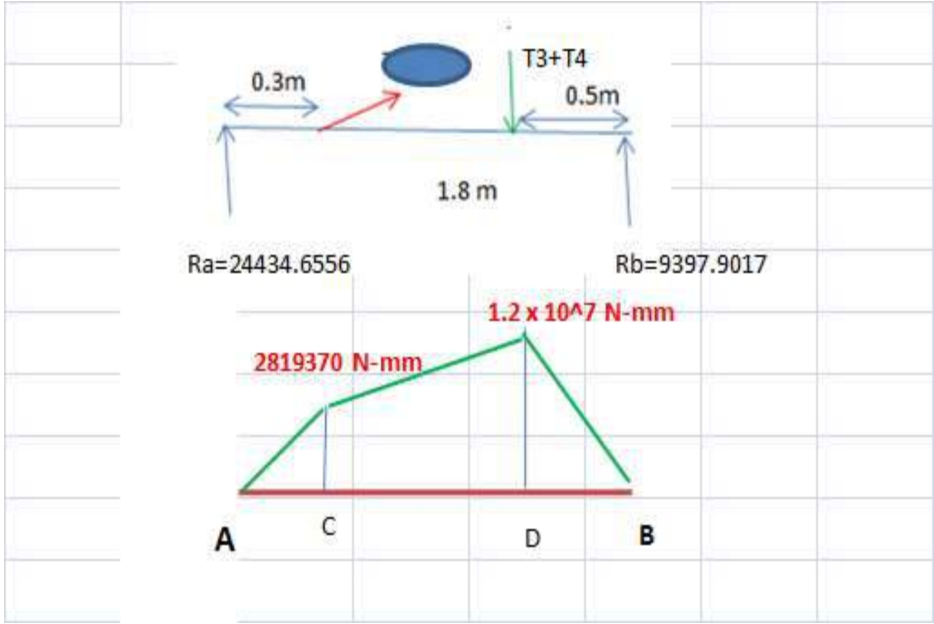
BM= F x L

Apply moment about A, to find Ra & Rb

1800Rb= 1300 x 33832.6

Rb = 24434.65556 N

Ra= 9397.901655 N



Now find moment at a,b,c,d

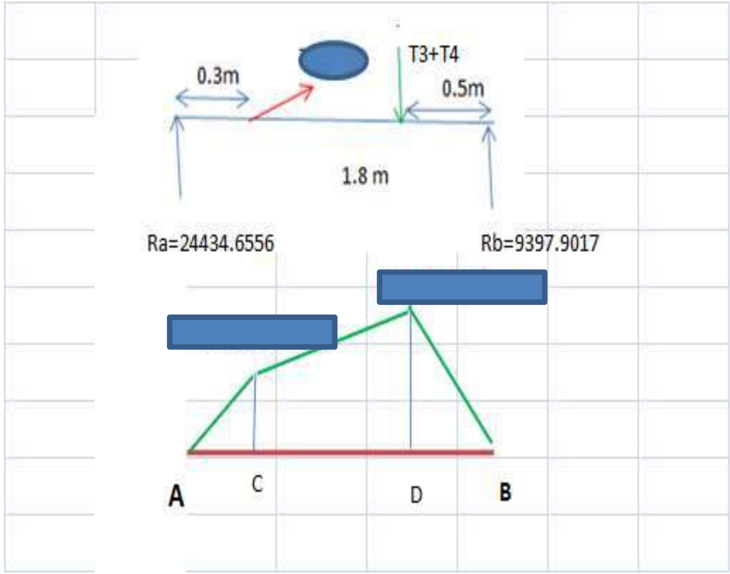
At A & B= 0

BM at Cv

2819370(Ra x dist from c) N-mm

BM at Dv

12217328(Rb x dist from d) N-mm



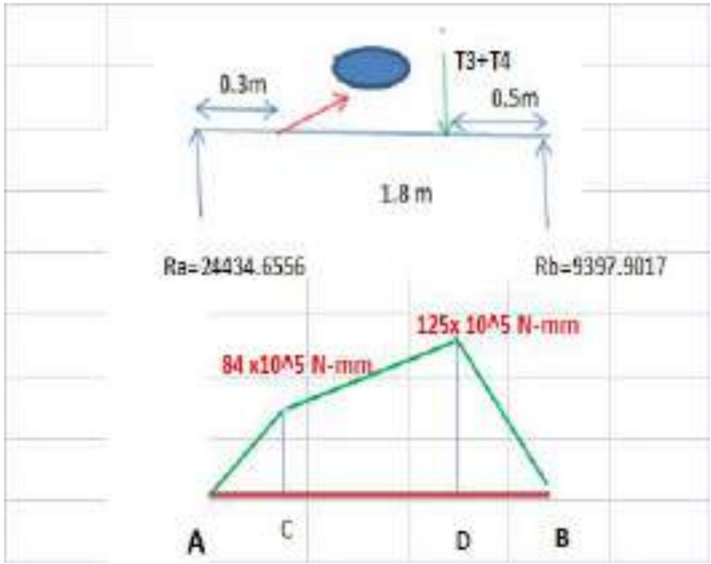
Now to find Resutant BM at C & D

RBM.C= SQRT[(BMCh)^2 + (BMCv)^2]

8482266.796 N-mm

RBM.D= SQRT[(BMDh)^2 + (BMDv)^2]

12504967.38 N-mm



Chose RBM at D as Max. BM

Step 5 CALCulAtion of Diameter

Refer Pg. 7.21/DDB

$$d_o^3 = \frac{16}{\pi [\tau] \left\{1-\left(\frac{d_i}{d_o}\right)^4\right\}} \sqrt{\left[K_b M_b + \alpha \frac{P d_o}{8}\left(1+\frac{d_i^2}{d_o^2}\right)\right]^2 + (K_t M_t)^2}$$

Di = 0, for solid shaft

[Ʈ]	180
Mb	12504967.38N-mm
Mt	2400000N-mm

Put α = 0 No axial load

Kb & Kt = ?

Refer Pg. 7.21/DDB

Type	K_b	K_t
STATIONARY SHAFT		
Gradually applied load	1	1
Suddenly applied load	1.5 - 2	1.5 - 2
REVOLVING SHAFT		
Gradual loading	1.5	1
Minor shock loads	1.5 - 2	1 - 1.5
Heavy shock loads	2 - 3	1.5 - 3

Take : revolving condition

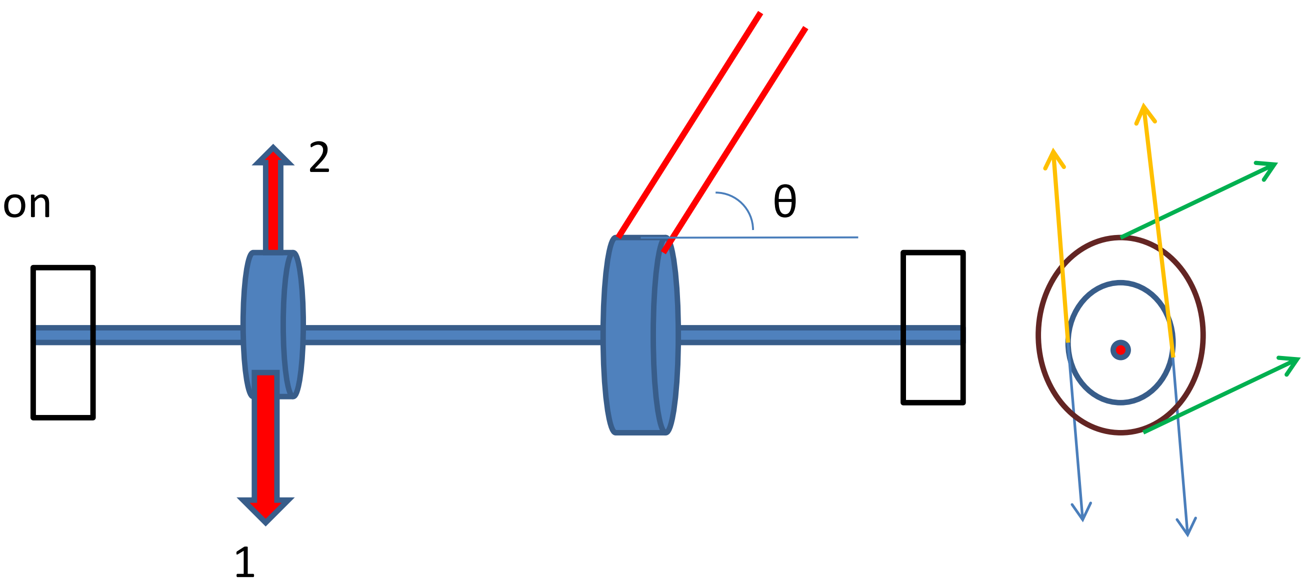
Given Minor shock loading

Kb	1.8
Kt	1.3

D^3=	??		
Kb x Mb	22508941.28	5.067E+14	(Kb x Mb)^2)
Kt x Mt	3120000	9.734E+12	(Kt x Mt)^2)
sqrt(Kbmb^2+KtMt^2)=		22724146.6	
16/(π x [Ţ]) =		0.02829955	
	d^3 =	643083.139	
	d =	85.9316106	
Pg.7.20/DDB			
	STD R20 series	90	mm

Other important points to be considered

Belt pulley inclination

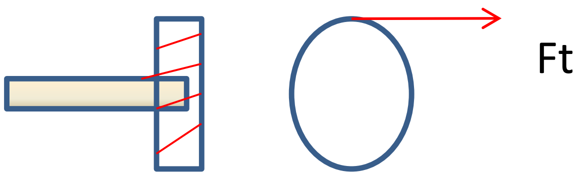


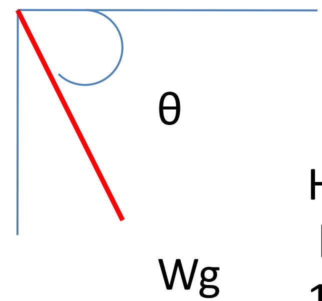
1.Vertical downward = easy to solve

2.Vertical upward= balance the force with the support.

3. Inclined = to be resolved into two components , horizontal cosθ and sinθ

Other important points to be considered





Hence two point loads will come 1.ver. & 2. horizontal

This Ft acts downward or upward based on the rotation direction.

Wt of the gear component will act at given pressure angle to the vertical or horizontal

hence $Wg = Ft/\cos\alpha$ to be taken into two component $Wg \cos\alpha$ and $Wg \sin\alpha$

Shaft problems for Sketching

1.A solid steel shaft is supported on two bearings 1.8 m apart. A 20^0 involute gearD,is keyed to the shaft at a distance of 150mm to the left of the right hand bearing. Two pulleys B and C are located on the shaft at distances of 600 mm and 1350 mm respectively to the right of left hand bearing. The drive B is vertically downward while from C the drive is downward at angle of 60^0 to the horizontal. Draw the arrangement.

2. A shaft carrying a pulley A and gear B and supported in two bearings C and D. The tangential force Ft on the gear acts vertically upwards. The pulley delivers the power Vertically down ward. B is located at 500mm to right of C and A is located 400mm to the left of D. The span between c and D is 2000mm.Draw the arrangement on the shaft.

3. A horizontal shaft of 2.5 m AD supported in bearings at A and B and carrying the pulleys at C and D. The C pulley drives the power vertically downward and D pulley delivers the power horizontally. The span of AB is 1.8 m. C located at 600mm to right of A bearing. D is located at 2500mm to the left of A bearing. Draw the arrangement

Unit2 Tutorial 1

1. A horizontal nickel steel shaft rests on two bearings , A at the left and B at the right end and carries two gears at C and D located at distances of 250 mm and 400 mm respectively from the centre line of the left and right bearings. The pitch diameter of the gear C is 600 mm and the gear D is 200 mm. The distance between the centre line of the bearings is 2400mm. The shaft transmits 20 kW at 120 rpm. The power is delivered to the shaft at gear C is taken out at the gear D in such a manner that the tooth pressure F_{tC} of gear C and F_{tD} of the gear D act vertically downwards. Find diameter of the shaft. Take weight of the gears C and D 950 N and 350 N respectively. The combined shock and fatigue factors for bending and torsion may be taken as 1.5 and 1.2. The working stress is 100 MPa in tension and 56MPa in shear.

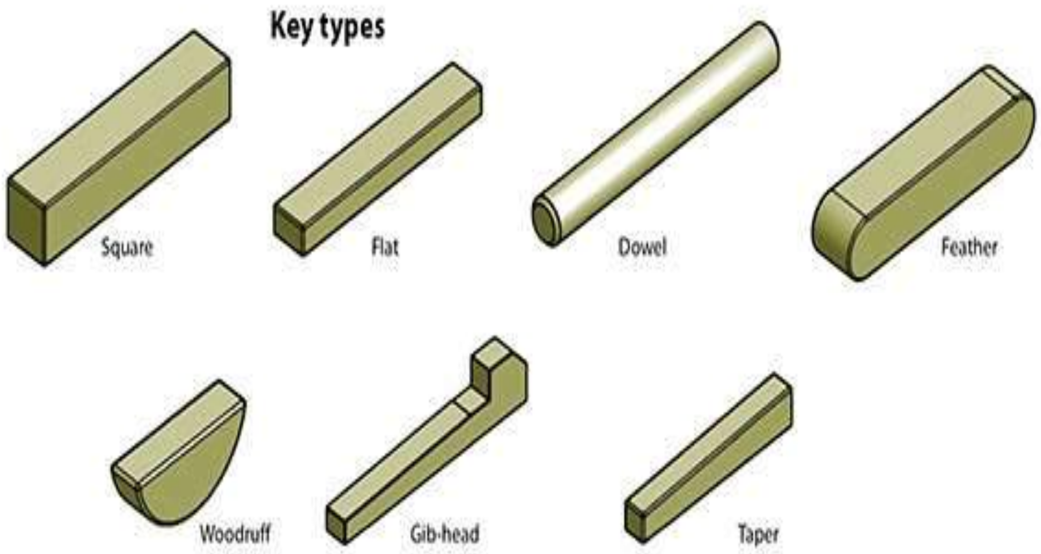
Key Design

What is key?

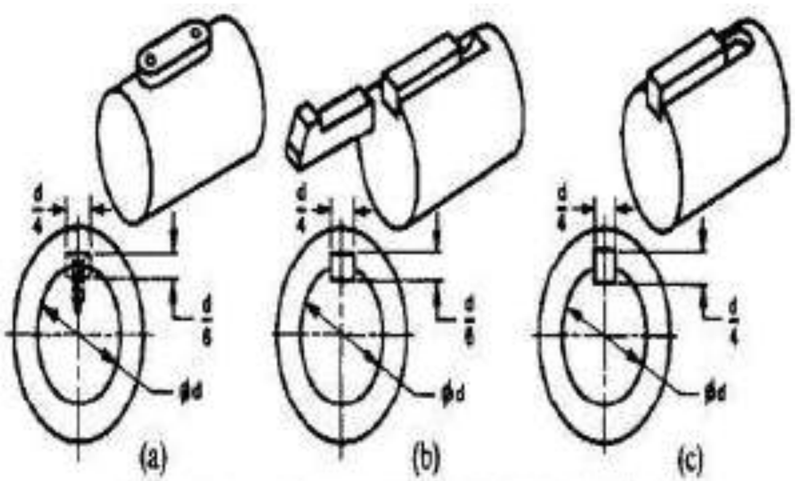
Key is a mils steel piece machine element inserted between the shaft and boss or hub of the pulley used to prevent the relative motion between the shaft and rotating element.

Types of keys

1. **Sunk keys**
2. **Saddle keys**
3. **Tangent keys**
4. **Round keys**
5. **Splines**

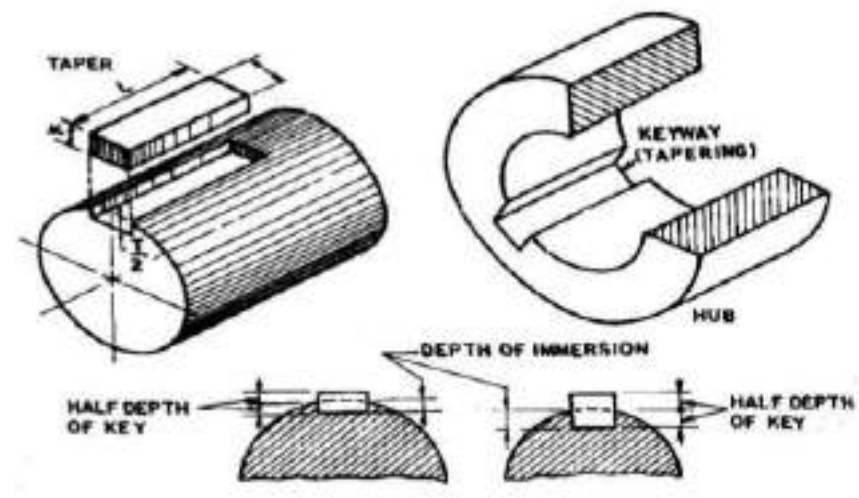


KEYS Design

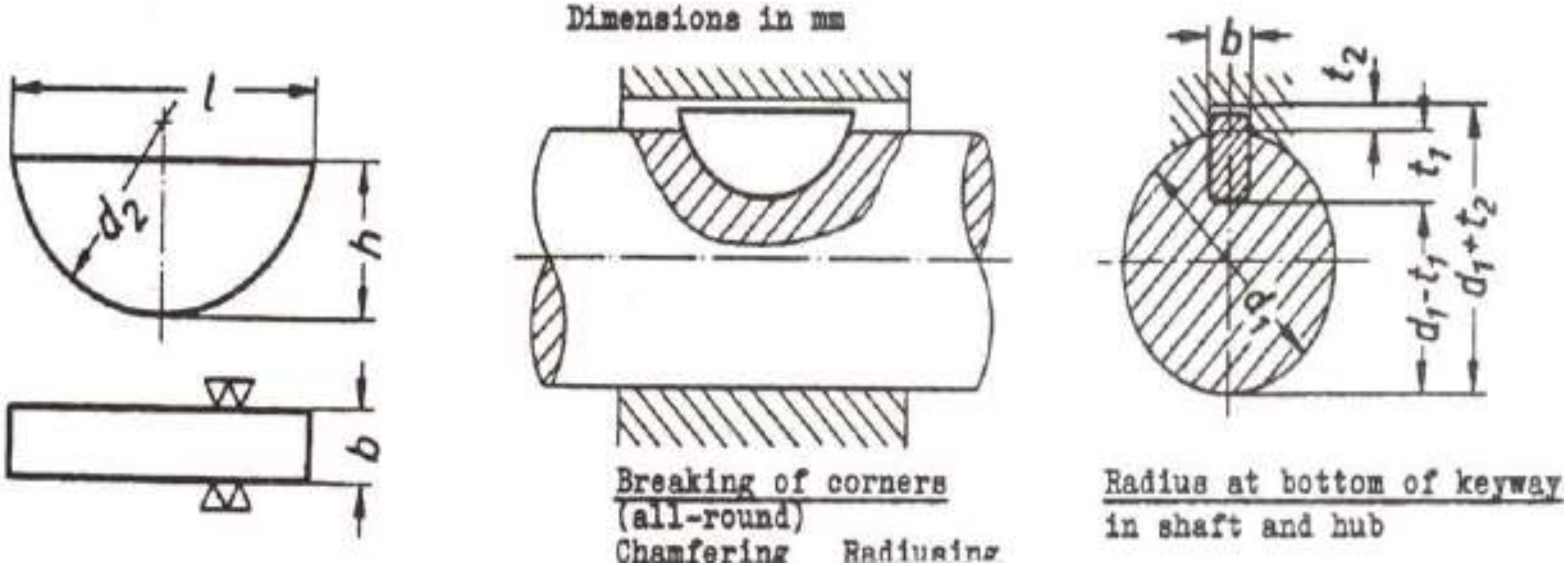


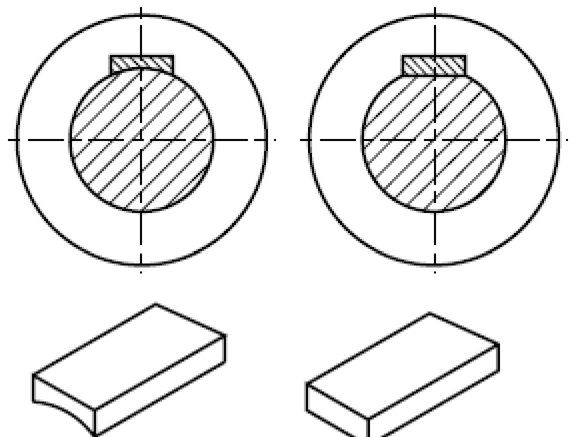
(a) Feather key, (b) rectangular key, and (c) square key

Sunk taper key



Woodruff key (used in automobile and machine tools, easy tilting in recess)





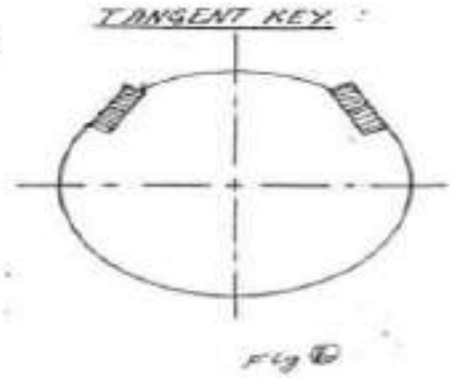
Hollow saddle key   Flat saddle key

saddle keys

Used for light loads,
Key is fitted in the hub and flat is rest on shaft surface

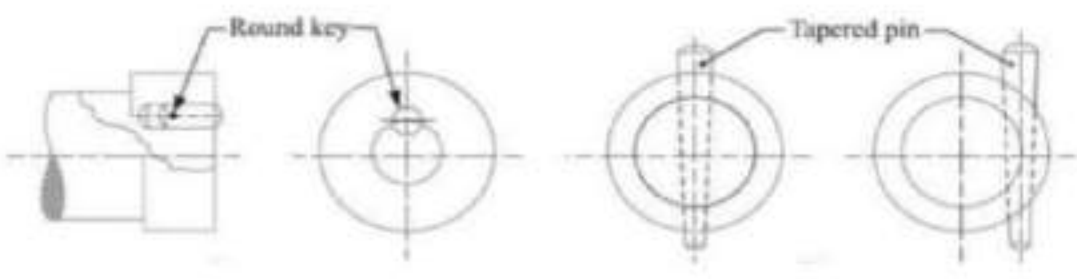
Tangent Keys

- The tangent key are fitted in a pair at right angles.
- Each key is to withatand torsion in one direction only.
- sThey are used in large heavy duty shaft.



Round key

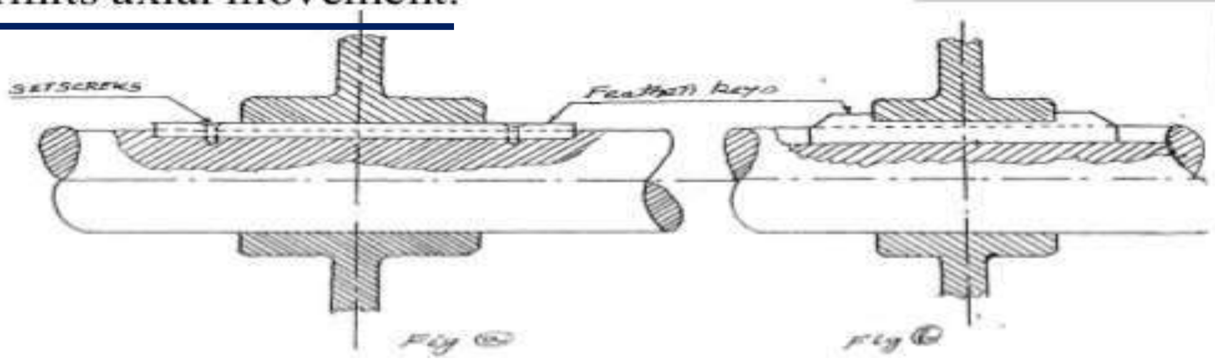
- The round keys, are circular in section and fit into holes drilled partly in the shaft and partly in the hub.
- They have advantage that their keyways may be drilled and reamed after the mating parts have been assembled.



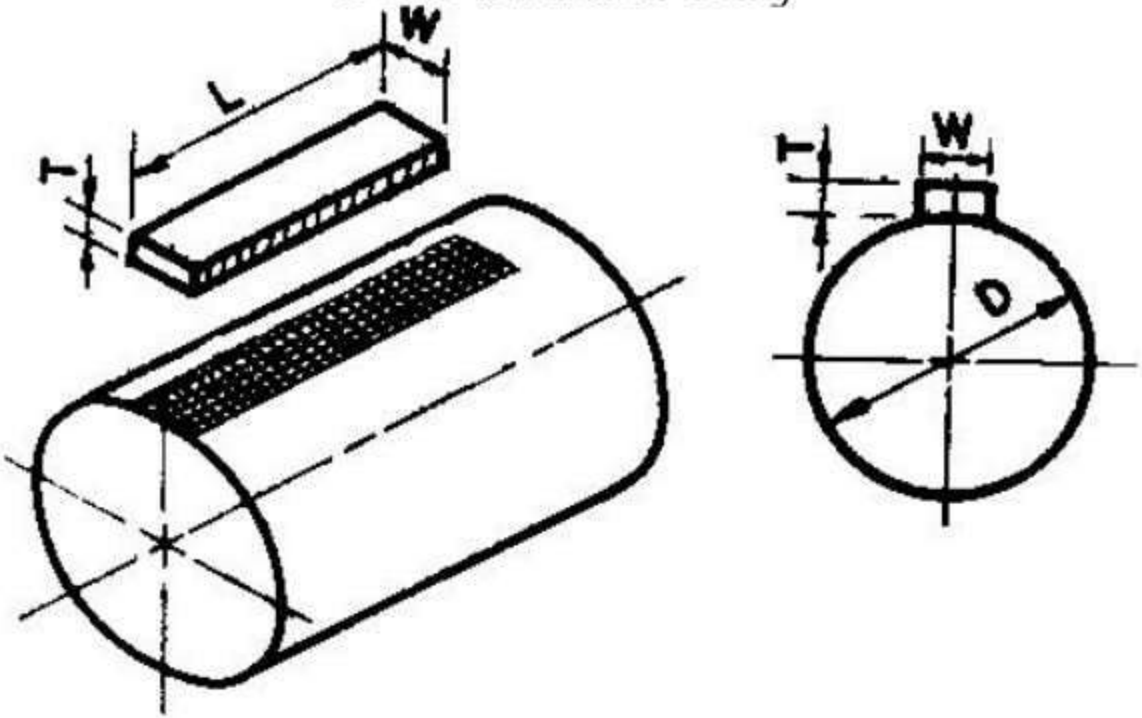
Feather key

- A key attached to one member of a pair and which permits relative axial movement is known as **feather key**.

- It is a special type of parallel key which transmits a turning movement and also permits axial movement.



Flat saddle key



Key design Procedure

Step1: find torque. Using power eqn. **$P=2\pi NT/60$**

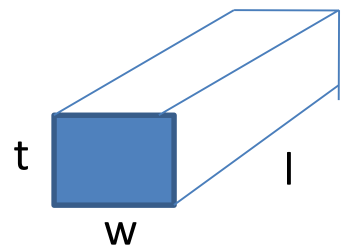
Step2: Calculate diameter of shaft "d" As per Max τ theory., **$T = \pi/16 \times [\tau] \times d^3$**

Step3: Based on d of shaft find key sizes – according to type of key selected.

w=?, t=? And L=?

Step4: Check for induced shear stress

Material of shaft = material of Key



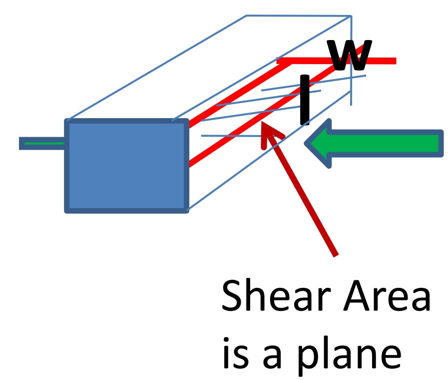
$T= F.R$

σ or $\tau = F/A$, Therefore $F= \tau \times A$

$T=\tau \times A \times R$

Where R= d/2 , d- shaft dia,

A for rectangular key, A = l x w



$T=\tau_{ind} \times l \times w \times d/2$

$\tau_{ind} < [\tau]$ safe design

Kp1. Design a parallel key for the following data . Power= 20Kw at 1200 rpm. The shaft’s allowable shear stress is 50 Mpa.

DATA					
P	20000	w			
N	1200	Rpm			
[τ]	50	N/mm^2			
Key	parallel key				
Step1	find torque. Using power eqn. P=2πNT/60				
		T=	60 x P/(2 x 3.141 x N)		
			159.18	N-m	
		T =	159185	N-mm	

Step2: Calculate diameter of shaft "d" As per Max Ţ theory., T = π/16 x[Ţ] x d^3

			d^3=	????						
			d =	????						
			D^3=	T x[Ţ] x 16/3.141						
			159185	9.815625	d^3					
				16217.50759						
			d=	25.23043327	mm					
pg.7.20/ DDB			d=	28	mm	(R20 series)				

Step3: Based on d of shaft find key sizes – according to type of key selected.

w=?, t=? And L=?

given key = parallel key

Refer page 5.16/DDB

to find sizes

shaft d= 28 mm

for d value , above 22 and upto 30 mm,

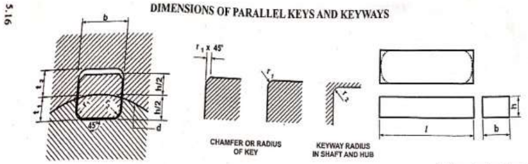
w=	8	mm
t /h=	7	mm

refer pg5.17/DDB

l =	45	mm

5.16

DIMENSIONS OF PARALLEL KEYS AND KEYWAYS



All dimensions in millimeters

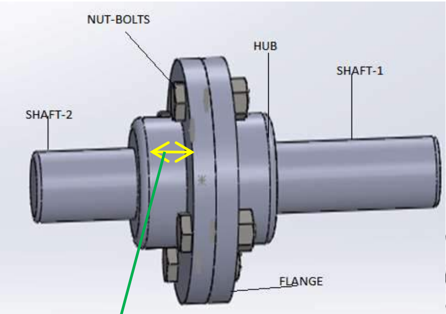
For shaft Diameters	Above	6	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110	130	150	170	200	230	260	290	330	380	440
	Upto	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110	130	150	170	200	230	260	290	330	380	440	500
Key cross Section	Width b	2	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32	36	40	45	50	56	63	70	80	90	100
	Height h	2	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18	20	22	25	28	32	32	36	40	45	50
Key way Depth	In Shaft t_1	1.2	1.8	2.5	3	3.5	4	5	5	5.5	6	7	7.5	8.5	9	10	11	12	13	15	17	19	20	22	25	28	31
(nominal)	In hub t_2	1	1.4	1.8	2.3	2.8	3.3	3.3	3.3	3.8	4.3	4.4	4.9	5.9	5.4	6.4	7.4	8.4	9.4	10.4	11.4	13.4	12.4	14.4	15.4	17.4	19.5
Length of Key L	Min.	6	6	8	10	14	18	22	28	36	45	50	56	63	70	80	90	100	110	125	140	160	180	200	220	250	280
	Max.	20	36	45	56	70	90	110	140	160	180	200	220	250	280	320	360	400	400	400	400	400	400	400	400	400	400

Tolerance on Keywaydepth	t_1	+ 0.1	+ 0.2	+ 0.3
	t_2	+ 0.1	+ 0.2	+ 0.3

Chamfer or Raidus of key r_1	Max.	0.25	0.35	0.55	0.8	1.3	2	2.95
	Min.	0.16	0.25	0.4	0.6	1	1.6	2.5
Key way radius r_2	Max.	0.16	0.25	0.4	0.6	1	1.6	2.5

Designation: A Parallel Key of width 10mm height 8mm and length 50mm shall be designated as : Parallel Key 10 × 8 × 50 IS : 2048 - 1962

DESIGN DATA - PSG TECH



Length of key = Hub length

Hub length=1.5d
Where d= shaft dia.

PREFERRED LENGTHS OF PARALLEL KEYS

Preferred Length (L), mm

Width (b), mm	2	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32	36	40	45	50	56	63	70	80	90	100
Height (h), mm	2	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18	20	22	25	28	32	32	36	40	45	50
	6	6	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	8	8	8	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	10	10	10	10	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	12	12	12	12	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	14	14	14	14	14	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	16	16	16	16	16	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	18	18	18	18	18	18	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	20	20	20	20	20	20	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	22	22	22	22	22	22	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	25	25	25	25	25	25	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	28	28	28	28	28	28	28	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	32	32	32	32	32	32	32	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	36	36	36	36	36	36	36	36	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
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	-	-	-	50	50	50	50	50	50	50	50	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	56	56	56	56	56	56	56	56	56	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	63	63	63	63	63	63	63	63	63	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	70	70	70	70	70	70	70	70	70	70	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	80	80	80	80	80	80	80	80	80	80	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	90	90	90	90	90	90	90	90	90	90	90	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	100	100	100	100	100	100	100	100	100	100	100	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	110	110	110	110	110	110	110	110	110	110	110	110	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	-	125	125	125	125	125	125	125	125	125	125	125	125	-	-	-	-	-	-	-
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	-	-	-	-	-	-	-	-	160	160	160	160	160	160	160	160	160	160	160	160	160	-	-	-	-	-
	-	-	-	-	-	-	-	-	-	180	180	180	180	180	180	180	180	180	180	180	180	180	-	-	-	-
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	-	-	-	-	-	-	-	-	-	-	-	220	220	220	220	220	220	220	220	220	220	220	220	220	-	-
	-	-	-	-	-	-	-	-	-	-	-	-	250	250	250	250	250	250	250	250	250	250	250	250	250	-
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	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	360	360	360	360	360	360	360	360	360	360	360
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	400	400	400	400	400	400	400	400	400	400

Step4: Check for induced shear stress

Material of shaft = material of Key
[Ʈ] shaft= [Ʈ] key also

T=	F x R
F=	stress x A
T=	stres x A x D/2
A=	l x w
T=	l x w x Ʈ x d/2
T=	45 x 8 xƮ x 28/2
Ʈ inducd: 31.6	N/mm^2 [Ʈ]=50
31.584 < [50]	

design is safe

Couplings

To transmit **motion** from driver to driven, act as connector between them

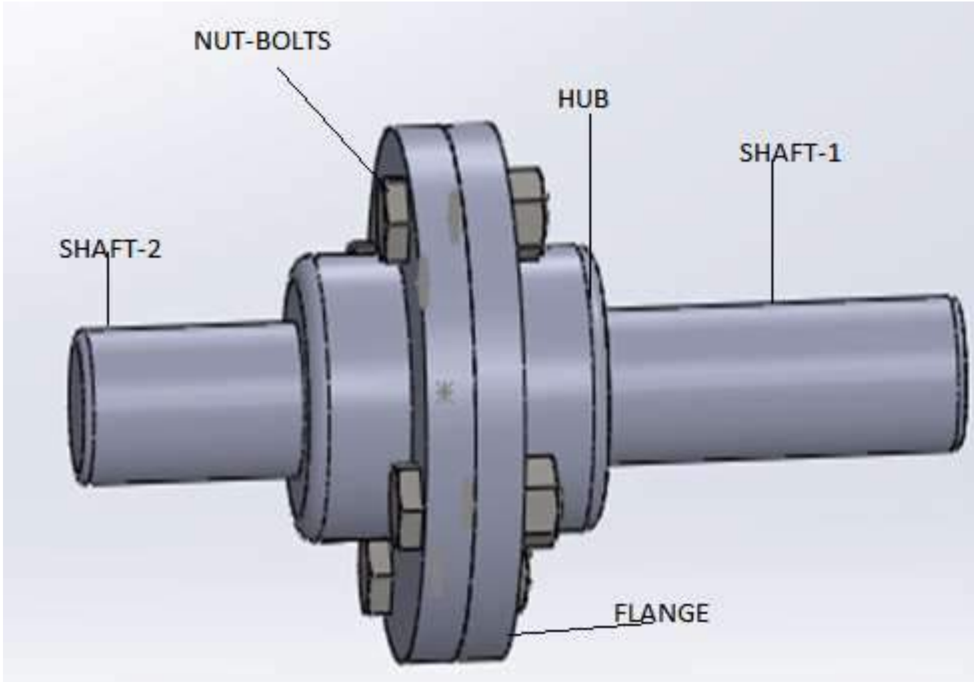
Types

1. Rigid flange coupling ----a. Protective type
   b. non protective

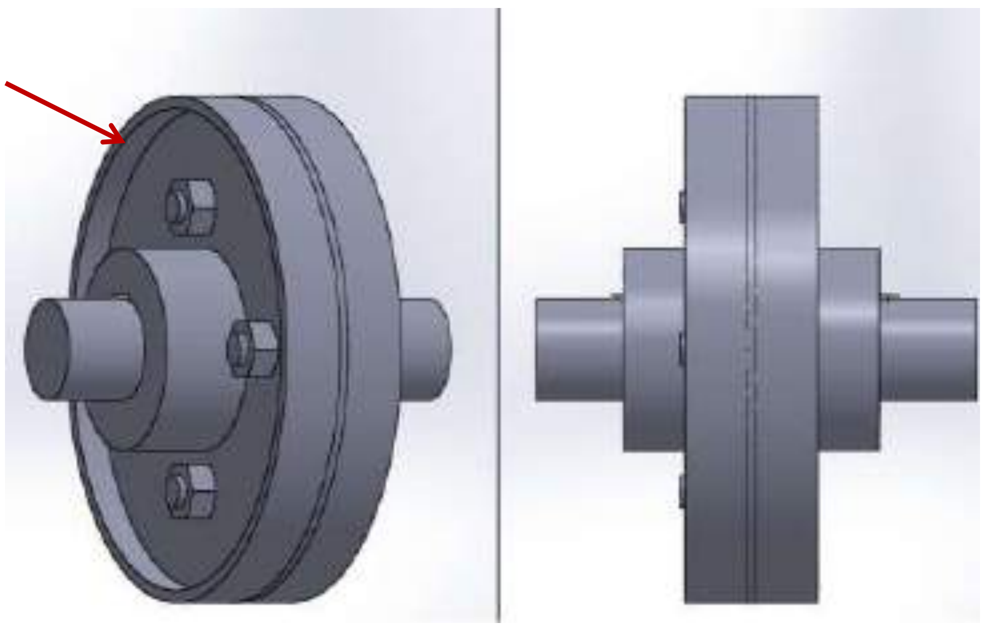
2. Flexible coupling– Bushed pin type

Clutch and coupling:

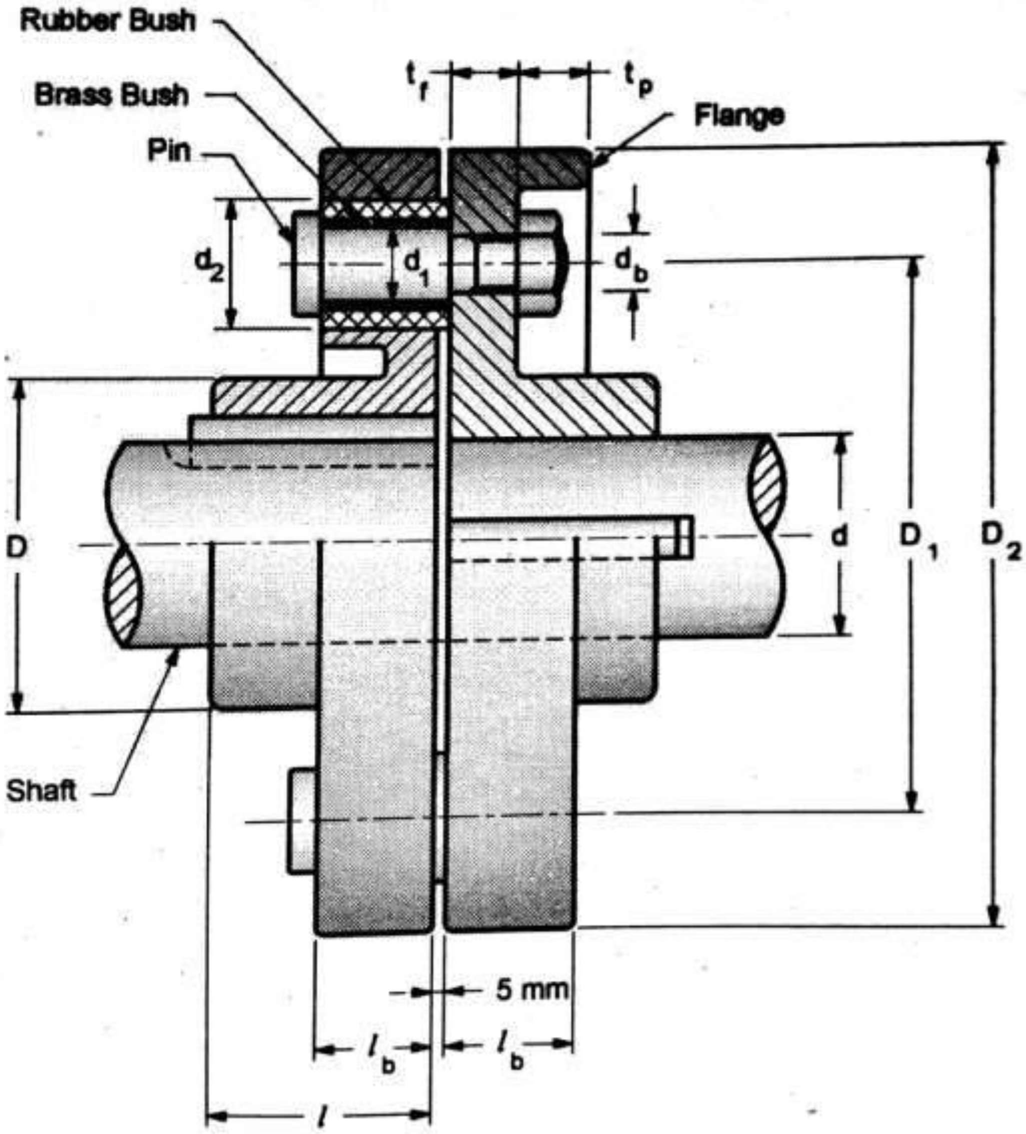
both functions are same , but clutch is distinguished by the transmission of motion can be intermittent . that is, when desired to stop or start is possile by disengage or enage the clutch.



Non protective flange



Protective flange



Bushed Pin-Flexible coupling

Design procedure for Flange coupling (protective and non protective)

Step1: Find Torque using power eqn.

Step2: Find ‘ d’ of the shaft using Max. T equation

Step3: List the basic sizes of the coupling using ‘ d’ of the shaft refer Pg. 7.134/DDB

Step4: design for hub(treating it as hollow shaft, di= d of the shaft, do=2d

Step5: Design for key

Step6: Design for flange

Step7:Design for bolt

Step8: Draw the coupling with NTS free hand sketching with dimensions

CP1. Design and draw a rigid coupling for the following specifications.
Power 15 kW at 300 rpm. Allowable shear stress for Shaft and Key is 40 N/mm^2. Bolts working stress should not exceed 30 N/mm^2. Flange is made of Castiron and its limited shear stress is 14 N/ mm^2. The toque transmission is 25% higher than the actual torque. The crushing stress for key is 2.5 times of its shear stress.

DATA			
P	15000	Watts	
N	300	rpm	
[T] shfat& key =		40	N/mm^2
bolts working stress=		30	N/mm^2
crushing is thrice the Shear stress			
CI for FLANGE[T] =		14	N/mm^2
Tmax =	1.25 T		

Step1 : Torque finding

P= 2 x 3.141 x N x T/ 60

Some times service factor will be given,

T= service factor x Tcal.

T= p x 60/(2 x 3.141 x N)

P=	(2x3.141xN x T)/60	
T=	477.5549188	Nm
	477554.9188	N-mm
Tmax =	596943.6485	N-mm

(Tmax = 1.25 x T cal)

Step2: Find diameter of The shaft

T= 3.141/16 x Ʈ x d^3

Std the " d " to R20 series

	D^3=	T x 16/(3.141 x Ʈ)	
		76019.567	
	d =	42.203469	mm
R20 series	d=	45	mm

Step3: List the basic proportions of coupling use d std

Refer Pg.7.134 /DDB

d = dia of the shaft

D= 2d (d0 = outter dia of the hub),

di= inner dia of the hub= dia of theshaft

L=1.5 d hub length

D1=PCD= 3d

D2 = flange dia= 4 d

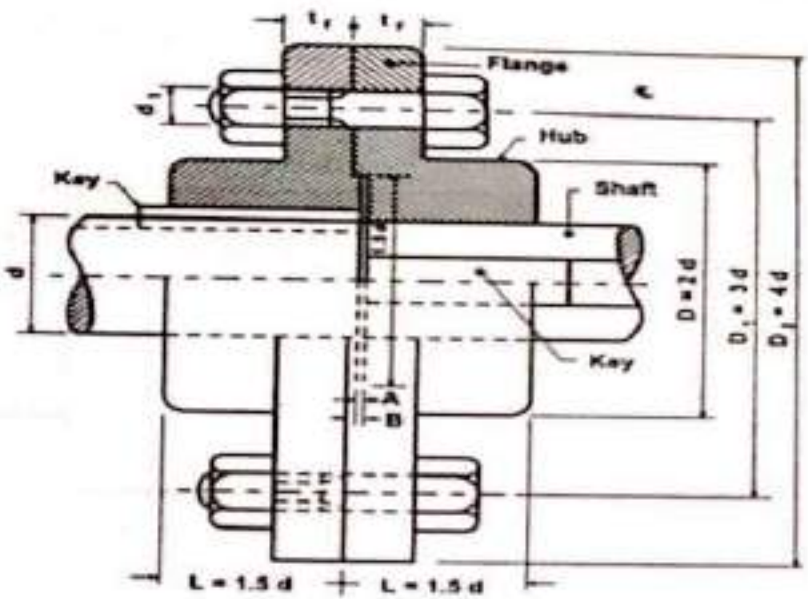
n = no.of bolts selecte according to d of the shaft

Tf= thickness of flange= d/2

Refer pg.7.134/DDB				
		shaft d=	45	mm
	HUB	di =	45	mm
		do =	90	mm
		L of Hub=	67.5	mm
	PCD for bolts =		135	mm
	Flange dia =		180	mm
	No.of bolts n=	(40<d<100)	4	nos

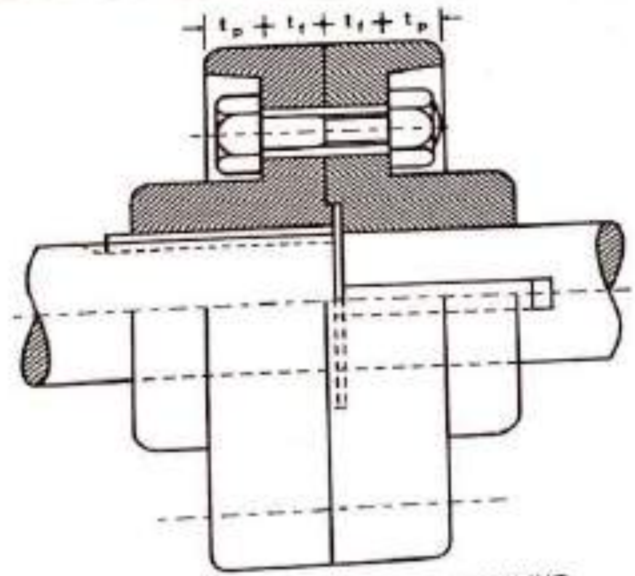
FLANGE COUPLINGS

Equation	Nomenclature
$D = 2d$	d_1 Nominal diameter of bolts
$L = 1.5d$	d Shaft diameter
$D_1 = 3d$	D Outside diameter
$D_2 = 4d$	D_1 Diameter of bolt circle
$t_f = \left(\frac{d}{2}\right)$	n number of bolts
n = 3 for d upto 40 mm = 4 for d upto 100 mm = 6 for d upto 180 mm	t_f Thickness of flange
$t_p = \left(\frac{d}{4}\right)$	t_p Thickness of protecting flange



UNPROTECTED TYPE FLANGE COUPLING

	Equation	Nomenclature
HUB	$T = \frac{\pi}{16} \tau_c \left[\frac{D^4 - d^4}{D}\right]$	τ_s Allowable shear stress for shaft
KEY	$T = l\, w\, \tau_k \left(\frac{d}{2}\right)$	τ_b Allowable shear stress for bolt
KEY	$T = \left(\frac{t}{2}\right) \sigma_c \left(\frac{d}{2}\right)$	τ_k Allowable shear stress for key
FLANGE	$T = \pi \left(\frac{D^2}{2}\right) \tau_c\, t_f$	τ_c Allowable shear stress for flange
BOLTS	$T = n \left(\frac{\pi}{4}\right) d_1^2\, \tau_b \left(\frac{D_1}{2}\right)$	σ_{cb} Allowable shear stress for crushing stress for bolt
BOLTS	$T = n\, d_1\, t_f\, \sigma_{cb} \left(\frac{D_1}{2}\right)$	σ_{ck} Allowable shear stress for crushing stress for key
		T Torque transmitted by the coupling

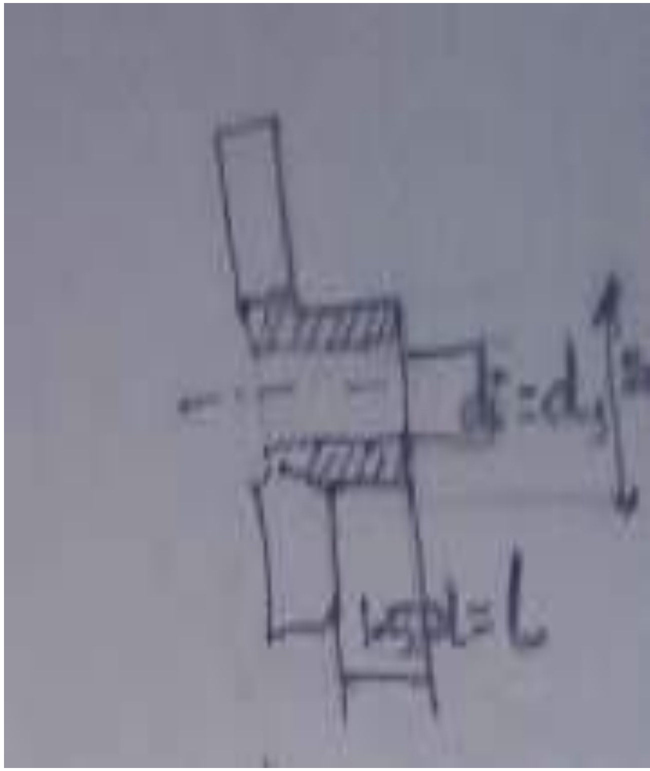


PROTECTIVE TYPE FLANGE COUPLING

Step 4: Hub design

T= 3.141/16 x τ x [(do^4-di^4)/d0]

hollow shaft with torsion only				
[τ] =	40	N/mm^2		
T=	(π/16) xτind x[(do^4-di^4)/d0]			
(π/16)	0.196313			
(do^4-di^4)	61509375			
(do^4-di^4)/d0)	683437.5			
T=	596943.6			
596943.6485	134167.3	τind		
τind =	4.449	N/mm^2		
4.449 < [40]				
τind < [τ]	safe design			

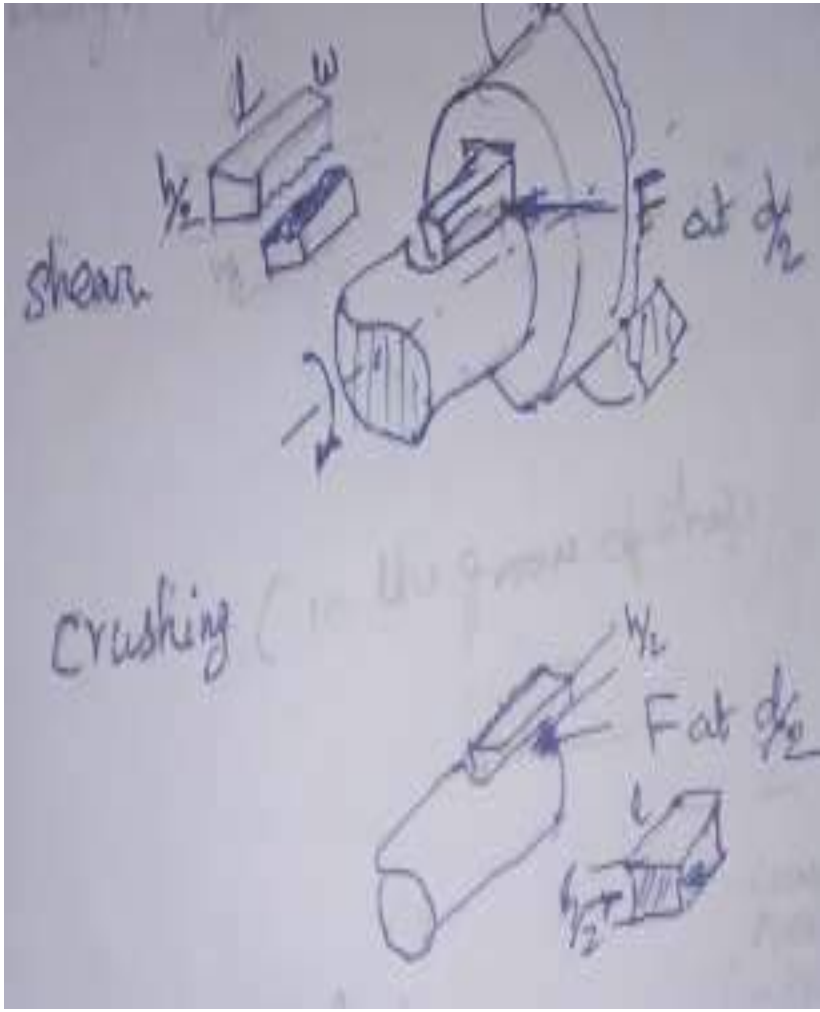


STEP.5 Key Design

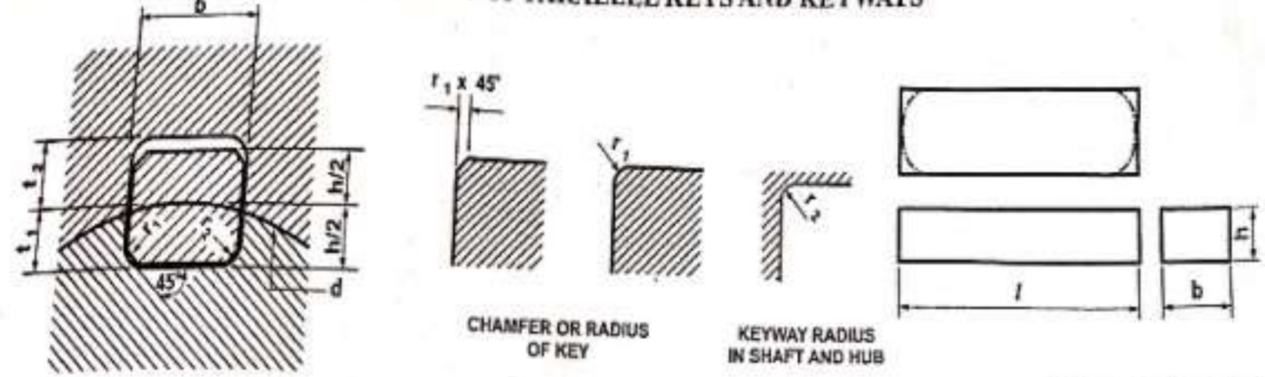
In shear

In crushing

	Rectangular Key (Assumed)		parallel
	Shaft d=	45	mm
	Refer Pg.5.16 & 5.17		
	above	44	mm
	upto	50	mm
	b =	14	mm
	h =	9	mm
	l = hub length	67.5	
Pg.5.17/DDB	prefered length	70	mm



DIMENSIONS OF PARALLEL KEYS AND KEYWAYS



All dimensions in millimeters

For shaft Diameters	Above	6	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110	130	150	170	200	230	260	290	330	380	440
	Upto	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110	130	150	170	200	230	260	290	330	380	440	500
Key cross Section	Width b	2	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32	36	40	45	50	56	63	70	80	90	100
	Height h	2	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18	20	22	25	28	32	32	36	40	45	50
Key way Depth (nominal)	In Shaft t_1	1.2	1.8	2.5	3	3.5	4	5	5	5.5	6	7	7.5	8.5	9	10	11	12	13	15	17	19	20	22	25	28	31
	In hub t_2	1	1.4	1.8	2.3	2.8	3.3	3.3	3.3	3.8	4.3	4.4	4.9	5.9	5.4	6.4	7.4	8.4	9.4	10.4	11.4	13.4	12.4	14.4	15.4	17.4	19.5
Tolerance on Keywaydepth	t_1	+0.1				+0.2										+0.3											
	t_2	+ 0.1									+ 0.2									+ 0.3							
Chamfer or Raidus of key r_1	Max.	0.25			0.35			0.55					0.8					1.3				2			2.95		
	Min.	0.16			0.25			0.4					0.6					1				1.6			2.5		
Key way radius r_2	Max.	0.16			0.25			0.4					0.6					1				1.6			2.5		
Length of Key L	Min.	6	6	8	10	14	18	22	28	36	45	50	56	63	70	80	90	100	110	125	140	160	180	200	220	250	280
	Max	20	36	45	56	70	90	110	140	160	180	200	220	250	280	320	360	400	400	400	400	400	400	400	400	400	400

Designation: A Parallel Key of width 10mm height 8mm and length 50mm shall be designated as : Parallel Key 10 × 8 × 50

IS : 2048 - 1962

PREFERRED LENGTHS OF PARALLEL KEYS

Width (b), mm	2	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32	36	40	45	50	56	63	70	80	90	100
Height (h), mm	2	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18	20	22	25	28	32	32	36	40	45	50
Preferred Length (L), mm	6	6	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	8	8	8	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	10	10	10	10	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	12	12	12	12	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	14	14	14	14	14	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	16	16	16	16	16	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	18	18	18	18	18	18	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	20	20	20	20	20	20	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	22	22	22	22	22	22	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	25	25	25	25	25	25	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	28	28	28	28	28	28	28	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	32	32	32	32	32	32	32	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	36	36	36	36	36	36	36	36	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	40	40	40	40	40	40	40	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	45	45	45	45	45	45	45	45	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	50	50	50	50	50	50	50	50	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	56	56	56	56	56	56	56	56	56	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	63	63	63	63	63	63	63	63	63	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	70	70	70	70	70	70	70	70	70	70	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	80	80	80	80	80	80	80	80	80	80	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	90	90	90	90	90	90	90	90	90	90	90	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	100	100	100	100	100	100	100	100	100	100	100	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	110	110	110	110	110	110	110	110	110	110	110	110	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	-	125	125	125	125	125	125	125	125	125	125	125	125	-	-	-	-	-	-	-
	-	-	-	-	-	-	-	140	140	140	140	140	140	140	140	140	140	140	140	140	-	-	-	-	-	-
	-	-	-	-	-	-	-	-	160	160	160	160	160	160	160	160	160	160	160	160	160	-	-	-	-	-
	-	-	-	-	-	-	-	-	-	180	180	180	180	180	180	180	180	180	180	180	180	180	-	-	-	-
	-	-	-	-	-	-	-	-	-	-	200	200	200	200	200	200	200	200	200	200	200	200	200	-	-	-
	-	-	-	-	-	-	-	-	-	-	-	220	220	220	220	220	220	220	220	220	220	220	220	220	-	-
	-	-	-	-	-	-	-	-	-	-	-	-	250	250	250	250	250	250	250	250	250	250	250	250	250	-
	-	-	-	-	-	-	-	-	-	-	-	-	-	280	280	280	280	280	280	280	280	280	280	280	280	280
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	320	320	320	320	320	320	320	320	320	320	320	320
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	360	360	360	360	360	360	360	360	360	360	360
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	400	400	400	400	400	400	400	400	400	400
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

a.) SHEAR failure

T= F x R

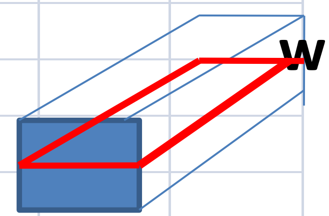
R= the tangential force acting at peripery of shaft
because the key(half key) is located at shaft

Stress= F/A Therfeore, F= stress x A

T= Stress x Area X R

A= l x W	980	mm^2
T =	596943.6	N-mm
R=	22.5	mm
596943.6485	22050	τind
τind=	27.07227	

27.0722 <[40]
τind< [τ] Safe design



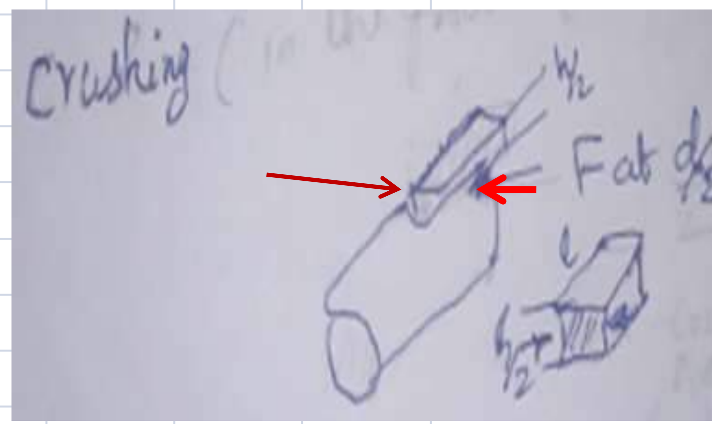
b.) crushing failure

T= F x R

R= the tangential force acting at peripery of shaft
because the key(half key) is located at shaft

Stress= F/A Therfeore, F= stress x A

A =	l x h/2		(crushing occurs in contained material zone)
			(half in shaft key way only to be considered)
A =	315	mm^2	
T=	596943.6	N-mm	
R=	22.5	mm^2	
596943.6485	7087.5	σc	
σc =	84.22485		



To find [σc]

[σc]= 2.5 Ʈ
[σc]= 100 N/mm^2

84.224 < [100]

σc ind < [σc] Safe design

Step 5: Flange design

Flange is made of CI- cast Iron material

Flange failure occurs at the junction of HUB

T= F x R

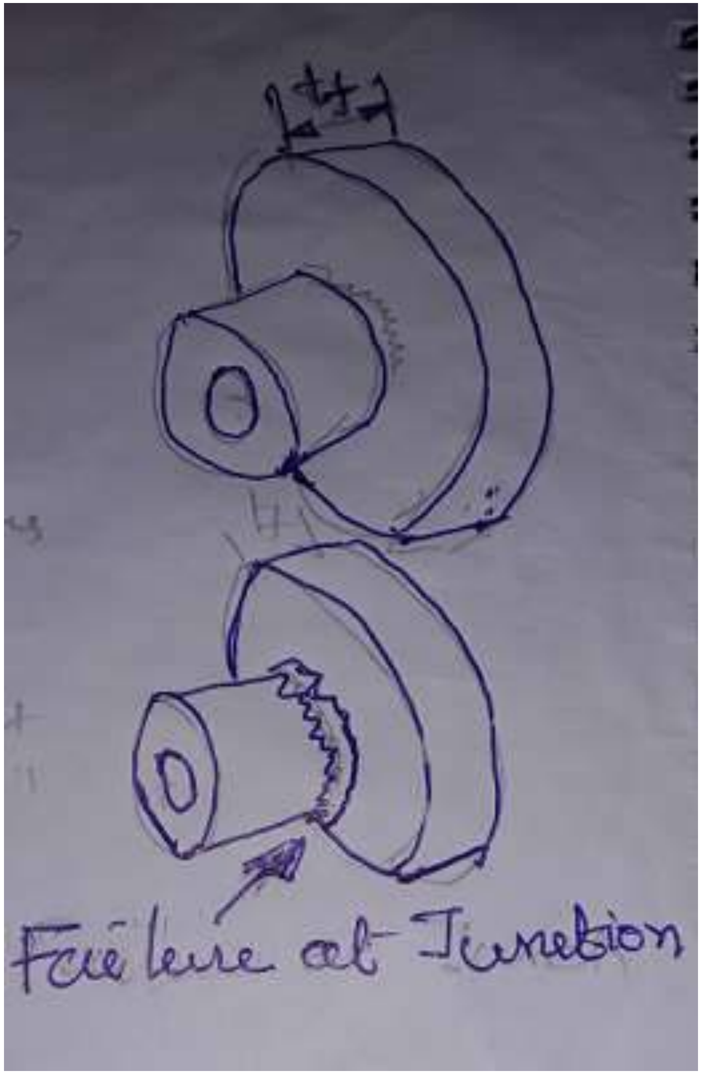
T= Stess x A x R

for ,R, the hub diameter "do" to be selected

becuase The tangential force is taken at peripery of HUB

tf= Thck ness of Flage= 0.5 d

tf= 22.5 mm

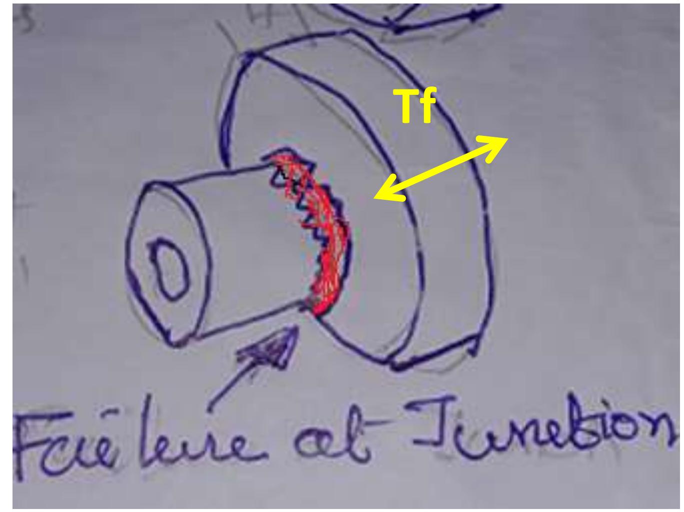


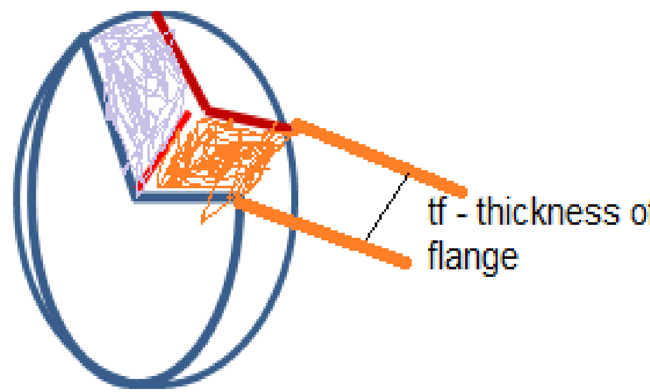
A= π x do x tf	6360.525	mm^2

Failure takes around it circumference,
at the **junction** of Hub and Flange

Circumference = πDo
A = = $\pi Do\ tf$

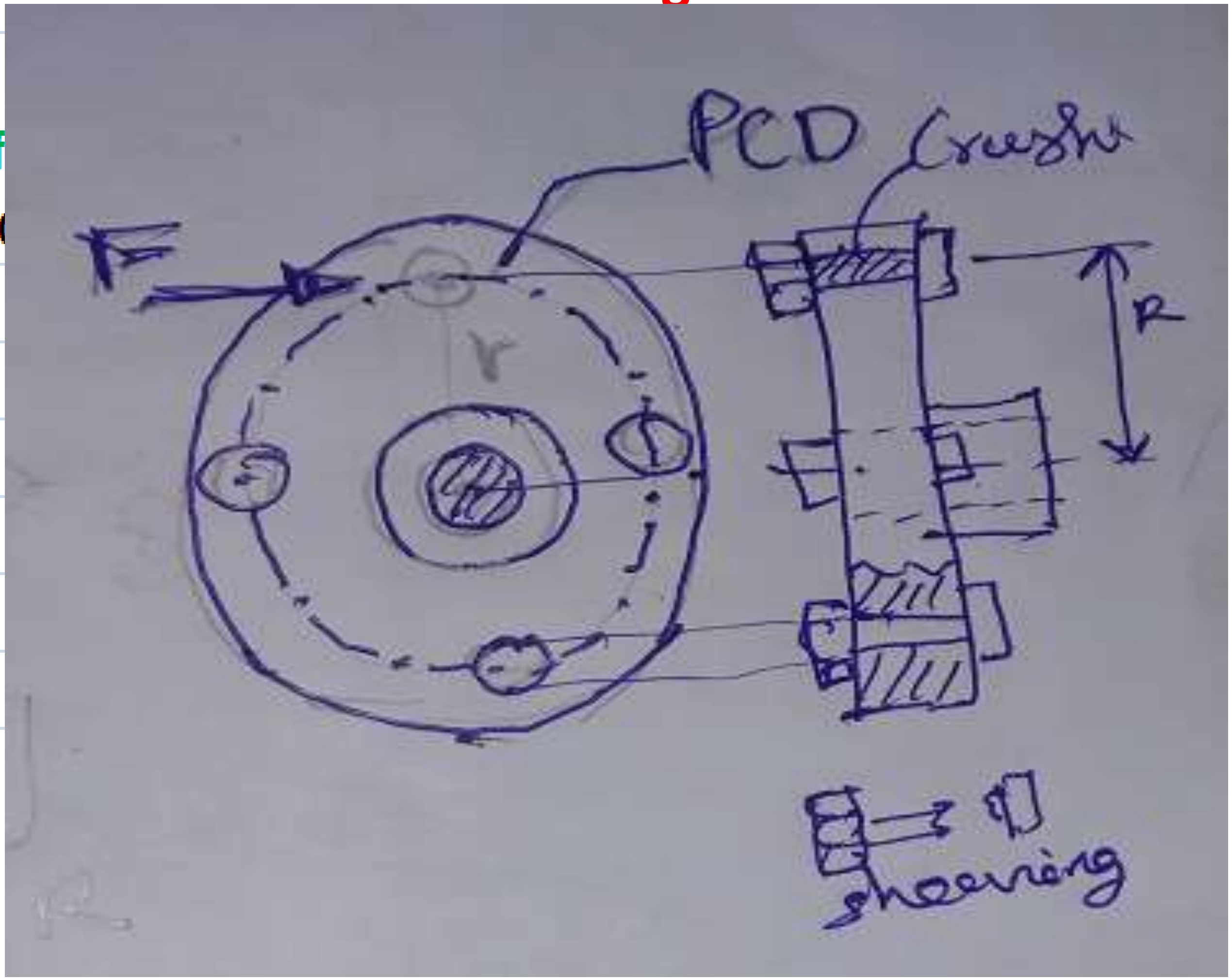
A= π x do x tf	6360.525	mm^2	
T =	596943.6	N-mm	
R=	45	mm	
596943.6485 =	286223.6	Tind	
Tind =	2.085585	N/mm^2	
2.08 <[14]	**safe design**		





Design of bolts in shear and crushing

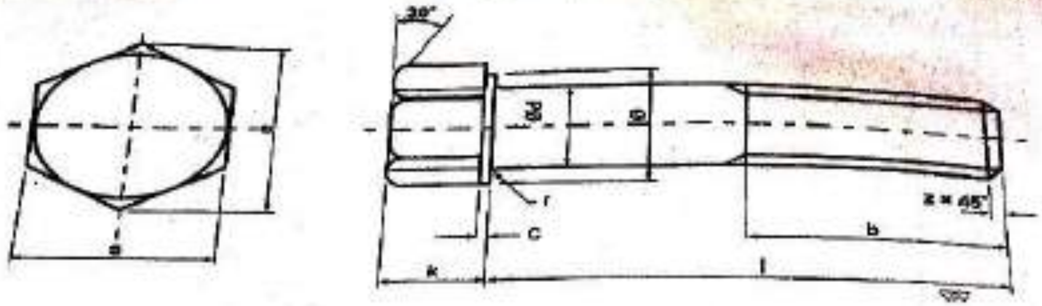
$T = F \times R$	
(R, the radial distance of	
Tangential force acts at P(	
$F =$ stress x A	
$A = \pi/4 \times d^2$	
no.of .bolts= 4	
$T = [\tau].(\pi/4 \times d^2).n.R$	



T =	596943.6	N-mm	
[τ] =	30	N/mm^2	
n	4	nos	
R = (PCD value)	67.5	mm	
d=	???		
596943.6485	=	6360.5	d^2
d^2 =	93.85132		
d =	9.687689		
Ref.pg.5.49/DDB, std,			
d minimum= 9.78 to d max		M10 selected	
		d= 10 mm	

Precision (P) Grade

M5 - M30



All dimensions in mm

Size		M 5	M 6	M 8	M 10	M 12	M 16	M 20	M 24	M 30
d	Max.	5	6	8	10	12	16	20	24	30
	Min.	4.82	5.82	7.78	9.78	11.73	15.73	19.67	23.67	29.67
S	Max.	8	10	13	17	19	24	30	36	46
	Min.	7.85	9.78	12.73	16.73	18.67	23.67	29.67	35.38	45.38
k	Max.	3.65	4.15	5.65	7.18	8.18	10.18	13.22	15.22	19.26
	Min.	3.35	3.85	5.35	6.82	7.82	9.82	12.78	14.78	18.74
b Min. for l ≤ 130		16	18	22	26	30	38	46	54	66
for 130 < l < 300		-	24	28	32	36	44	52	60	72
for l > 200		-	-	-	-	-	57	65	73	85
a	Min.	7.2	9	11.7	15.3	17.1	21.6	27	32.4	41.4
C	~	0.2	0.3	0.4	0.4	0.4	0.4	0.4	0.5	0.5
e	Min.	8.87	11.05	14.38	18.9	21.1	26.75	33.53	39.98	51.28
r	Min.	0.2	0.25	0.4	0.4	0.6	0.6	0.8	0.8	1
z	Max.	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5
l		30 - 75	30 - 80	30 - 90	35 - 120	40 - 120	50 - 220	60 - 220	70 - 220	90 - 220

IS : 2389

IS : 1364

Crushing failure of Bolt

Crushing = straining of material in confined zone

σc = 2.5 Ʈ

σc ind < [σc]

[σc] = 2.5 Ʈ

[σc] = ~~100~~ N/mm^2

75

T= F X R

T=Stress x A x R

(Here the R, , tangetial force acts at PCD of flange,

n=	4	Nos	
T=	596944	N-mm	
A = n x d xtf			
d=	10	mm	
tf=	22.5	mm	
n =	4	nos	
R= pcd/2	67.5	mm	
A= 4 x 10 x 2	900	mm^2	

T = n x dx tf x σc x pcd/2				
596943.65	=	60750	σcind	
σcind	9.8262			
9.8 < ~~[100]~~	Safe design			
[75]				

Flexible couplings

Types

1. Bushed Pin type
2. Oldham's couplings
3. Universal coupling

Why?

1. Abutting ends of shaft to be connected
2. Not exact alignment of shafts
3. Permits axial misalignment without power loss at the shaft

Design procedure for
(Bushed pin flexible coupling)

Step1: Find Torque using power eqn.

Step2: Find ‘ d’ of the shaft using Max. T equation

Step3: List the basic sizes of the coupling using ‘ d’ of the shaft refer Pg. 7.134/DDB

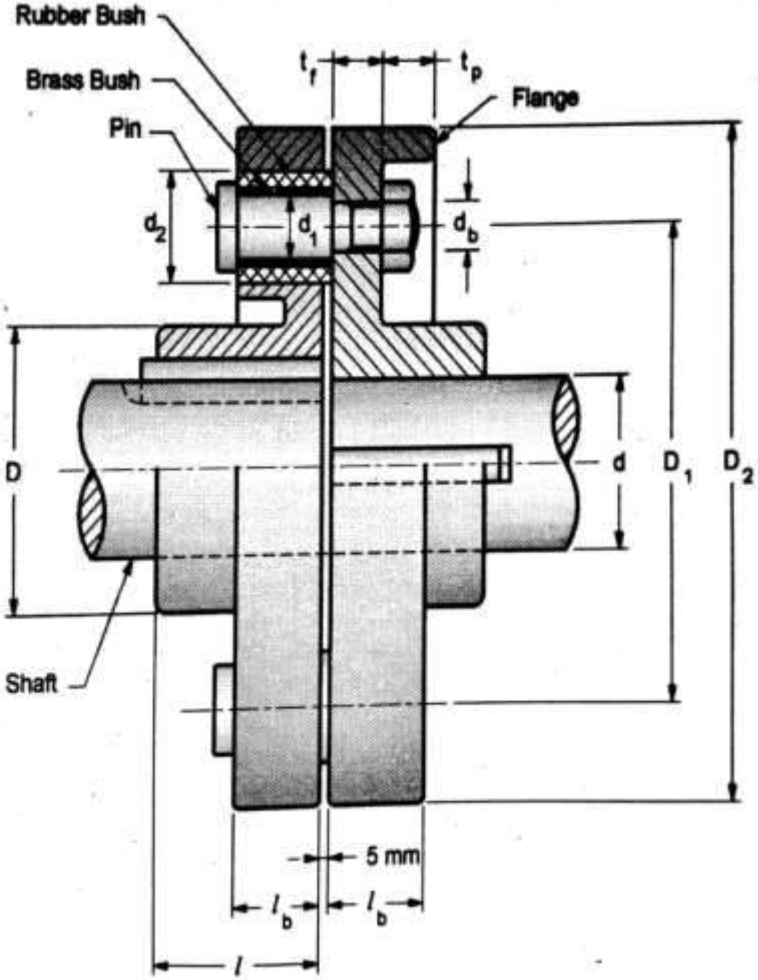
Step4: design for hub(treating it as hollow shaft, di= d of the shaft, do=2d

Step5: Design for key

Step6: Design for flange

Step7:Design for PIN(instead of Bolt

Step8: Draw the coupling with NTS free hand sketching with dimensions



HUB

	Equation	Nomenclature
HUB	$T = \frac{\pi}{16} \tau_c \left[\frac{D^4 - d^4}{D}\right]$	τ_s Allowable shear stress for shaft
KEY	$T = l w \tau_k \left(\frac{d}{2}\right)$	τ_b Allowable shear stress for bolt
KEY	$T = \left(\frac{t}{2}\right) \sigma_c \left(\frac{d}{2}\right)$	τ_k Allowable shear stress for key
FLANGE	$T = \pi \left(\frac{D^2}{2}\right) \tau_c t_f$	τ_c Allowable shear stress for flange
BOLTS	$T = n \left(\frac{\pi}{4}\right) d_1^2 \tau_b \left(\frac{D_1}{2}\right)$	σ_{cb} Allowable shear stress for crushing stress for bolt
BOLTS	$T = n d_1 t_f \sigma_{cb} \left(\frac{D_1}{2}\right)$	σ_{ck} Allowable shear stress for crushing stress for key
		T Torque transmitted by the coupling

Pg.7.135/DDB

CP2. Design and draw a flexible coupling for the following specifications.

Power 32 kW at 960 rpm. Allowable shear stress for Shaft and Key is 40 N/mm^2. Bolts working stress should not exceed 30 N/mm^2. Flange is made of Castiron and its limited shear stress is 15 N/ mm^2. The toque transmission is 20% higher than the actual torque. The crushing stress for key 80 N/mm^2.

DATA				
P	32000	Watts		
N	960	rpm		
[T] shfat& key	=	40	N/mm^2	
bolts working stress=		30	N/mm^2	
crushing is		80	N/mm^2	
CI for FLANGE[T] =		15	N/mm^2	
Tmax =	1.20 T			

STEP1	Toruque Finding		
	P=	(2x3.141xN x T)/60	
	T=		Nm
			N-mm
	Tmax =		N-mm

STEP2	Shaft dia 'd calculating			
	T=	**3.141/16 x Ʈx d^3**		
	D^3=	**T x 16/(3.141 x Ʈ)**		
	d =		mm	
R20 serie	**d=**		**mm**	

STEP 3	LIST the BASIC dimensions						
Refer pg.7.134/DDB							
		shaft d=		mm			
	HUB	di =		mm			
		do =		mm			
		L of Hub=		mm			
	PCD for bolts =			mm			
	Flange dia =			mm			
	No.of bolts n=	(40<d<100)		nos	but use pin in case 6		

STEP4	HUB design and check Ţind < [Ţ]				
	hollow shaft with torsion only				
	[Ţ] =	40	N/mm^2		
	T=	(π/16) xŢind x[(do^4-di^4)/d0]			
	(π/16)				
	(do^4-di^4)				
	(do^4-di^4)/d0)				
	T=				
	382043.9351	94230	Ţind		
	Ţind =	4 [illegible]	^2		
	Ţind < [Ţ]	safe design			

STEP 4	KEY deisgn			
	Rectangular Key (Assumed)		parallel keys	
	Shaft d=	40	mm	
	Refer Pg.5.16 & 5.17			
	above	38	mm	
	upto	44	mm	
	b =	12	mm	
	h =	8	mm	
	l = hub length	60		
Pg.5.17	prefered length =	63	mm	

5.17

a.) SHEAR failure			
	T= F x R		
		R= the tangential force acting at peripery of sl	
		because the key(half key) is located at shaft	
		Stress= F/A	Therfeore, F= stress x A
	T= Stress x Area X R		
	A= l x W	756	mm^2
	T =	382043.94	N-mm
	R=	20	mm
	382043.9351	15120	Ţind
	Ţind=	25.267456	
	25.267		
	~~27.0722~~ <[40]		
	Ţind< [Ţ]	Safe design	

b.) crushing failure

T= F x R

R= the tangential force acting at peripery of shaft
because the key(half key) is located at shaft

Stress= F/A Therfeore, F= stress x A

A = l x h/2 (crushing occurs in contained material zone)
(half in shaft key way only to be considered)

A =	252	mm^2
T=	382043.94	N-mm
R=	20	mm
382043.9351	5040	σc
σc =	75.802368	

To find [σc]

[σc]=	80	N/mm^2

75.802 < [80]
σc ind < [σc] Safe design

STEP5 Design of Flange

Flange is made of CI- cast Iron material

Flange failure occurs at the junction of HUB

T= F x R

T= Stess x A x R

for ,R, the hub diameter "do" to be selected
becuase The tangential force is taken at peripery of HUB

tf= Thck ness of Flage= 0.5 d		
tf=	20	mm
A= π x do x tf	5025.6	mm^2
T =	382043.94	N-mm
R=	40	mm
~~596943.6485~~ =	201024	Tind
Tind =	1.9004892	N/mm^2
1.900 [15]		
~~2.08~~ <[14] safe design		

STEP6 Design of (bolts) Pin

T= F x R

(R, the radial distance of the tangential force, which acts at PCD of The flange)

Tangential force acts at PCD of The flange

F= stress x A

$A = \pi/4 \times d^2$

no.of .Pins= 6 always

Pin is Sujected to Torsion & Bending

Dia of the Pin =?

$$D_{pin} = 0.5 \times d/\sqrt{n}$$

Where d= shaft diameter

n = 6

Direct torsion

Ʈ= W/A

A= π/4 x d^2

W= looad on the pin - Pressure on the pin

p= 0.8 N/mm^2

Bending Stress

Pin trreated As UDL loaded shaft

σ= M/Z

M= (W x l)/2

w= P x d x l ← P= w/ A

d- dia of Pin

l- length of the pin

Z- section modulus of circular Pin

Z= π/64 x d^4/(d/2)

[z= I/y]

Apply prinicpal stress equation

$\sigma 1,2=?$

$\sigma 1<[\tau]$ prove it

T =	382043.9351	N-mm			
R=	120	mm			
P=	0.8	N/mm^2			
W=	Load on the pin due to pressue				
T=	n xW x R				
Due tp Pressure					
W=	P x A(Projected area)				
	A= Dpin x l				
		Pin dia=	8.16497	8.1650	
	Dpin = 2dpin	(secured in the left flange enlarged portion)			
	Dpin =	32.33 mm		(2dpin)	(add rubber bush thick= 6 x 2=12 & brass bush thick = 2 x 2= 4 in mm)
					Totally= 12+4=16
	l=tf	in general			
	L=	25 mm			

w=	646.5986324	
Wtotal=	3879.591794	(6 x W)
Now		
Shear Stress calculation		
τ= W/A		
W=	3879.591794	N
A= π/4*Dpin^2		
A=	820.7625217	mm^2
τ=	4.726814019	N/mm^2

Now Bendng stress calculation			
σ= M/Z			
[z= I/y]			
M= (W x l)/2		(udl & simply supported beam)	
I= π/64 x Dpin^4			
Y= Dpin/2			
Z=	3316.899525	mm^3	
M=	48494.89743	N-mm	
σb=	14.62	N/mm^2	

Now Apply Pricipal stress equation, σ1,2

σb=σx	Ţxy= Ţ		
Ref.Pg.7.2 /DDB			
σb=σy=	14.62	N/mm^2	
Ţxy= Ţ=	4.73	N/mm^2	
σ1,2= (σx+σy)/2 ±1/2√(σx-σy)^2 + 4. Ţ^2			

RHS		
σy/2=	7.310	
σy^2/2=	106.880	
(4. Ţ^2)/2 =	44.686	
Sqrt(√(σx-σy)^2 + 4. Ţ^2)=		

17.410

8.707

Now

(σx+σy)/2 ±1/2√(σx-σy)^2 + 4. Ʈ^2 = 16.017 N/mm^2

16.017 N/mm^2

σ1= N/mm^2

16.017 <[40] Safe design

But aspee max.shear stress theory ,

Ʈmax = ±1/2√(σx-σy)^2 + 4. Ʈ^2

Ʈmax = 8.707 N/mm^2

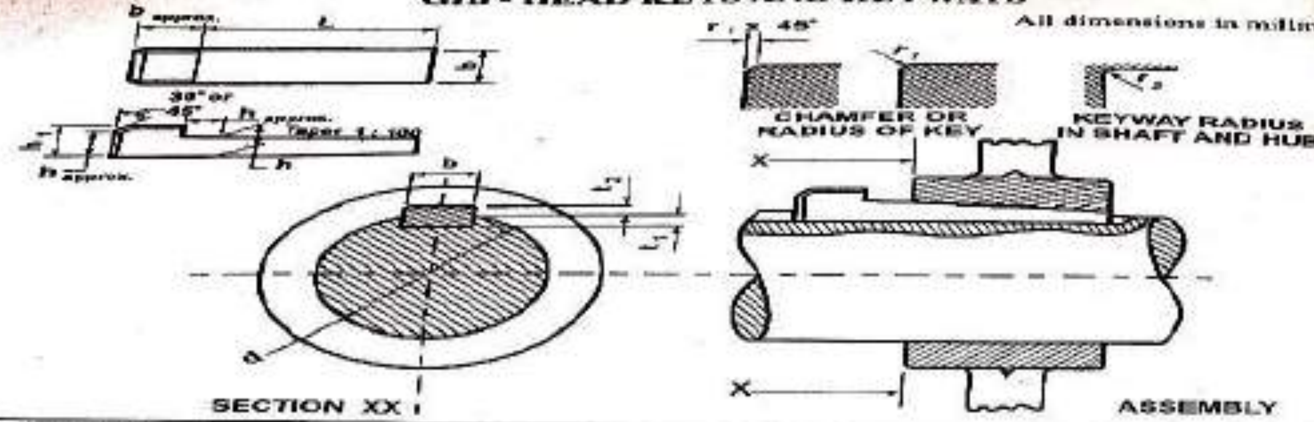
8.707 2<[40] safe design

key design

1	Taper Key		5.21/5.22 DDB
2	Gib-Head Key		5.19/5.20 DDB
3	Tangential Key		
	Power=	40000	Watts
	N=	100	rpm
	stress	20	N/mm^2

GIB - HEAD KEYS AND KEYWAYS

All dimensions in millimetres



SHAFT DIAMETER, d		KEY					KEYWAY IN SHAFT AND HUB					
Above	Up to and including	Width b (h 9)	Height, h (nominal)	Tolerance on h	Height of Gib-head h_1	Chamfer or Radius r_1 (Min.)	Width of Key way (D10)	Depth in Shaft, t_1	Tol. on t_1	Depth in Hub, t_2	Tol. on t_2	Radius at Bottom of Key way r_2 (max)
10	12	4	4	+0.1	7	0.16	4	2.5	+0.1	1.2	+0.1	0.16
12	17	5	5		8	0.25	5	3		1.7		0.25
17	22	6	6		10		6	3.5		2.1		
22	30	8	7		11		8	4		2.5		
30	38	10	8	+0.2	12	0.40	10	5		2.5		0.40
38	44	12	8		12		12	5		2.5		
44	50	14	9		14		14	5.5		2.9		
50	58	16	10		16		16	6		3.4	+0.15	
58	65	18	11		18		18	7	+0.15	3.3		
65	75	20	12		20	0.60	20	7.5		3.8		0.60
75	85	22	14		22		22	8.5		4.8		
85	95	25	14		22		25	9		4.3		
95	110	28	16		25		28	10		5.3		
110	130	32	18		28		32	11		6.2		
130	150	36	20	+0.3	32	1.00	36	12		7.2		1.00
150	170	40	22		36		40	13		8.2		
170	200	45	25		40		45	15		9.2		
200	230	50	28		45		50	17		10.1	+0.3	
230	260	56	32		50	1.60	56	19		11.1		1.60
260	290	63	32		56		63	20		12.1		
290	330	70	36		63		70	22		13.1		
330	380	80	40		70	2.50	80	25		14.1		2.50
380	440	90	45		75		90	28		16.1		
440	500	100	50		80		100	31		18.1		

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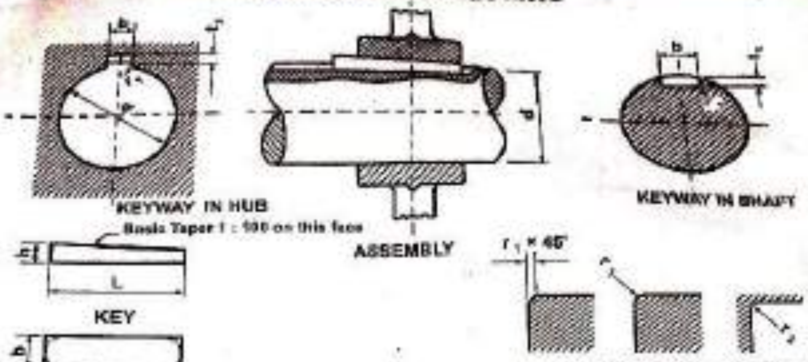
mm	4	5	6	8	10	12	14	16	18	20	22	25	28	32	36	40	45	50
Height (h) mm	4	5	6	7	8	8	9	10	11	12	14	14	16	18	20	22	25	28
Preferred length (L), mm	14	14	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	16	16	16	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	18	18	18	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	20	20	20	20	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	22	22	22	22	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	25	25	25	25	25	-	-	-	-	-	-	-	-	-	-	-	-	-
	28	28	28	28	28	-	-	-	-	-	-	-	-	-	-	-	-	-
	32	32	32	32	32	32	-	-	-	-	-	-	-	-	-	-	-	-
	36	36	36	36	36	36	-	-	-	-	-	-	-	-	-	-	-	-
	40	40	40	40	40	40	40	-	-	-	-	-	-	-	-	-	-	-
	45	45	45	45	45	45	45	45	-	-	-	-	-	-	-	-	-	-
	-	50	50	50	50	50	50	50	50	-	-	-	-	-	-	-	-	-
	-	56	56	56	56	56	56	56	56	56	-	-	-	-	-	-	-	-
	-	-	63	63	63	63	63	63	63	63	63	-	-	-	-	-	-	-
	-	-	71	71	71	71	71	71	71	71	71	71	-	-	-	-	-	-
	-	-	-	80	80	80	80	80	80	80	80	80	80	-	-	-	-	-
	-	-	-	90	90	90	90	90	90	90	90	90	90	90	-	-	-	-
	-	-	-	-	100	100	100	100	100	100	100	100	100	100	100	-	-	-
	-	-	-	-	110	110	110	110	110	110	110	110	110	110	110	110	-	-
	-	-	-	-	-	125	125	125	125	125	125	125	125	125	125	125	125	-
	-	-	-	-	-	140	140	140	140	140	140	140	140	140	140	140	140	140
	-	-	-	-	-	-	160	160	160	160	160	160	160	160	160	160	160	160
	-	-	-	-	-	-	-	180	180	180	180	180	180	180	180	180	180	180
	-	-	-	-	-	-	-	-	200	200	200	200	200	200	200	200	200	200
	-	-	-	-	-	-	-	-	-	220	220	220	220	220	220	220	220	220
	-	-	-	-	-	-	-	-	-	-	250	250	250	250	250	250	250	250
	-	-	-	-	-	-	-	-	-	-	-	280	280	280	280	280	280	280
	-	-	-	-	-	-	-	-	-	-	-	-	315	315	315	315	315	315
	-	-	-	-	-	-	-	-	-	-	-	-	-	355	355	355	355	355
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	400	400	400	400
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

TAPER KEYS AND KEYWAYS



All Dimensions in millimetres

SHAFT DIAMETER, d		KEY				KEYWAY IN SHAFT AND HUB					LENGTH	
Above	Upto and including	Cross Section Width, b (h9)	Cross Section Height h	Chamfer or radius r_1 (Min)	Keyway width b (D10)	Depth in shaft, t_1	Tolerance on t_1	Depth in Hub, t_2	Tolerance on t_2	Radius, r_2 (Max)	Minimum	Maximum
6	8	2	2	0.16	2	1.2		0.5			6	20
8	10	3	3		3	1.8	+0.05	0.9		0.16	8	36
10	12	4	4		4	2.5		1.2			10	45
12	17	5	5	0.25	5	3.0		1.7	+0.1	0.25	12	56
17	22	6	6		6	3.5		2.1			16	70
22	30	8	7		8	4.0	+0.1	2.5			20	90
30	38	10	8		10	5.0		2.5			25	110
38	44	12	8		12	5.0		2.5			32	140
44	50	14	9	0.40	14	5.5		2.9		0.40	40	160
50	58	16	10		16	6.0		3.4			45	180
58	65	18	11		18	7.0		3.3			50	200
65	75	20	12		20	7.5		3.8			56	220
75	85	22	14		22	8.5		4.8			63	250
85	95	25	14	0.60	25	9.0		4.3	+0.2	0.60	70	280
95	110	28	16		28	10.0		5.3			80	315
110	130	32	18		32	11.0		6.2			90	355
130	150	36	20		36	12.0		7.2			100	400
150	170	40	22		40	13.0		8.2		1.00	110	400
170	200	45	25	1.00	45	15.0	+0.15	9.2			125	400
200	230	50	28		50	17.0		10.1			140	400
230	260	56	32		56	19.0		12.1				
260	290	63	32	1.6	63	20.0		11.1		1.6		
290	330	70	36		70	22.0		13.1	+0.3			
330	380	80	40		80	25.0		14.1				
380	440	90	45	2.5	90	28.0		16.1		2.5		
440	500	100	50		100	31.0		18.1				

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PREFERRED LENGTHS OF TAPER KEYS

Width (b) mm	2	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32	36	40	45	50
Height (h) mm	2	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18	20	22	25	28
Preferred length (L), mm	6	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	8	8	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	10	10	10	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	12	12	12	12	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	14	14	14	14	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	16	16	16	16	16	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	18	18	18	18	18	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	20	20	20	20	20	20	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	22	22	22	22	22	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	25	25	25	25	25	25	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	28	28	28	28	28	28	-	-	-	-	-	-	-	-	-	-	-	-	-
	-	32	32	32	32	32	32	32	-	-	-	-	-	-	-	-	-	-	-	-
	-	36	36	36	36	36	36	36	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	40	40	40	40	40	40	40	-	-	-	-	-	-	-	-	-	-	-
	-	-	45	45	45	45	45	45	45	45	-	-	-	-	-	-	-	-	-	-
	-	-	-	50	50	50	50	50	50	50	50	-	-	-	-	-	-	-	-	-
	-	-	-	56	56	56	56	56	56	56	56	56	-	-	-	-	-	-	-	-
	-	-	-	-	63	63	63	63	63	63	63	63	63	-	-	-	-	-	-	-
	-	-	-	-	70	70	70	70	70	70	70	70	70	70	-	-	-	-	-	-
	-	-	-	-	-	80	80	80	80	80	80	80	80	80	80	-	-	-	-	-
	-	-	-	-	-	90	90	90	90	90	90	90	90	90	90	90	-	-	-	-
	-	-	-	-	-	-	100	100	100	100	100	100	100	100	100	100	100	-	-	-
	-	-	-	-	-	-	110	110	110	110	110	110	110	110	110	110	110	110	-	-
	-	-	-	-	-	-	-	125	125	125	125	125	125	125	125	125	125	125	125	-
	-	-	-	-	-	-	-	140	140	140	140	140	140	140	140	140	140	140	140	140
	-	-	-	-	-	-	-	-	160	160	160	160	160	160	160	160	160	160	160	160
	-	-	-	-	-	-	-	-	-	180	180	180	180	180	180	180	180	180	180	180
	-	-	-	-	-	-	-	-	-	-	200	200	200	200	200	200	200	200	200	200
	-	-	-	-	-	-	-	-	-	-	-	220	220	220	220	220	220	220	220	220
	-	-	-	-	-	-	-	-	-	-	-	-	250	250	250	250	250	250	250	250
	-	-	-	-	-	-	-	-	-	-	-	-	-	280	280	280	280	280	280	280
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	315	315	315	315	315	315
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	355	355	355	355	355
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	400	400	400	400
	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

Fits and Tolerance

Its a manufacturing consideration for the sizes of the product

Its relation between the dimensions of the mating parts

Fit ?

The degree of tightness or looseness between the mating parts known as Fits

Three categories:

1.clearance,
2.location or transition, and
3. interference.

General example t

Hole and shaft assembly arrangement

1. Clearance Fit

Hole size > Shaft Size, there is clearance between the shaft and hole.

Examples of clearance fit are door hinges, wheel, and axle, shaft and bearing

2. Interference fit

Hole size < shaft size, then the assembly of parts made by means of forcing the shaft

Example **: cotter pins in sleeve and cotter joint, Keys in couling**,

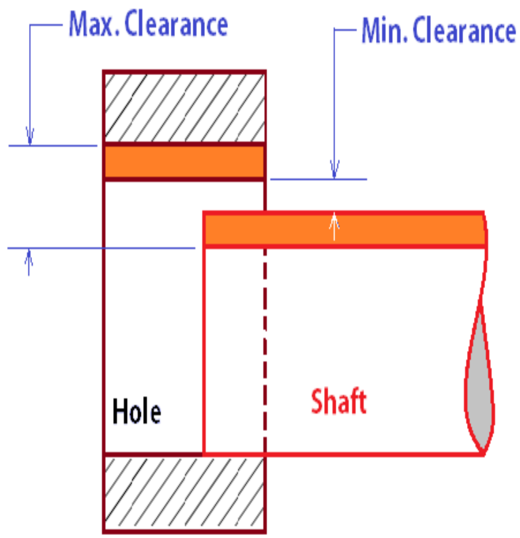
3. Transition Fit

Almost hole size and shaft size are closer or equal or there sizes overlap

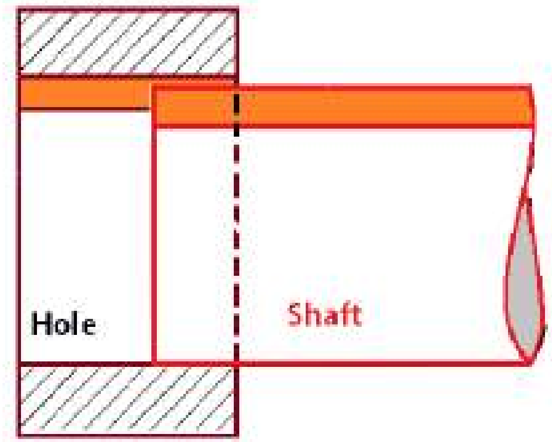
It is either clearance fit or interference fit

Example : coupling rings, Spigot mating

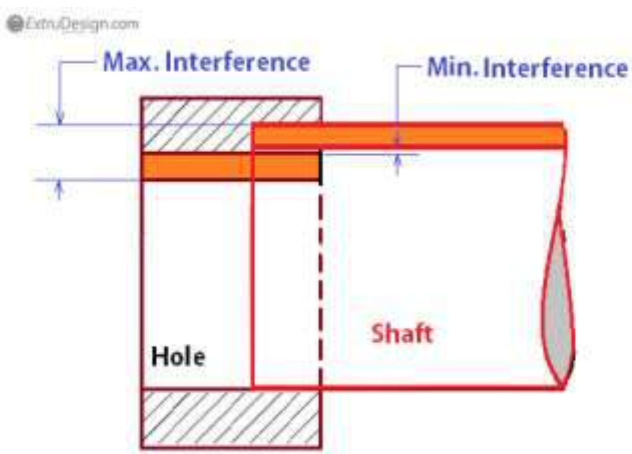
Clearance fit



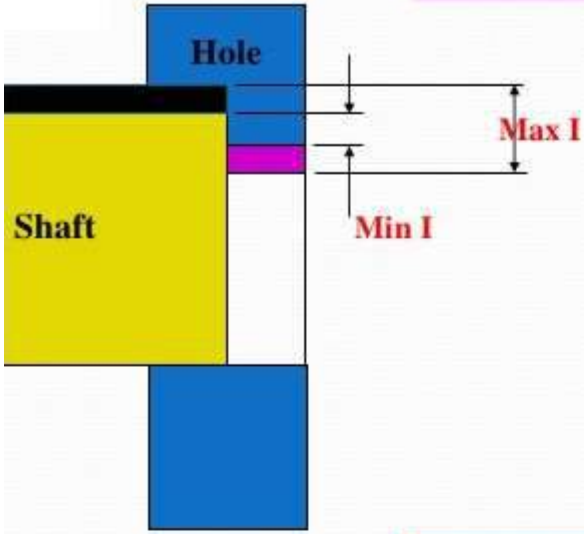
Transition fit



Interference fit



Interference Fits



Tolerance zones never meet but crosses each other

Max. I = LL of hole - UL of shaft
Min. I = UL of hole - LL of shaft

The interference fits may be shrink fit, heavy drive fit and light drive fit.

Some basics

Nominal size= size provided in the drawing sheet

Basic size= production size, which might be nominal size

Example:

Dia 50 mm of shaft is required

Nominal or Basic size 50 mm

During machining , 50 will not be attained, the size may be lower or higher than the requirement, due to various errors. Eg. Machine error, operator error

50. 05 mm- upper size or
49.95mm – lower size
Higher is upper limit and lower is the lower limit
Representation= $50^{\pm 0.05}$ **mm** **Limits: UL, LL**

Tolerance values

$50^{\pm 0.05}$

+0.05 mm is uL/uD--- denoted by **ES/es**

-0.05mm is lL/lD----- denoted by **EF/ef**

What is the tolerance?

The difference between upper and lower limits of dimesions

ES- EI= Tolerance

es- ei = tolerance

Types

1. **Unilateral---- variation of basic size in one direction either + or -**
2. **Bilateral------- variation of basic size in both direction + & -**

Limits systems

1. HOLE basis system
2. Shaft basis System

1. HOLE basis system: hole size kept as constant, shaft size will allow to vary
Denoted by Capital letters like H, G, L,,, etc
Always lower devation is zero(EI=0)

2. Shaft basis System = Shaft size kept as constant and hole size allowed to vary
Denoted by small letters like h,g,i,,, etc
Always upper devation is zero(es=0)

DATA BOOK usage; 3.1. 3.3 to 3.17 pages

3.3 to 3.10 more important for the calculation

Problems

Hole size: 25. 00 and 25.02 mm, shaft size : 24.97 andd 24.95mm

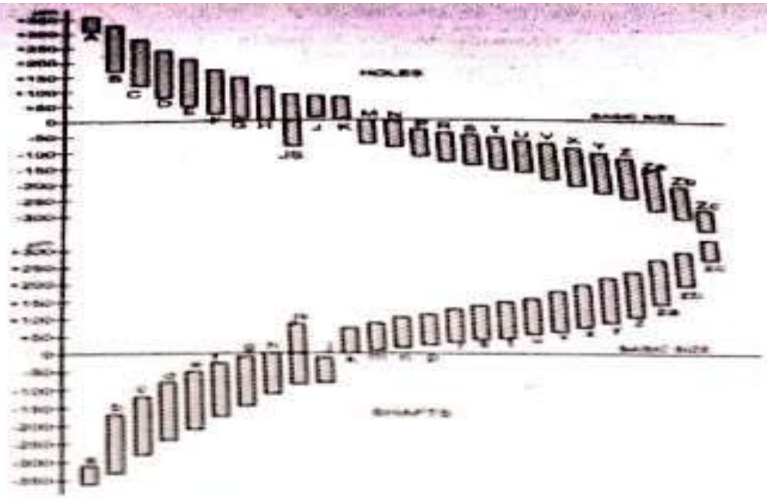
Tolerance calculation:

Hole: Upper limit- Lower limit of hole
=25.02-25.00
= 0.02 mm

Shaft: upper limit – lower limit
= 24.97-24.95
= 0.02 mm

Allowance: lower limit hole – upper limit shaft
=25.00-24.97
= 0.03 mm

LETTER SYMBOLS FOR TOLERANCES

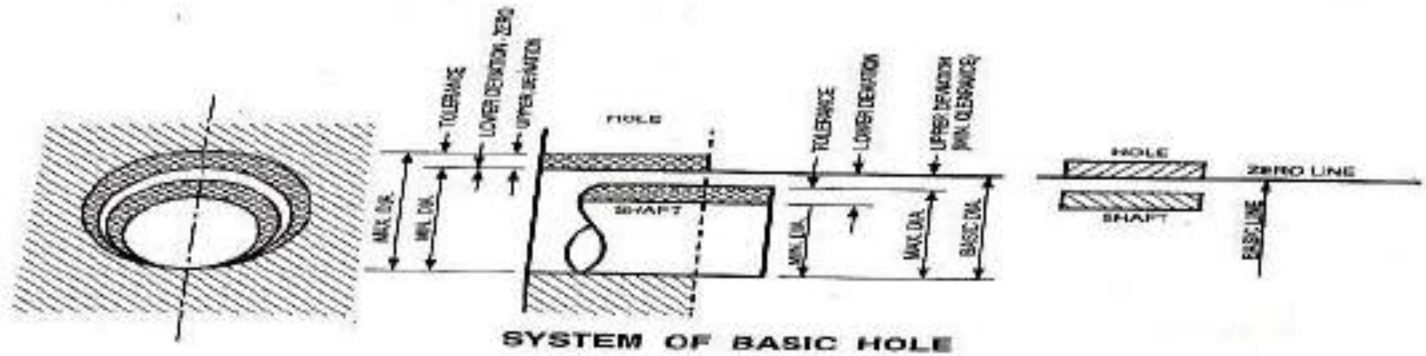


FUNDAMENTAL TOLERANCES OF GRADES 01, 0 AND 1 TO 16

Values of tolerances in Microns — 1 micron = 0.001 mm

Diameter Steps in mm	Tolerance Grades																	
	01	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14*	15*	16*
To and inc 3	0.3	0.5	0.8	1.2	2	3	4	6	10	14	25	40	60	100	140	250	400	600
Over 3 To and inc 6	0.4	0.6	1	1.5	2.5	4	5	8	12	18	30	48	75	120	180	300	480	750
Over 6 To and inc 10	0.4	0.6	1	1.5	2.5	4	6	9	15	22	36	58	90	150	220	360	580	900
Over 10 To and inc 18	0.5	0.8	1.2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100
Over 18 To and inc 30	0.6	1	1.5	2.5	4	6	9	13	21	33	52	84	130	210	330	520	840	1300
Over 30 To and inc 50	0.6	1	1.5	2.5	4	7	11	16	25	39	62	100	160	250	390	620	1000	1600
Over 50 To and inc 80	0.8	1.2	2	3	5	8	13	19	30	46	74	120	190	300	460	740	1200	1900
Over 80 To and inc 120	1	1.5	2.5	4	6	10	15	22	35	54	87	140	220	350	540	870	1400	2200
Over 120 To and inc 180	1.2	2	3.5	5	8	12	18	25	40	63	100	160	250	400	630	1000	1600	2500
Over 180 To and inc 250	2	3	4.5	7	10	14	20	29	46	72	115	185	290	460	720	1150	1850	2900
Over 250 To and inc 315	2.5	4	6	8	12	16	23	32	52	81	130	210	320	520	810	1300	2100	3200
Over 315 To and inc 400	3	5	7	9	13	18	25	36	57	89	140	230	360	570	890	1400	2300	3600
Over 400 To and inc 500	4	6	8	10	15	20	27	40	63	97	155	250	400	630	970	1550	2500	4000

*Up to 1 mm, Grades 14 to 16 are not provided



SYSTEM OF BASIC HOLE

RUNNING AND SLIDING FITS

Combination of Hole and shaft	Quality of fit	Typical uses
H 6 g 5 Fine H 7 g 6 Normal H 8 g 7 Coarse	Precision	Small clearance - used in precision equipment under very light load - Bearings for accurate link work and for piston and slide valves - *Also used for spigot or location fits.*
H 6 f 6 Fine H 7 f 7 Normal H 8 f 8 Coarse	Close running	Widely used as grease or oil lubricated bearings having low temperature differences - bearings for gear shafts, small electric motor shafts and pump shafts.
H 7 e 7 Fine H 8 e 8 Normal H 9 e 9 Coarse	Normal running	Used for properly lubricated bearings with appreciable clearance. Finer grades for high speeds and heavy loads. Turbo generator and large electric motor bearings.
H 8 d 8 Fine H 8 d 9 Normal H 9 d 9 Coarse	Loose running	For plamber block bearings and loose pulleys
H 8 c 8, H 8 b 8, H 9 a 9 } Fine H 9 c 9, H 11 c 11 } Normal H 11 c 9, H 11 b 9, H 11 a 9 } Coarse	Slack running or positional fit	Large clearance - not widely used.

Over / To		3	3 / 6	6 / 10	10 / 18	18 / 24	24 / 30	30 / 40	40 / 50	50 / 65	65 / 80	80 / 100	100 / 120	120 / 140	140 / 160	160 / 180	180 / 200	200 / 225	225 / 250	250 / 280	280 / 315	315 / 355	355 / 400	400 / 450	450 / 500
		Diameter Steps, mm																							
a9	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
b9	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
c8	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
c9	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
c11	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
d8	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
d9	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
d10	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
e6	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
e7	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
e8	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
e9	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
f6	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
f7	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
f8	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
g4	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
g5	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
g6	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
g7	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h5	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h6	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h7	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h8	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h9	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h10	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	
h11	G / N	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]		[illegible]			[illegible]			[illegible]		[illegible]		[illegible]	

Note : G - Go; N - No go; Tolerances in Microns   1 Micron = 0.001 mm = 1×10^{-6} m

TABLE OF TOLERANCES

Diameter Steps, mm

	Over To	3	3 6	6 10	10 18	18 24	24 30	30 40	40 50	50 65	65 80	80 100	100 120	120 140	140 160	160 180	180 200	200 225	225 250	250 280	280 315	315 355	355 400	400 450	450 500
A 9	N	295	300	316	333	352		372	382	414	434	467	497	560	620	680	775	855	935	1050	1180	1340	1490	1655	1805
	G	270	270	280	290	300		310	320	340	360	380	410	460	520	580	660	740	820	920	1050	1200	1350	1500	1650
B 9	N	165	170	186	193	212		232	242	264	274	307	327	360	380	410	455	495	535	610	670	740	820	915	995
	G	140	140	150	150	160		170	180	190	200	220	240	260	280	310	340	380	420	480	540	600	680	760	840
C 9	N	85	100	116	138	162		182	192	214	224	257	267	300	310	330	355	375	395	430	460	500	540	595	635
	G	60	70	80	95	110		120	130	140	150	170	180	200	210	230	240	260	280	300	330	360	400	440	480
E 8	N	28	38	47	59	73		89		106		126		148			172			191		214		232	
	G	14	20	25	32	40		50		60		72		85			100			110		125		135	
F 7	N	16	22	28	34	41		50		60		71		83			96			108		119		131	
	G	6	10	13	16	20		25		30		36		43			50			56		62		68	
G 7	N	12	16	20	24	28		34		40		47		54			61			69		75		83	
	G	2	4	5	6	7		9		10		12		14			15			17		18		20	
H 5	N	4	5	6	8	9		11		13		15		18			20			23		25		27	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
H 6	N	6	8	9	11	13		16		19		22		25			29			32		36		40	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
H 7	N	10	12	15	18	21		25		30		35		40			46			52		57		63	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
H 8	N	14	18	22	27	33		39		46		54		63			72			81		89		97	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
H 9	N	25	30	36	43	52		62		74		87		100			115			130		140		155	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
H10	N	40	48	58	70	84		100		120		140		160			185			210		230		250	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
H 11	N	60	75	90	110	130		160		190		220		250			290			320		360		400	
	G	0	0	0	0	0		0		0		0		0			0			0		0		0	
J 7	N	4	6	8	10	12		14		18		22		26			30			36		39		43	
	G	-6	-6	-7	-8	-9		-11		-12		-13		-14			-16			-16		-18		-20	
K 6	N	0	2	2	2	2		3		4		4		4			5			5		7		8	
	G	-6	-6	-7	-9	-11		-13		-15		-18		-21			-24			-27		-29		-32	
K 7	N	0	3	5	6	6		7		9		10		12			13			16		17		18	
	G	-10	-9	-10	-12	-15		-18		-21		-25		-28			-33			-36		-40		-45	
M 7	N	-2	0	0	0	0		0		0		0		0			0			0		0		0	
	G	-12	-12	-15	-18	-21		-25		-30		-35		-40			-46			-52		-57		-63	
N 7	N	-4	-4	-4	-5	-7		-8		-9		-10		-12			-14			-14		-16		-17	
	G	-14	-16	-19	-23	-28		-33		-39		-45		-52			-60			-66		-73		-80	
P 7	N	-6	-8	-9	-11	-14		-17		-21		-24		-28			-33			-36		-41		-45	
	G	-16	-20	-24	-29	-35		-42		-51		-59		-68			-79			-88		-98		-108	
S 6	N	-14	-16	-20	-25	-31		-38		-47	-53	-64	-72	-85	-93	-101	-113	-121	-131	-149	-161	-179	-197	-219	-239
	G	-20	-24	-29	-36	-44		-54		-66	-72	-86	-94	-110	-118	-126	-142	-150	-160	-181	-193	-215	-233	-259	-279
S 7	N	-14	-15	-17	-21	-27		-34		-42	-48	-58	-66	-77	-85	-93	-105	-113	-123	-138	-150	-169	-187	-209	-229
	G	-24	-27	-32	-39	-48		-59		-72	-78	-93	-101	-117	-125	-133	-151	-159	-169	-190	-202	-226	-244	-272	-292

Note : G - Go; N - No go; Tolerances in Microns   1 Micron = 0.001 mm = 1×10^{-6} m

Designation : Number- (hole)capital letter .IT gradeNo./(shaft)small letter. It grade No

↓

Diameter

Shaft designation : 40 H8/f7

Geometric mean diameter= Sqrt(D_1 x D_2)

D_1 and D_2 are shaft range

$D_1 < D_{given} < D_2$

Tolerance calculation

$$i = 0.45 \sqrt[3]{D} + 0.0001 D_{GM}$$

pg.3.6/ddb.

UNIT III

- ME18503-Design Of Machine Elements

UNIT III DESIGN FOR SPRINGS

Design of Close coil helical springs under varying load condition. Design of Leaf spring, Disc Spring and Torsion spring

OBJECTIVE

This course will familiarize the design principles of springs under dynamic and static conditions

COURSE OUT COME 3 - CO3

1.Examining the close coil helical springs under variable loading .
2.Acquiring the knowledge of leaf, disc and torsion springs.

What is spring?

a steel wire wound around an imaginary cylinder- Helical springs
a steel wire wound around an imaginary cone - conical spring

Elastic member absorbs energy when it is loaded and releases the energy when it is unloaded

It will distort when loaded and recover its original shape when unloaded

General - coil spring

Springs classifications

1. Open coil and closed coil
   helix angle > 10^o → helix angle < 10^o

Clo

1. Helical springs
   compression and tension

2. Conical & volute

3.Torsion springs & spiral springs- bi-cycle hand lever brake, writng pads, toys, clocks

4. Leaf or laminated springs; pre stressed plates of different lengths held together by means of central bolt and clamps (U- clamps)

5.Disc or Belleville springs.- conical discs held together by central bolt
Needed: high spring rate in compact unit.

6. Special purpose springs
rubber , air or liquid, ring springs

Spring diagrams

Helical Compression Spring

Helical Extension Spring

Conical Spring

Torsion Spring

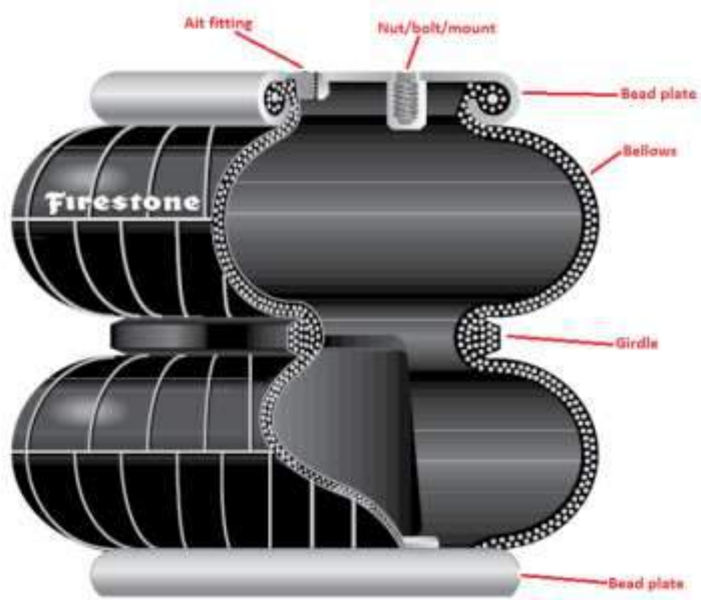
Laminated or Leaf Spring

Disc or Belleville Spring

Spiral spring

Disc springs--- industrial uses, brakes, clutches piping- shock mounting

Rubber spring



Air spring

Uses/functions/applications

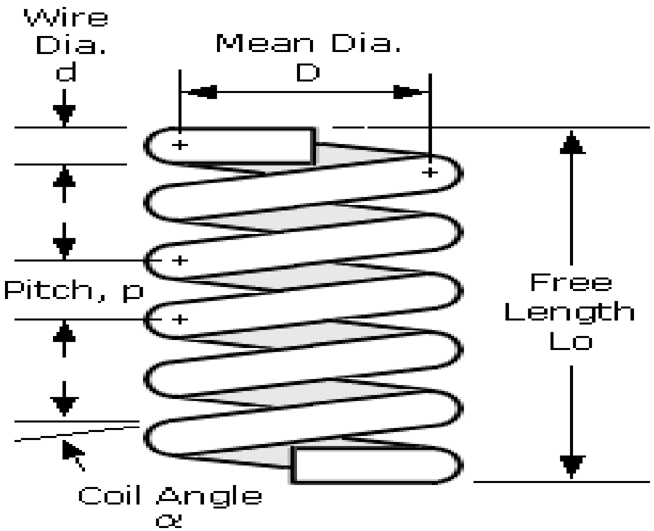
To absorb the shock or vibration as in-car springs, railway buffers, etc.

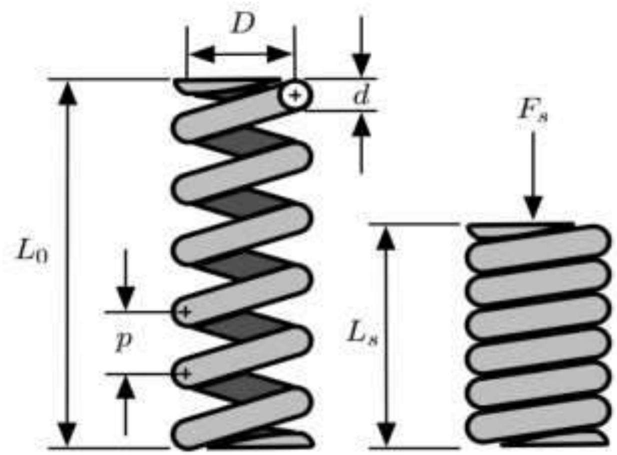
To measure the forces in a spring balance.

Apply forces in brakes and clutches to stop the vehicles.

Spring is also used to store the **energy** as in clocks, toys, etc.

Spring Terminologies





Spring stiffness(k): Load required to produce Unit deflection

$k = w/\delta$ N/mm, w= load , δ= deflection

Spring Index (C): C= Dm/d - No unit

Where Dm= mean diameter of coil

d = wire diameter

Note: the clearance between the two adjacent turn is 1 mm always.

Pitch of the coil

$$p = \frac{\text{Free length}}{n' - 1}$$

$$p = \frac{L_F - L_S}{n'} + d$$

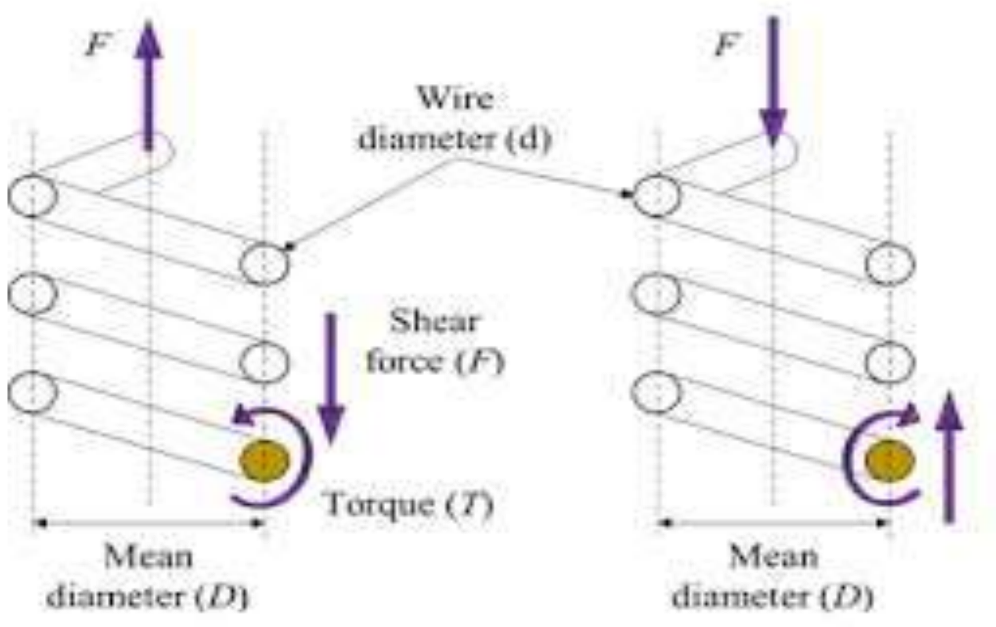
where LF = Free length of the spring.

LS = Solid length of the spring,

n' = Total number of coils, and

d = Diameter of the wire.

Stresses



1. Shear stress by external load
2. Torsion at wire curvature- T= F x R

Radius of curvature is neglected in static loading

to consider this radius of curvature, **Wahl's factor used. K**

DESIGN PROCEDURE spring compression spring

Step1: Find d, Dm, Do, & Di for the spring using Shear stress(pg.7.100/DDB)

Step2: Find the deflection of spring ‘ y ‘ (Pg. 7.100/DDB) (either y or n based on The available either y or n

Step3: Find the stiffness of the spring ‘q ‘ (Pg. 7.100/DDB)

Step4: Find Lf- free length of coil

Lf = Ls + y + 15% of y

Ls= nxd n= no. of turns, d- wire diameter.

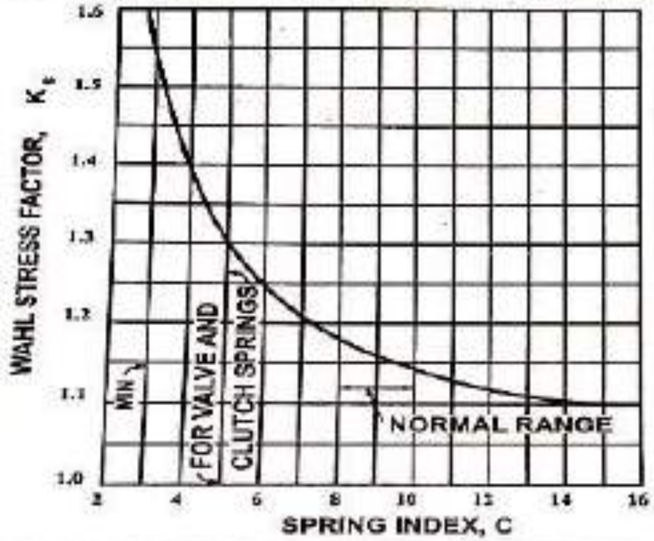
Step 5: Find pitch ‘p ‘

Step 6 ; check for buckling Pg.7.101/DDB

Step 7: Check for surging (optional step) pg.7.101/DDB

Helical Springs

Circular Section	Rectangular Section
$\tau = K_s \frac{8PD}{\pi d^3} = K_s \frac{8WC}{\pi d^2}$	$\tau = \frac{Q_2 PD}{2tb^2}$
$y = \frac{8PD^3 n}{Gd^4} = \frac{8PC^3 n}{Gd}$	$y = \frac{Q_1 \pi P D^3 n}{4Gt^3 b}$
$q = \frac{Gd^4}{8D^3 n} = \frac{Gd}{8C^3 n}$	$q = \frac{4Gt^3 b}{Q_1 \pi D^3 n}$

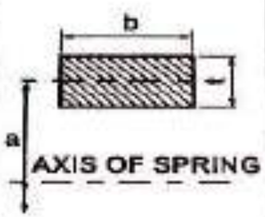


$U = \frac{Py}{2}$

Wahl stress factor

$K_s = \frac{4C-1}{4C-4} + \frac{0.615}{C}$

(may be obtained from the graph also)



Factors for rectangular wire section

b / t	1	1.5	2	3	4	6	8	10	α
Q_1	7.09	5.10	4.36	3.80	3.56	3.36	3.26	3.21	3
Q_2	4.79	4.35	4.05	3.71	3.52	3.35	3.25	3.2	3

P	axial load, kgf
D	mean diameter of spring, cm
d	diameter of wire, cm
C	spring index, D / d
b	breadth of wire, cm
t	thickness of wire, cm
n	number of active coils
τ	shear stress, kgf / cm^2
G	modulus of rigidity, kgf / cm^2
y	deflection of spring, cm
q	spring rate or stiffness, kgf / cm
K_s	Wahl stress factor
Q_1, Q_2	factors for springs of rectangular section
U	resilience, kgf cm
f	lowest natural frequency for circular coil helical springs, cycles per second
γ	specific weight of spring material, kgf / cm^3
g	gravitational constant, 981 cm / s^2
L_f	free length of spring, cm
L_s	solid length, cm
p	pitch of coils, cm
α	helix angle, $< 12°$
λ	solid deflection, cm
Δ	clearance between concentric springs

lowest natural frequency

$$f = \frac{(q/m)^{1/2}}{2} \quad \frac{d}{\pi D^2 n}\sqrt{\frac{G\,g}{8\,\gamma}}$$

Check for surging

$$f \geq 12\, f_l$$

Check for solid stress

$\tau_s < 0.5\,\sigma_u$ for hard drawn carbon steels

$< 0.5\,\sigma_u$ for alloy steels

To avoid Buckling

$$\frac{L_f}{D} < 3$$

for $\frac{L_f}{D} > 3$ the spring must be suitably guided

Coaxial springs

(suffixes 1, 2 refer to outer and inner springs respectively)

$$\tau_1 = \tau_2 < [\tau]$$

$$\frac{D_1}{d_1} = \frac{D_2}{d_2} = C$$

$$\frac{P_1}{P_2} = \left(\frac{C}{C-2}\right)^2$$

$$\Delta = \frac{(D_1 - D_2) - (d_1 + d_2)}{2} \approx \frac{d_1 - d_2}{2}$$

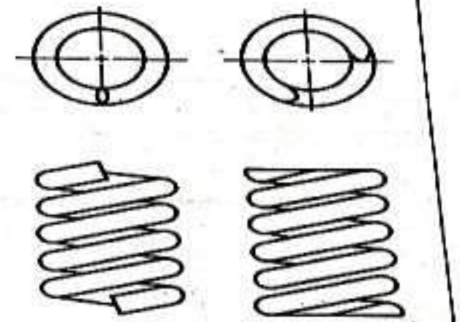
$$d_1 \leq \frac{D_1 - D_2}{2}$$

The winding of the springs should be of opposite hands

m mass of the active coil in the spring

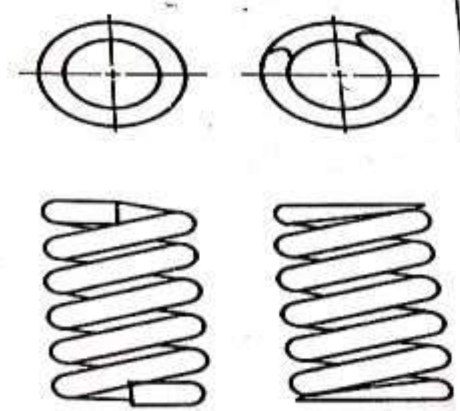
f_l frequency of the applied load

τ_s shear stress in spring when compressed solid



Plain Ends

Plain and Ground Ends



Squared Ends

Squared and Ground Ends

End conditions and length of springs

Type of end	Total coils	Free length, L_f	Solid length, L_s
Plain	n	p n + d	d n + d
Plain and Ground	n	p n	d n
Squared	n + 2	p n + 3 d	d n + 3 d
Squared and Ground	n + 2	P n + 2 d	d n + 2 d

S1 Design a helical spring subjected to a load of 1000 N for a deflection of 25 mm and the spring index is 5. The allowable shear stress is 420Mpa. Take modulus of Rigidity as 84 kN/mm^2.

DATA

Load W or P = 1000 N

C = D/d= 5

Y = 25 mm

[Ʈ]= 420 N/mm^2

G= 84 x 1000 N/mm^2

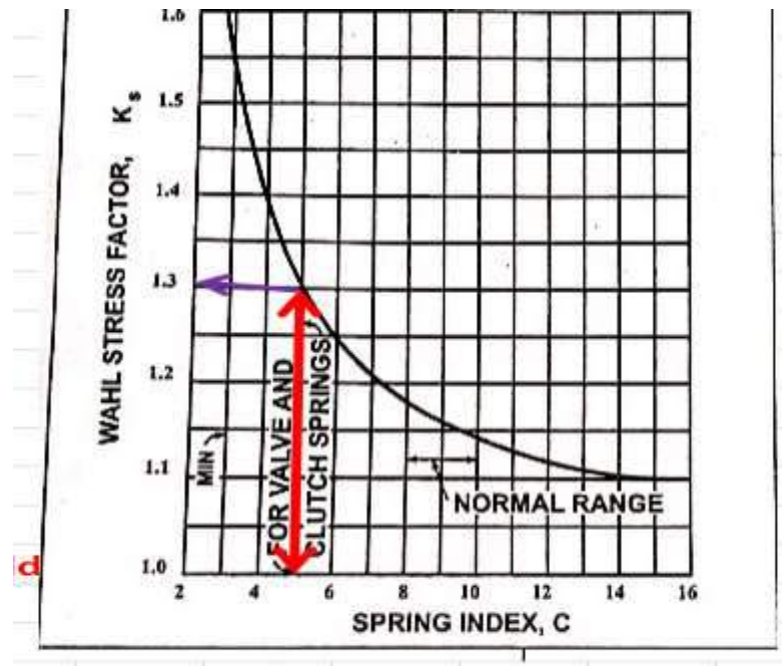
Step1: find d, Dm, etc refer 7.100/DDB

$$\tau' = K_s \frac{8PD}{\pi d^3} = K_s \frac{8WC}{\pi d^2}$$

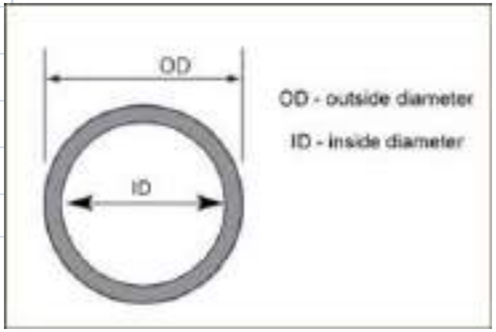
P=	1000
Pi=	3.14
[τ] =	420
c=	5

To find Ks: refer pg .7.100/dd

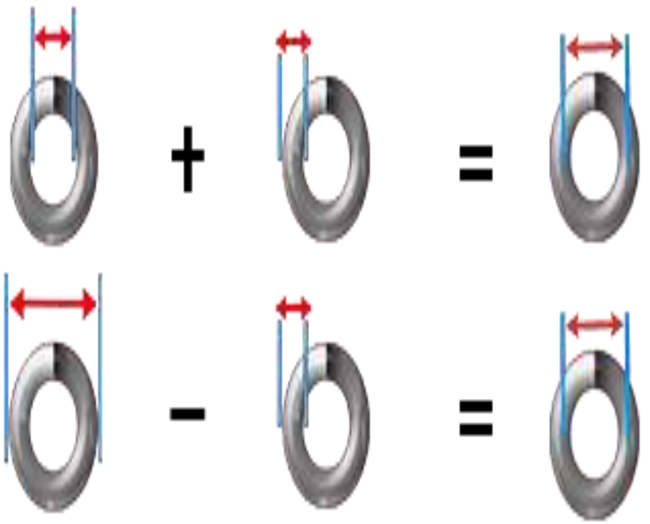
For c = ks= 1.3



LHS =	420	
RHS =	[1.3 x (8 x 1000 x 5)]/(3 .141 x d^2)	
	16555.24	/d^2
LHS	=	RHS
420	=	16555.2 /d^2
d^2 =		39.4172
d =		6.28



Mean D=?			
C= D/d			
D=	31.39	mm	
Inner dia=? Di			
Di =	Dm-2r		
	Dm-d	[2r=d]	
Di =	25.11	mm	
Outter diameter Do?			
Do=	Dm+2r		
	Dm+ d		
Do=	37.67	mm	



Step2:

Find the deflection of spring ' y ' (Pg. 7.100/DDB) (either y or n based on

The available either y or n

$$y = \frac{8PD^3n}{Gd^4} = \frac{8PC^3n}{Gd}$$

LHS =	25				
RHS =	8x1000 x31.39xn/(84x10^3 x6.28)				
			Nr	247473770	n
			Dr	130512323	
25	1.8961717	n		1.8961717	n
n=	13.18446				
n=	13	turns			

Step3 : Finding stiffness of the spring ' q '

ref. Pg.7.100/DDB

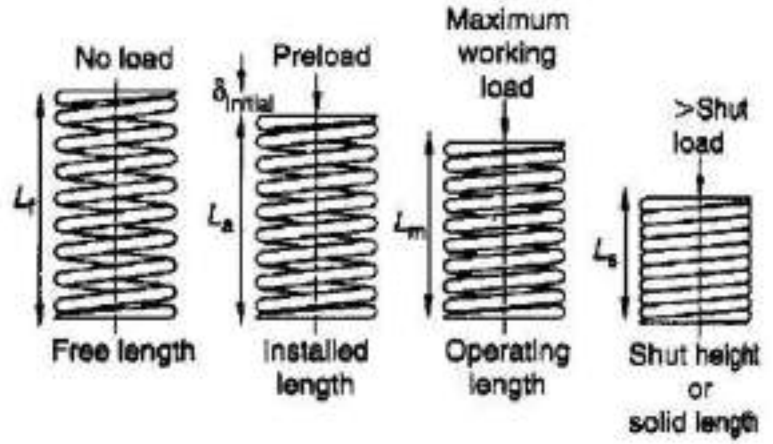
$$q = \frac{G d^4}{8 D^3 n} = \frac{G d}{8 C^3 n}$$

D=	31.39	mm
D=	6.28	mm
n=	13	nos
G=	84000	N/mm^2
q =	[984 x 10^3)x (6.28)^4] /[8 x (31.39)^3 x	
q=	40.56757	N/mm

Step4: Find Lf- free length of coil

Lf = Ls + y + 15% of y

Ls= nxd   n= no. of turns,   d- wire diameter



Ls= n x d
= 13x 6.28
= 81.64 mm

Y= 25 mm
15 % of Y= 0.15 x 25
= 3.75 mm

Lf = 81.64 + 25 + 3.75
=110.39 mm

STEP 4: Calculation of Lf - free length of the coil.

Refer Pg.7.101/DDB

applying End conditions

Already Assume d

Plain End

ls= (dxn)+d **solid length**

Lf = Ls + y + 15% of y

n=	13		
d=	6.28		
LS=	81.64	mm	[Ls = 13 x 6.28]
y=	25	mm	
15% of Y =	3.75	mm	

Now		
Lf=	81.64 +25+ 3.75	
Lf =	**110.39**	**mm**

STEP 5: Find pitch 'p '

$$p = \frac{\text{Free length}}{n' - 1}$$

Lf=	110.39
n=	13
p=	110.39/(13-1)
P=	**9.199 mm**

STEP 6 Check for Buckling		
REFER pg. 7.101/DDB		
	LF/D> 3	
	Lf=	110.39
	D=	31.39
	Therfore	
	Lf/D =	3.516549
	Lf/D > 3,	
	"Guidance support is must for the spring"	

Lf/d < 3 = no buckling

This shows buckling is there

Step 7	Check for solid stress		
	Material selected: steel wire unalloyded cold drawn		
	for d= 7.00 mm		**(Pg.7.105/DDB)**
	σu= kgf/mm^2		
	σu =111	**1110**	
	allowable shear stress is =[420]		
		420< 0.5 x σu	
	420 <	**555**	**safe design/ material selection is correct.**

S2: design and draw a valve spring of a petrol engine for the following specifications

Spring load when the valve is open=400 N
Spring load when the valve is closed = 250 N
Maximum inside diameter of the spring= 25 mm
Length of the spring when the valve is open=40 mm
Length of the spring when the valve is closed = 50 mm
Maximum permissible stress = 400 Mpa.

DATA

Basic needs C, y, Load

C= ?
Y= ?
Dm=?

But " Di " is given in the

Y = can be calculated by comparing length of spring when valve is opened and closed

Closed length- open length
Y= 50 – 40
= 10 mm
***Load is max: 400 N for shear stress**

***Load for Y: max- min:**
400-250= 150 N

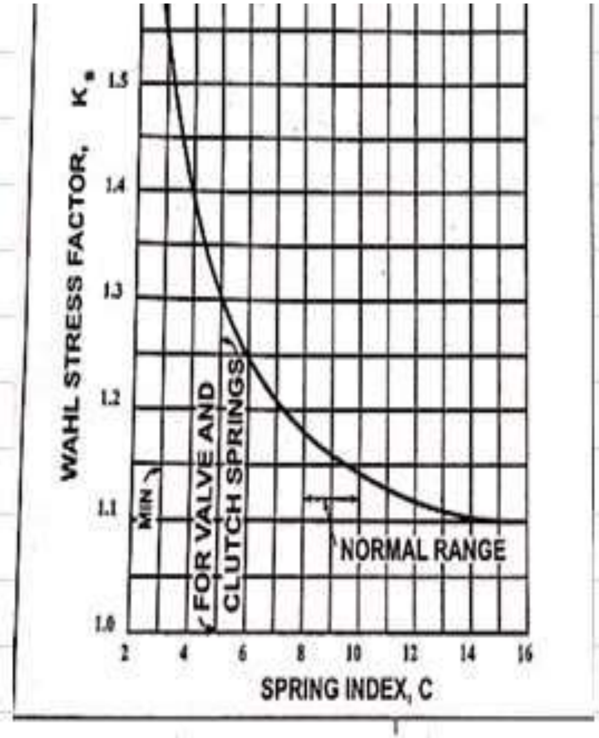
DATA				
W o=	400	N	opened	
Wc=	250	N	closed	
C = D/d	???			
Y =	???	mm		
[T] =	400	N/mm^2		
G =	84000	N/mm^2		
lenght of spring (opened)Lo			40	mm
length of closed coil Lc			50	mm
Inside dia. Of coil (Di)			25	mm

1. y =	Lc-Lo			
	10	mm		
2. C =D/d	(to be assumed from the Ks graph)			
C=	5	(initial assumption)		
	Ks=	1.3		
3. Load				
Max=	400	(for shear stress)		
Max- Min	150	(for y equation)		

Step1 **Findin d, Dm etc**

Refer Pg.7.100 /DDB

$\tau = K_s \frac{8PD}{\pi d^3}$ = $K_s \frac{8WC}{\pi d^2}$



P=	400	
Pi=	3.141	
[T] =	400	
c=	5	(Assumed)

LHS =	400	
RHS =	[1.3 x (8 x 400 x 5)]/(3 .141 x d^2)	
=	6622.09487	/d^2
LHS	=	RHS

LHS	=	RHS	
400	=	6622.09	/d^2
d^2 =		16.5552372	
d	=	4.06881275	mm
	d=	4.5 SWG pg.7.105/ddb	

Now to find Dm

Dm= Di+d
Do=Dm+d

C= D/d=5

Corrected C= 6.556
C= 6.6

Dm= 29.5mm
D0= 34mm

PROPERTIES OF SPRING STEELS

Diameter of wire mm	Average values of Tensile strength, kgf / mm², σ_u									
	Steel wire unalloyed cold drawn				Spring steel oil hardened and tempered		Spring steels for moderately elevated temperatures			
	Gr 1	Gr 2	Gr 3	Gr 4	†SW	⊕VW	**1 S	*1 D	**2 S	*2 D
0.1	-	-	258	-	-	-	-	-	-	-
0.5	170	205	244	263	-	-	-	-	-	-
1.0	160	194	228	245	190	180	-	-	210	210
1.5	152	184	214	231	180	165	190	180	200	200
2.0	145	175	203	220	172	160	190	180	200	200
2.5	140	167	193	209	166	155	180	170	195	195
3.0	135	160	187	202	162	151	180	170	195	195
3.6	130	154	178	193	158	148	170	160	190	190
4.0	128	151	173	188	158	148	170	160	190	190
4.5	125	147	169	184	154	145	162	155	185	185
5.0	121	142	163	178	154	145	162	155	185	185
5.6	117	138	158	172	150	142	155	150	180	180
6.0	115	135	156	170	150	142	155	150	180	180
6.5	113	132	151	165	146	138	150	145	175	175
7.0	111	129	149	164	146	138	150	145	175	175
7.5	109	127	146	160	146	138	-	-	170	170
8.0	107	124	143	157	140	-	-	-	170	170
8.5	104	122	140	153	140	-	-	-	-	-
9.0	102	120	138	151	140	-	-	-	-	-
9.5	101	117	134	-	140	-	-	-	-	-
10.0	100	115	132	-	135	-	-	-	-	-
10.5	-	112	-	-	135	-	-	-	-	-
11.0	0	110	-	-	135	-	-	-	-	-
12.0	-	106	-	-	135	-	-	-	-	-
12.5	-	105	-	-	135	-	-	-	-	-
13.0	-	104	-	-	135	-	-	-	-	-

† SW intended for general purposes
⊕ VW intended for value springs subjected to high dynamic stresses
* 1D and 2D intended for springs subjected to dynamic loads
** 1S and 2S intended for springs subjected to static loads

IS : 4454 - 1967

Step2:

Find the deflection of spring ' y ' (Pg. 7.100/DDB) (either y or n based on The available either y or n

Y=	**10**		
LHS =	**10**		
	6.55		1
RHS =	**8x150 x(5)^3xn/(84x10^3 x 4.5)**		
		Nr	**150000n**
		Dr	**378000**
10	**0.8943715n**		**0.3968254n**
n=	**11.181036**		
n=	**12 turns**		

Apply end conditions: plain end (Assumed)

n= n

STEP3: Finding stiffness of the spring ' q '

ref. Pg.7.100/DDB

D=	**29.50mm**
d=	**4.50mm**
n=	**12nos**
G=	**84000N/mm^2**

q = $[(84 \times 10^3) \times (4.5)^4] / [8 \times (29.5)^3 \times 12]$

q= 13.976295N/mm

STEP 4: Calculation of Lf - free length of the coil.

Refer Pg.7.101/DDB

applying End conditions
Already Assume d
Plain End
ls= (dxn)+d   solid length

Lf = Ls + y + 15% of y

n=	12	
d=	4.5	
LS=	54mm	[Ls = 12 x 4.5]
y=	10mm	
15% of Y =	1.5mm	

Now

Lf=	65.5mm
Lf =	65.5mm

Note:

To find Lf, the Y max to be obtained = (Y /difference ofload) x max.load

Lf = 54+(10/150*400)+ 0.15*(10/150*400)

=	**84.6666667**	**0.06666667**
		26.6666667
		4

STEP 5: Find pitch 'p '

$$p = \frac{\text{Free length}}{n' - 1}$$

Lf=	65.5
n=	12
p=	65.5/(12-1)
P=	5.955mm
	6mm

STEP 6 Check for Buckling

REFER pg. 7.101/DDB

LF/D< 3

Lf=	**65.5**
D=	**29.50**
Therfore	
Lf/D =	**2.22033898**

No guidance required, buckling is zero

Surging of spring

it must be prevented or it will cause failure of spring

Time interval of applied load/force= time taken by the wave propagation to and fro betweenth
Support and load taking end.

Resonance will occur

lowest natural frequency

$$f = \frac{(q/m)^{1/2}}{2} \frac{d}{\pi D^2 n} \sqrt{\frac{G g}{8 \gamma}}$$

S3: A safety valve of 60 mm diameter is to blow off at a pressure of 1.2 N/mm^2. It is held in its seat by a close coil helical spring. The maximum lift of valve is 10 mm. Design a suitable compression spring of spring index 5 and providing an initial compression of 35 mm. The maximum shear stress in the material of the wire is limited to 500 Mpa. G=80kN/mm^2.

DATA

Valve seat= 60mm

Pressure=1.2 N/mm^2

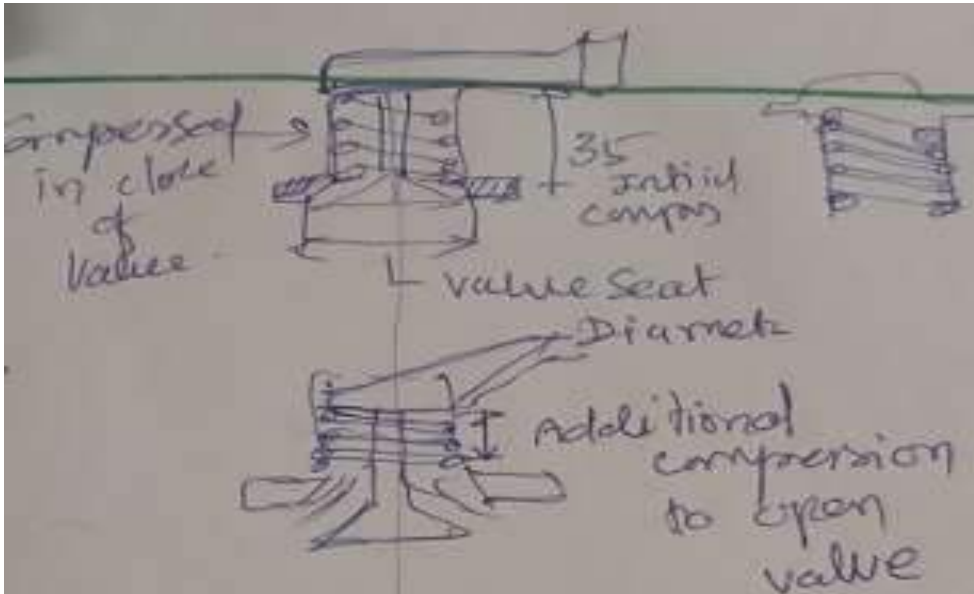
Y= 10mm

C=5

Allowable shear stress=500 N/mm^2

Initial compression = 35 mm

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



Pressure is the force or load

$$P = \frac{F}{A} \text{ or } \frac{W}{A}$$

$$\therefore W = P \cdot A \quad \left[A = \frac{\pi}{4} d_{valve}\right]$$

Y_{max} = Initial Compressi + Additional compression (to open valve)

$$Y_{Total} = 35 + 10$$

$$= 45\,mm.$$

Repeated Loading

$$\tau_a = \frac{8 k_s P_a D}{\pi d^3} = \frac{8 k_s P_a C}{\pi d^2}$$

$$\tau_m = \frac{8 k_{sh} P_m D}{\pi d^3} = \frac{8 k_{sh} P_m C}{\pi d^2}$$

$$\frac{1}{n} = \frac{\tau_m - \tau_a}{\tau_y} + \frac{2\tau_a}{\tau_e}$$

Curvature factor, k_c

C	3	4	6	7	8	9	10
k_c	1.33	1.25	1.15	1.12	1.11	1.1	1.09

Static Approach to Varying Loads

No. of cycles	Classification	*Recommended Design stress [τ]
$\geq 10^6$	Severe Service	$= 0.263\,\sigma_u$
$\geq 10^4$ but $< 10^6$	Average Service	$= 0.324\,\sigma_u$
$< 10^4$	Light Service	$= 0.405\,\sigma_u$

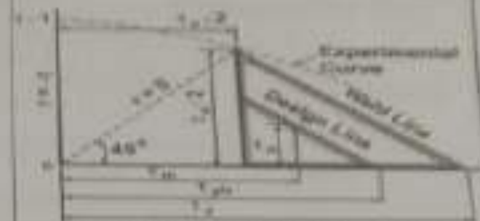
* For extension springs 0.8 times the values recommended for compression springs may be used.

EXTENSION SPRINGS

$$P_i = \frac{\pi \tau_i d^3}{8 D} = \frac{\pi \tau_i d^2}{8 C}$$

$$P = P_i + y q$$

The calculations for extension springs are done as per the compression springs based on the total load P.



VARIABLE STRESSES IN SPRINGS

P_a amplitude load, kgf

P_m mean load, kgf

τ_m mean shear stress, kgf / cm²

τ_a amplitude shear stress, kgf / cm²

τ_e endurance shear stress for repeated loading

k_s Wahl stress factor, k_{sh}, k_c

k_{sh} direct shear factor

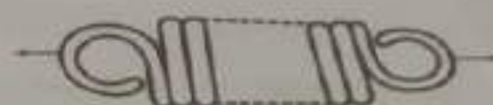
k_c curvature factor

n factor of safety

P total load on the spring, kgf

P_i initial load to separate the coils, kgf

τ_i maximum value of initial stress, kgf / cm²



Approximate maximum stresses at initial tension

C	3	4	5	6	7	8	9	10	11	12	13
τ_i kgf / cm²	1700	1600	1430	1300	1160	1000	930	830	760	690	50

DESIGN PROCEDURE for compression spring –varying load

Step1: find d, using soderberg eqn. for spring.7.102/DDB)
Find Mean load, amplitude load, Ʈm &Ʈa, finally find ' d'

Step2: Find the deflection of spring ' y ' (Pg. 7.100/DDB) (either y or n based on The available either y or n

Step3: Find the stiffness of the spring 'q ' (Pg. 7.100/DDB)

Step4: Find Lf- free length of coil

Lf = Ls + y + 15% of y

Ls= nxd n= no. of turns, d- wire diameter.

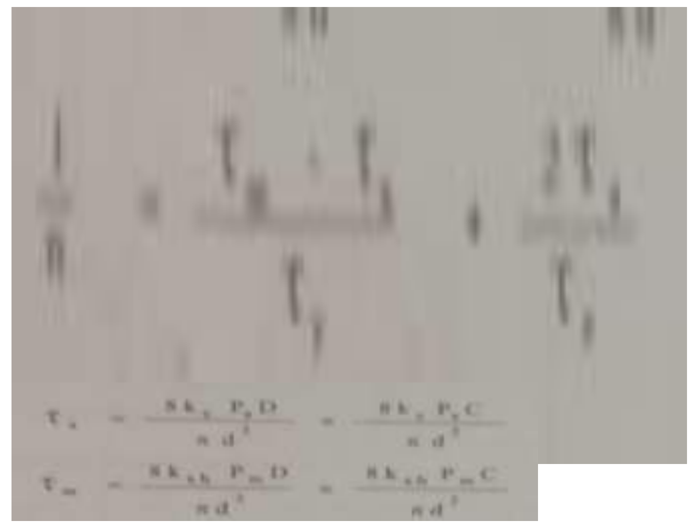
Step 5: Find pitch 'p '

Step 6 ; check for buckling Pg.7.101/DDB

Step 7: Check for surging (optional step) pg.7.101/DDB

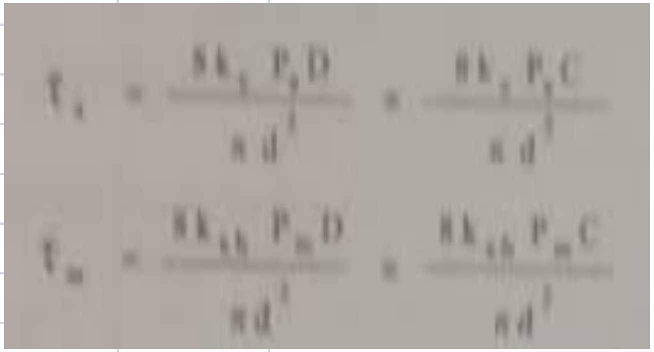
S3. A helical compression spring made of oil tempered carbon steel, is subjected to a Load which varies from 600N to 1600 N. The spring index is 6 and the factor of safety is 1.5 . The yield stress in shear is 700MPa. The endurance in shear is 350 N/mm^2. the compression at the maximum load is 40 mm. Take 80GPa. Design and draw the spring.

DATA		
Max load=		**1600N**
Min load=		**600N**
C=		**6**
[Ʈ] =		**700N/mm^2**
G=		**80Gpa**
	80 x 10^9	**N/m^2**
	80 x 10^3	**N/mm^2**
[Ʈ_1] = in shear		**350N/mm^2**
n=		**1.5**
Y=	**40 mm**	
Refer Pg.7.102/BBB		



Step1: Find 'd' Dm, Di,do using soderberg eqn. 7.102/DDB

Ʈm=?		
Ʈa=?		
Mean Load Pm	(Pmax+Pmin)/2	
	1100	N
amplitude Load	(Pmax - Pmin)/2	
Pa=	500	N



Ʈm=?		(8 x Ks xPm x C)/(3.141 x d^2)	
	=	21012.4	(1/d^2)
Ʈa=?		(8 x Ks xPa x C)/(3.141 x d^2)	
	=	9551.1	(1/d^2)

Use soderberg relation 7.102

$$\frac{1}{n} = \frac{\tau_m - \tau_a}{\tau_y} + \frac{2\tau_a}{\tau_e}$$

LHS=	1/1.5	0.66667	
	1 term		2 term
RHs=	16.3733	+	54.5777
	d^2		d^2
0.66667	=	70.951	(1/d^2)
d^2	=	106.427	
d	=	10.3163	
Std. 7.105/ddb	d=	10.5	mm

Now			
Dm=	C x d =	63	mm
Do=	Dm+d =	73.5	mm
Di=	Dm-d =	52.5	mm

N=13 +2=15

Lf=211 mm

Pitch=15mm

Leaf Spring Design

What is ?

A number of curved or cambered plates held together by by means of Centre bolt or shrank at its middle. Also known as semi – elliptic laminated lea spring. – (carriage spring)

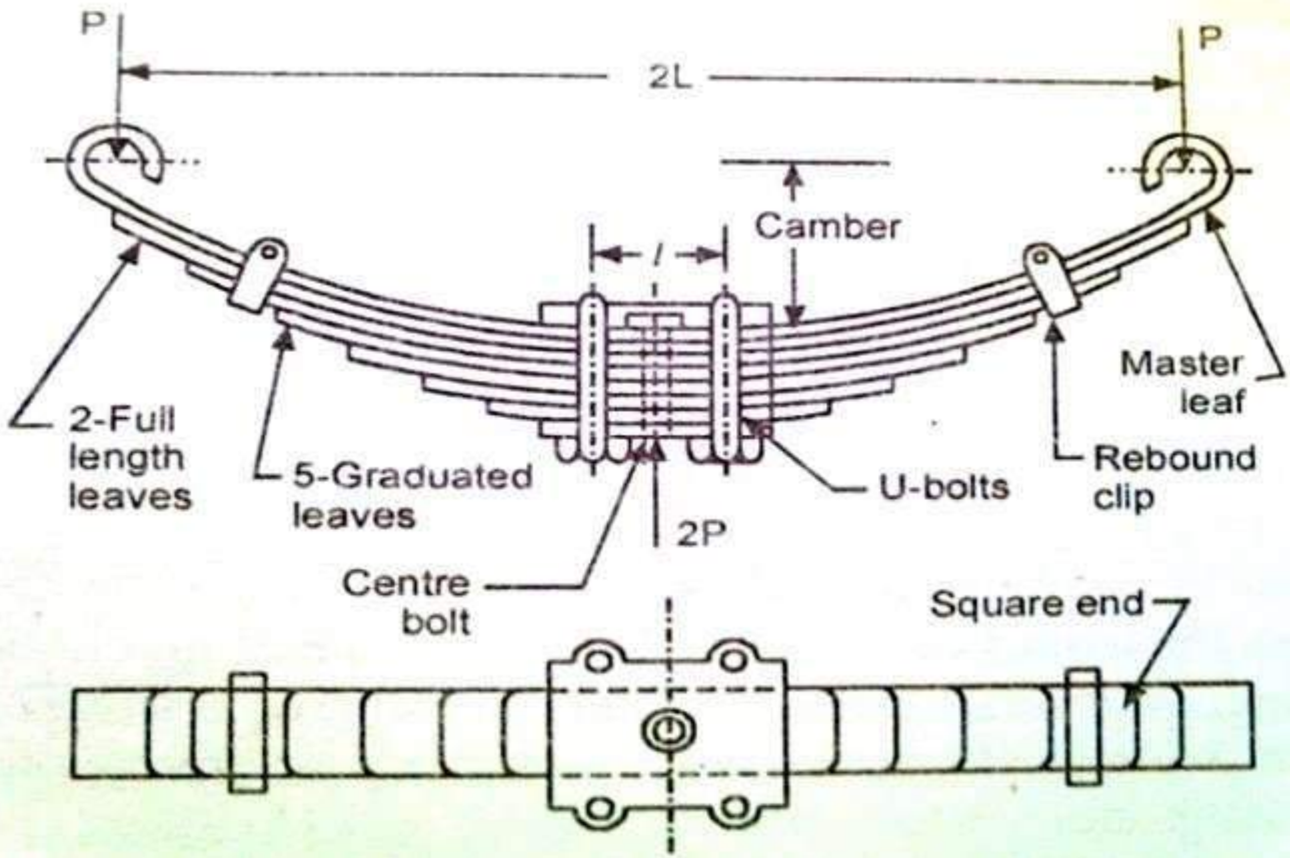
The leaves are two types 1. Full length leaves or master leaf.
2. graduated leaves

To avoid digging action between the leaves, the graduated leaves end are trimmed

To have curved or triangular shapes.

Master leaf has eyes at its ends. Through which , hanger or shackle pin are inserted
And supports the structure.

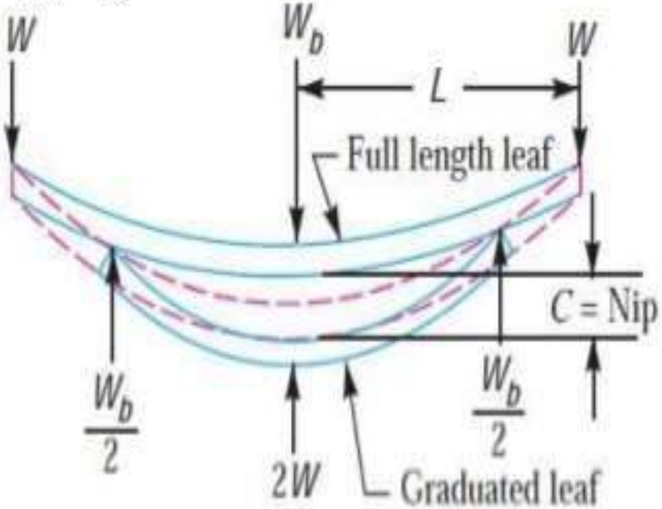
Widely used in automobile suspension systems

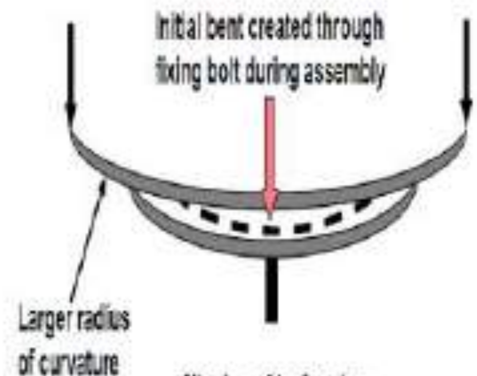


P
2L
P
Camber
l
Master leaf
2-Full length leaves
5-Graduated leaves
U-bolts
Rebound clip
2P
Centre bolt
Square end

NIPPING IN LEAF SPRING

Stress in the full length leaves is 50% greater than the stress in the graduated leaves. When the load is gradually applied to the spring, the full length leaf is relieved of the initial stress and then stressed in opposite direction. Such a pre stressing obtained by a difference of radii of curvature is known as nipping.





Nipping of leaf spring

Design procedure for laminated spring

Leaf spring is treated as cantilever beam

Springs with full length leaves

Refer Pg.7.104/DDB

Objectives

1. If the sizes b and t o f the leaves are given find stress induced and Y deflection , nip

2. Stress will be given, b & t to be calculated

Step1. Cal. Of σb pg.7.104/DDB

Step2. cal of y deflection

Step3. cal. Of Nip h

When you have extra full length and graduated leaves, the same method to be followed, one step to be added as follows

Step-4 : find the all the leaves length

Treating the spring as a cantilever beam of uniform strength

$$\sigma_b = \frac{6PL}{nbt^2} \quad ; \quad y = \frac{6PL^3}{Enbt^3}$$

Nip, $h = \dfrac{2PL^3}{nEbt^3}$

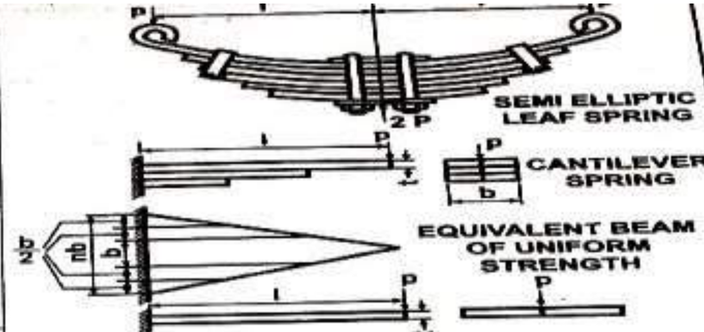
Load on the clip bolts required to close the gap, $P_b = \dfrac{2n_e n_g P}{n(2n_g + 3n_e)}$

Spring with extra full length leaves

$$\sigma_{bg} = \frac{12PL}{bt^2(3n_e + 2n_g)}$$

$$\sigma_{be} = \frac{18PL}{bt^2(3n_e + 2n_g)}$$

$$y = \frac{12PL^3}{bt^3(3n_e + 2n_g)}$$



2 P	load on the spring, kgf
y	deflection of the spring, cm
L	Length of the cantilever beam (spring) cm
b	width of the leaf, cm
t	thickness of the leaf, cm
n	number of leaves in the spring
n_g	number of graduated leaves
n_e	number of extra full length leaves
σ_b	bending stress, kgf / cm^2
σ_{bg}	bending stress in graduated leaves, kgf / cm^2
σ_{be}	bending stress in extra full length leaves kgf / cm^2
E	modulus of elasticity, kgf / cm^2

BELLEVILLE SPRINGS

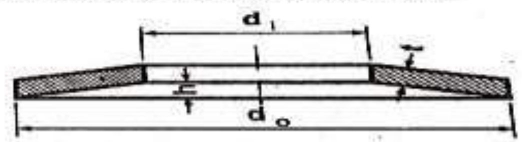
$$P = \frac{Ey}{(1-\nu^2)M\left(\frac{d_o}{2}\right)^2}\left[\left(h - \frac{y}{2}\right)(h-y)t + t^3\right]$$

$$\sigma = \frac{Ey}{(1-\nu^2)M\left(\frac{d_o}{2}\right)^2}\left[C_1\left(h - \frac{y}{2}\right) + C_2 t\right]$$

$$M = \frac{6}{\pi \log_e\left(\frac{d_o}{d_i}\right)}\left[\frac{\left(\frac{d_o}{d_i} - 1\right)}{\frac{d_o}{d_i}}\right]^2$$

$$C_1 = \frac{6}{\pi \log_e\left(\frac{d_o}{d_i}\right)}\left[\frac{\frac{d_o}{d_i} - 1}{\log_e\left(\frac{d_o}{d_i}\right)} - 1\right]$$

$$C_2 = \frac{6}{\pi \log_e\left(\frac{d_o}{d_i}\right)}$$



P	axial load on the spring, kgf
d_i	internal diameter, cm
d_o	outside diameter, cm
y	deflection, cm
t	thickness of the disc, cm
h	free height minus thickness, cm
E	modulus of elasticity, kgf / cm^2
ν	Poisson's ratio
σ	stress at the inside circumference, kgf / cm^2
M, C_1, C_2	constants

LS1:A truck spring has 10 leaves and is supported at a span of length 100cm with a central band of 80 mm wide. A load of 6 kN is applied at the centre of the spring whose permissible stress is 300 Mpa. The spring has the ratio of total depth to width of about 2.5. Design the spring.

DATA

Load 2P=	6000N	
P=	3000N	
Central band		
a =	80mm	
[σb] =	300N/mm^2	
span (2L)	1000mm	
L =	500mm	
No.of leaves (n)	10	
assume ne=	2	(at least 1 and maximun 3)
assume ng=	8	
n= ne+ng		
Effective length=	2L-a	
	920mm	
now new l=	460mm	

ratio= Total depth of spring/ width

total depth= n x t

breadth= b

therfore ,	nt/b=2.5	
	b=nt/2.5	
	b=4t	[n=10]

objective to find b & t

objective to find b & t

Step1: find d & t using extra full length leaves eqn.

refer pg.7.104/DDB

$$\sigma_{be} = \frac{18\,P\,L}{b\,t^2\,(3\,n_e + 2\,n_g)}$$

P=	3000
L=	460
b=	4t
ne=	2
ng=	8
[σ]=	300

LHS =						
	300					
RHS=						
	Nr=	18 x 3000 x 460		=	24840000	
	DR=	4t.t^2[(3.2)+(2.8)]			88	t^3
					282272.7	(1/t^3)
=	282273	(1/t^3)				
RHS	=	LHS				
300	=	282273	(1/t^3)			
t^3	=	940.91				
t	=	9.7767				
t	=	10	mm			
now, b	=	4 x t				
	=	40	mm			

Step2: Finding defelection y

$$y = \frac{12 P L^3}{b t^3 (3 n_e + 2 n_g)}$$

Refer pg.7.104/DDB

E= to be introduced

Y= Px L/AXE

p=	3000
L=	460
ne=	2
ng=	8
b=	40
t=	10

E= 2.0 x 10^5 N/mm^2

Y=

Step3: Find Nip h

$$h = \frac{2 P L^3}{n E b t^3}$$

h= 7.3002 mm

step4: Load exreted on band clips

$$P_b = \frac{2 n_e n_g P}{n(2n_g + 3n_e)}$$

Pg.7.104/DDB

Pb	436.364	N

Step5. Find the lengths of the leaves

Fisrt start with small leaf as l1

nth leaf length= effective length/(n-1) x n1 + Inefective length

	Effective length=		920
		n=	10
Ineffecctive length=		a=	80
		n=	2
nth leaf			

nth leaf

1	1st leaf=	182.22	mm	Ln=(920/(10-1) x1) +80
2	2nd leaf=	284.44	mm	
3	3rd leaf=	386.67	mm	
4	4th leaf=	488.89	mm	
5	5th leaf=	591.11	mm	
6	6thleaf=	693.33	mm	
7	7th leaf=	795.56	mm	
8	8th leaf=	897.78	mm	
9	9th leaf=	1000.00	mm	
10	10th leaf=	1102.22	mm	

But Masterleaf

But Masterleaf

actual span is = 2L

Now ,

eye length to be added

since its a circular shape

llinear of circular = π(d+t)

d= inside dia of eye

if not given, assume as 20mm

No.of Eyes =2

Therefore

Length of master leaf as follows

Lm= 2l+2 eyes circumference

= 2l+2(3.141x20 +20)

Lm= 1188.46 mm 10

Some special kind problems

(1) Safety Valve (Springs)

Important points

① . Initial compression given, deflection given

$y_i + Y$

$\{ Y_T = Y_i + Y \} \rightarrow$ used to find "n"

② . Value seat Diameter : given.

$\therefore W = P \cdot A$, $A = \frac{\pi}{4}(d_v)^2$

P = operating pressure

This "W_{op}" ⇒ used in both "τ & Y" equation

(3). Max. load Calculation (for)

Y_i = initial deflection given

Y = Lift or deflection obtained

$Y_T = Y_i + Y$

$\therefore W_{max} = \frac{W_o}{Y_i} \times Y_T = W_{max}$

(4). Using Energy Equ:

$U = \frac{1}{2} P \cdot Y$ P – load, Y – deflection

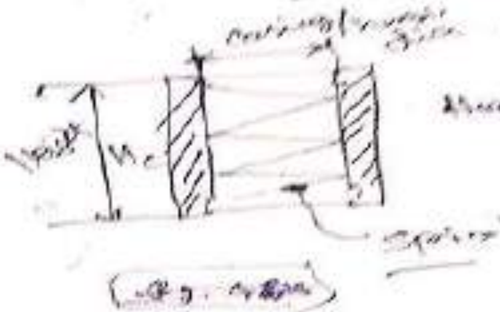
$\Rightarrow KE = \frac{1}{2} mv^2$ – Wagon (railway)

→ check for no. of springs. 2 (or) 3

$U = \frac{1}{2} P \cdot Y \cdot (n)$ n → no. of springs

(5). Condition Problems.

* "D" — outer coil diameter is fixed
C → spring index in [illegible] [illegible]
[illegible] [illegible]
L = length of the spring [illegible] [illegible]



Housing Inside/casing Inside Dia - given.

$\therefore$ D_o = casing D_i

$H_c = L_{spring}$

$\boxed{L_s \geq H_c}$ - This condition (453) also to be satisfied.

Spring's convention representation

Learn for Spring arrangement
Springs in series
in parallel.

Note: 1. Load Range " — " static only."
2. Varying load / fluctuating load.
→ Dynamic

Sp2.

Design a spring for spring loaded safety valve for the following condition

Operating pressure=1 Mpa

Diameter of the valve seat =110mm

Design shear stress or the spring =360MPa.

G= 82GPa.

The spring is kept in the casing of 130 mm inner diameter and 400 mm long, The spring should be at maximum lift of 6 mm when the pressure is 1.08 MPa.

Answers

C – to be assumed 4/5/6

Load: operating load and maxi load to lift the spring

Load 1= N

Load2= N

Load = P x area of valve seat,

d=110mm

Load1-load2 = N

Y=6 mm

1. Find Dm, Do is = Di= Do

Dm= mm

2. Find n , n=

W= L1-L2

Ls= mm

Lf= mm

P= mm

Lf/D=

Springs in Series

- *Consider two springs with force constants k_1 and k_2 connected in series supporting a load $F = mg$.*
- *Let the force constant of the combination be represented by k*
- *For the combination, supporting the load F=mg:*

$$F = kx \quad (\text{where } x = \text{the total stretch})$$

and $$x = \frac{F}{k}$$

- *For each spring*

  - the bottom supports mg=F and stretches by x_1

$$F = k_1 x_1 \quad or \quad x_1 = \frac{F}{k_1}$$

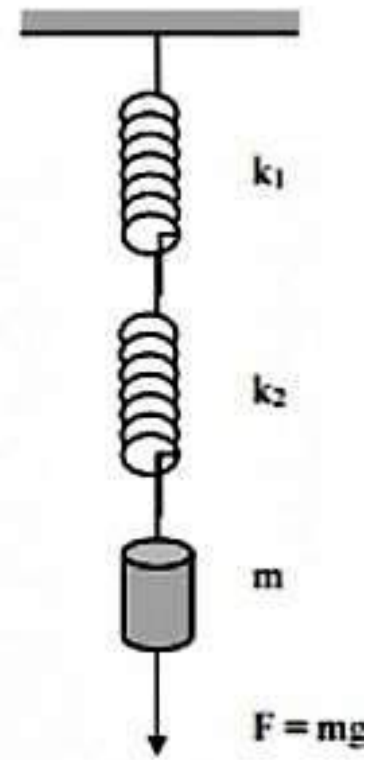
  - the top spring support mg plus the weight of the bottom spring (which is negligible - Thus F is the stretching force for both springs)

$$F = k_2 x_2 \quad or \quad x_2 = \frac{F}{k_2}$$

- *The total stretch*

$$x = x_1 + x_2 \quad or \quad \frac{F}{k} = \frac{F}{k_1} + \frac{F}{k_2}$$

and $$\boxed{\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2}}$$



Springs in Parallel

- *Consider two springs with force constants k_1 and k_2 connected in parallel supporting a load $F = mg$.*
- *Let the force constant of the combination be represented by k*
- *For the combination supporting the load $F=mg$:*

  $F = kx$ *(where x = the total stretch)*

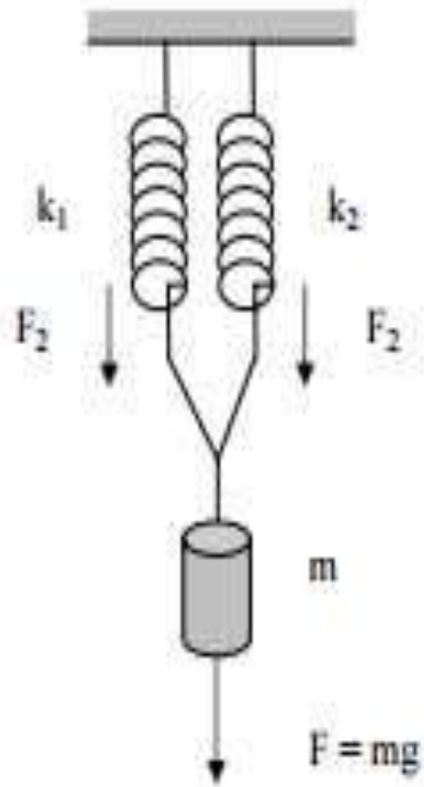
- *The two individual springs both stretch by x but share the load ($F = F_1 + F_2$) and*

  $F_1 = k_1 x$ *while* $F_2 = k_2 x$

- *Thus the total force is*

  $F = F_1 + F_2$ *or* $kx = k_1 x + k_2 x$

and $\boxed{k = k_1 + k_2}$



Design procedure for disc or Belleville spring

Used: Less or compact space need high stiffness

Step1 Find load 'P' or 'Y' on the spring

Refer Pg. 7.104/DDB,

P=? Based on data , iF 'p' given, find y or vice versa

Step2. Find stress "σ " on the spring

Refer Pg. 7.104/DDB,

σ=?

Prior to the above steps

TO be calculated M, C1 & C2.

Treating the spring as a cantilever beam of uniform strength

$$\sigma_b = \frac{6PL}{nbt^2} \quad ; \quad y = \frac{6PL^3}{Enbt^3}$$

Nip, $h = \dfrac{2PL^3}{nEbt^3}$

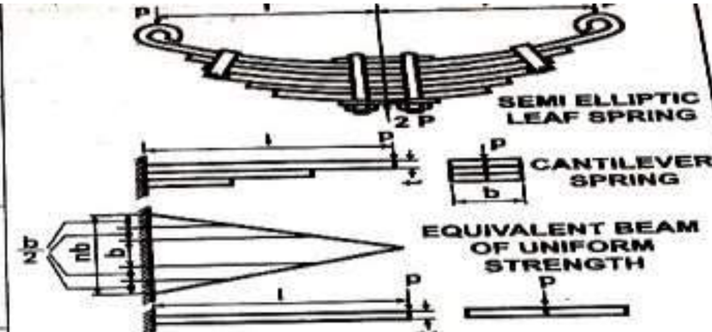
Load on the clip bolts required to close the gap, $P_b = \dfrac{2n_e n_g P}{n(2n_g + 3n_e)}$

Spring with extra full length leaves

$$\sigma_{bg} = \frac{12PL}{bt^2(3n_e + 2n_g)}$$

$$\sigma_{be} = \frac{18PL}{bt^2(3n_e + 2n_g)}$$

$$y = \frac{12PL^3}{bt^3(3n_e + 2n_g)}$$



2 P	load on the spring, kgf
y	deflection of the spring, cm
L	Length of the cantilever beam (spring) cm
b	width of the leaf, cm
t	thickness of the leaf, cm
n	number of leaves in the spring
n_g	number of graduated leaves
n_e	number of extra full length leaves
σ_b	bending stress, kgf / cm^2
σ_{bg}	bending stress in graduated leaves, kgf / cm^2
σ_{be}	bending stress in extra full length leaves kgf / cm^2
E	modulus of elasticity, kgf / cm^2

BELLEVILLE SPRINGS

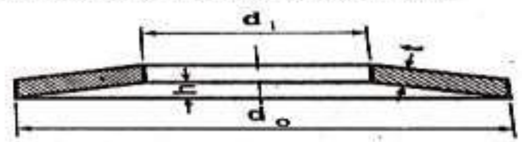
$$P = \frac{Ey}{(1-\nu^2)M\left(\frac{d_o}{2}\right)^2}\left[\left(h - \frac{y}{2}\right)(h - y)t + t^3\right]$$

$$\sigma = \frac{Ey}{(1-\nu^2)M\left(\frac{d_o}{2}\right)^2}\left[C_1\left(h - \frac{y}{2}\right) + C_2 t\right]$$

$$M = \frac{6}{\pi \log_e\left(\frac{d_o}{d_i}\right)}\left[\frac{\left(\frac{d_o}{d_i} - 1\right)}{\frac{d_o}{d_i}}\right]^2$$

$$C_1 = \frac{6}{\pi \log_e\left(\frac{d_o}{d_i}\right)}\left[\frac{\frac{d_o}{d_i} - 1}{\log_e\left(\frac{d_o}{d_i}\right)} - 1\right]$$

$$C_2 = \frac{6}{\pi \log_e\left(\frac{d_o}{d_i}\right)}$$



P	axial load on the spring, kgf
d_i	internal diameter, cm
d_o	outside diameter, cm
y	deflection, cm
t	thickness of the disc, cm
h	free height minus thickness, cm
E	modulus of elasticity, kgf / cm^2
ν	Poisson's ratio
σ	stress at the inside circumference, kgf / cm^2
M, C_1, C_2	constants

SP3 Design a disc spring or the following specifications Spring is made of 4 mm steel sheet has 120 mm outer diameter and 60 mm inner diameter . It is dished by 5 mm . Calculate when deflection of the spring is 2.5 mm due to an axial load P. Also calculate the stress induced in the spring take E=200kN/mm^2, Poisson's ratio as0.3

t= 4, do=120, di=60 mm, h=5,
y=2.5mm, E=200 x 10^3 N/mm^2

Answers:

M=[redacted]

C1=[redacted]

C2=[redacted]

Y=[redacted] **P=** [redacted]

Stress= [redacted]

UNIT IV

- ME18503-Design Of Machine Elements

UNIT IV DESIGN FOR RIVETED AND WELDING JOINTS, FASTNERS

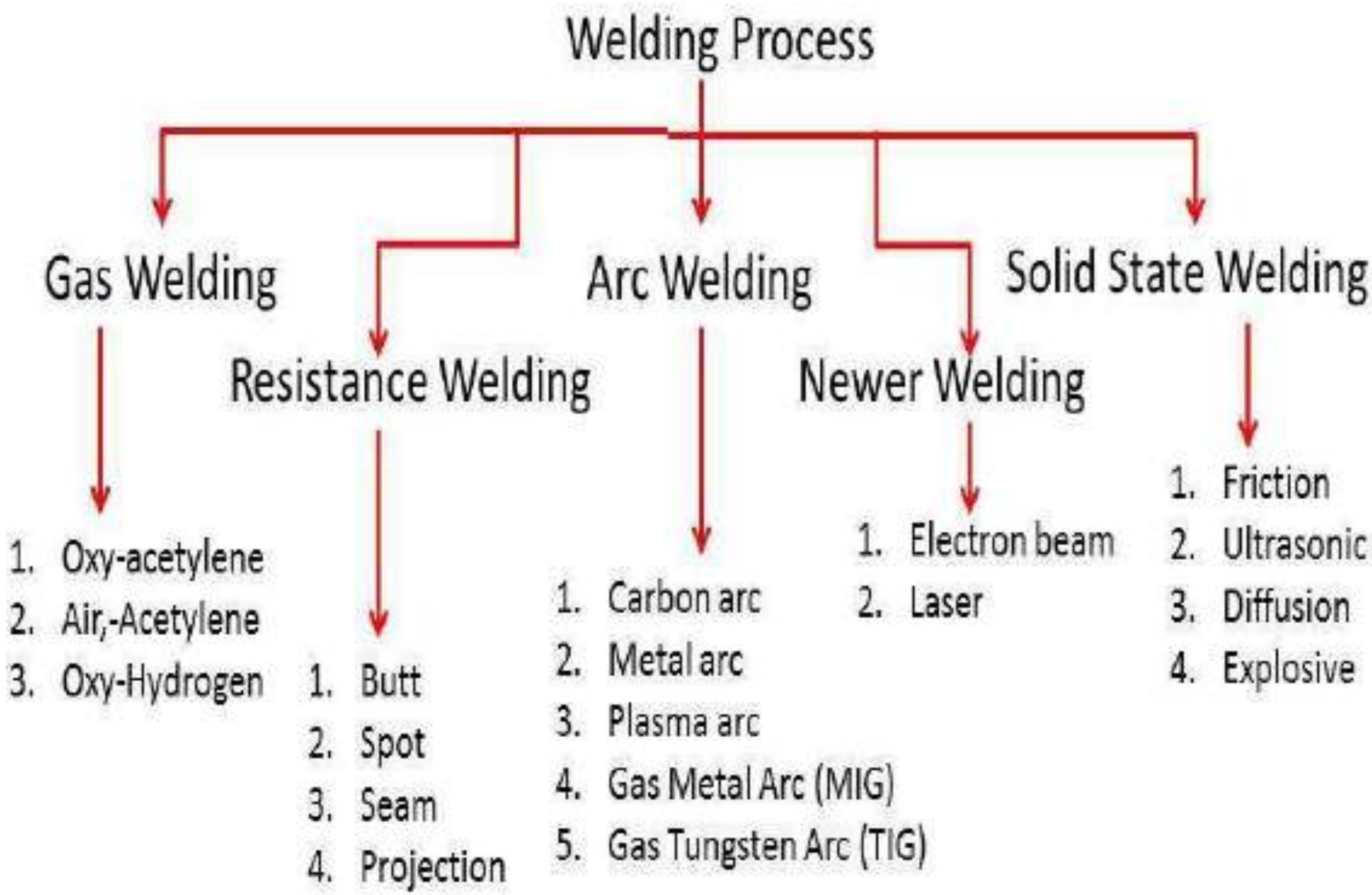
Rivet – Types of rivet joints, Caulking and fuluring, Design of riveted joints for structural and pressure vessels. Eccentrically loaded rivet joint, Welding – Welding symbols, Design of welded joints under eccentrically load. Geometry of thread forms, Terminology of screw threads.Design of screws and bolts.

OBJECTIVE

- **This course will enable to check strength of fasteners kind of both rivet and welding.**

COURSE OUT COME 4 - CO4

Proficient in Design of riveted joint and welding joints under eccentric loading


Welding Process
Gas Welding
Resistance Welding
Arc Welding
Newer Welding
Solid State Welding
1. Oxy-acetylene
2. Air,-Acetylene
3. Oxy-Hydrogen
1. Butt
2. Spot
3. Seam
4. Projection
1. Carbon arc
2. Metal arc
3. Plasma arc
4. Gas Metal Arc (MIG)
5. Gas Tungsten Arc (TIG)
1. Electron beam
2. Laser
1. Friction
2. Ultrasonic
3. Diffusion
4. Explosive

Designs

1. **Rivet design under axial load and eccentric loading**

2. **Welding design under axial load and eccentric loading**

Welding

What ?

joining metal by heat and with or with out application of pressure

Why?

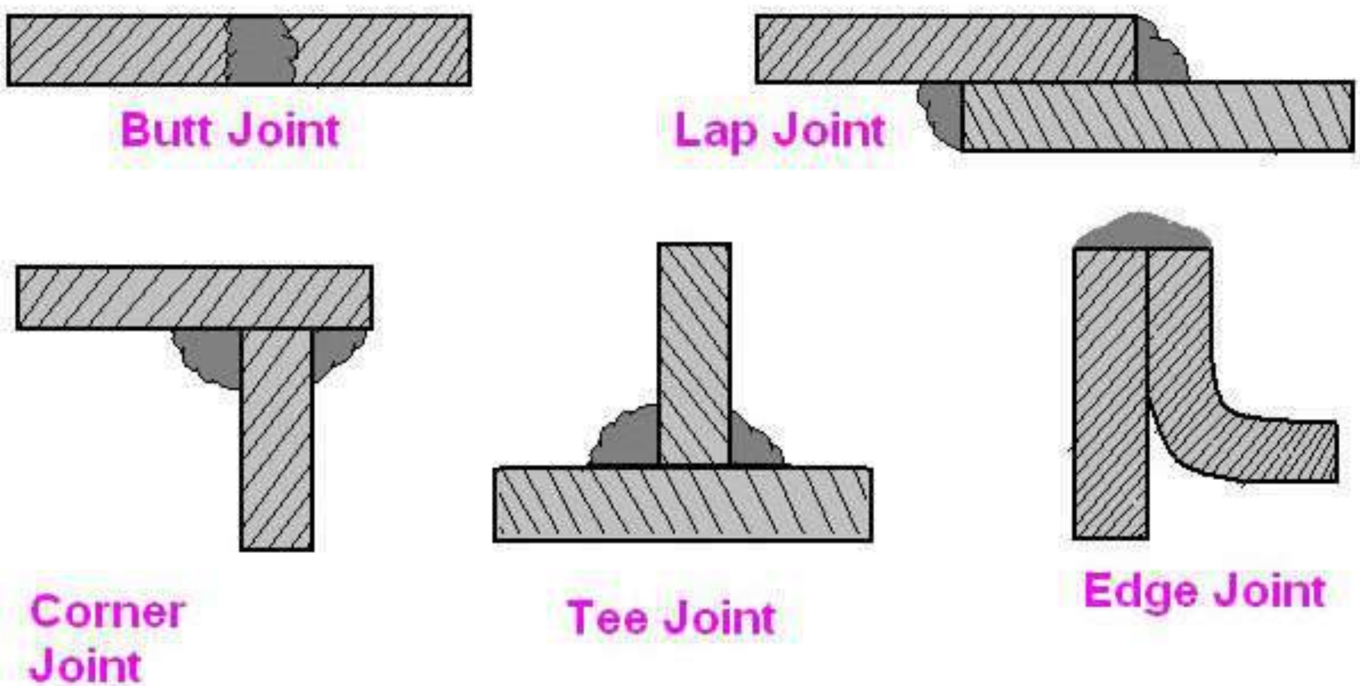
Large size parts can not be manuactuterd by casting process

Ex. Ship buliding

Types:

Lap joint butt joint joining method

other types are there

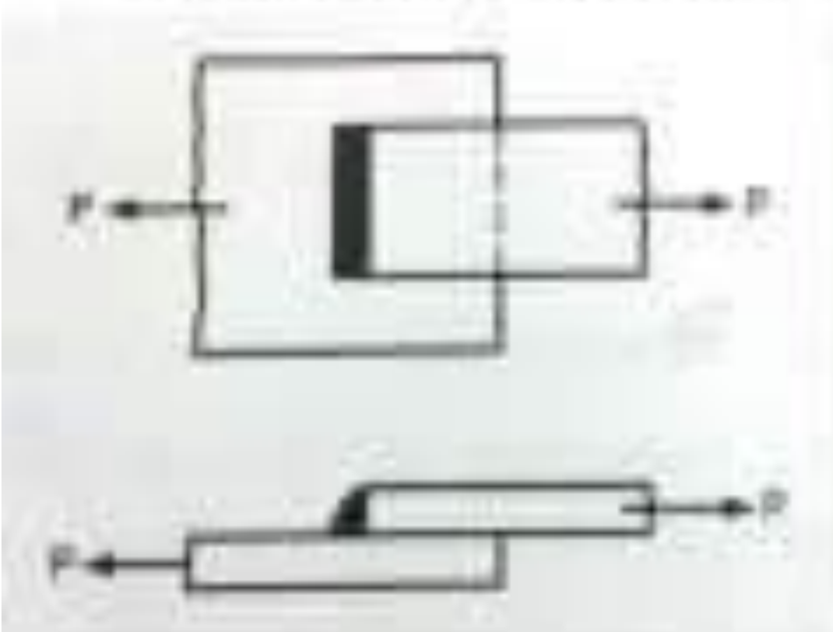

Butt Joint
Lap Joint
Corner Joint
Tee Joint
Edge Joint

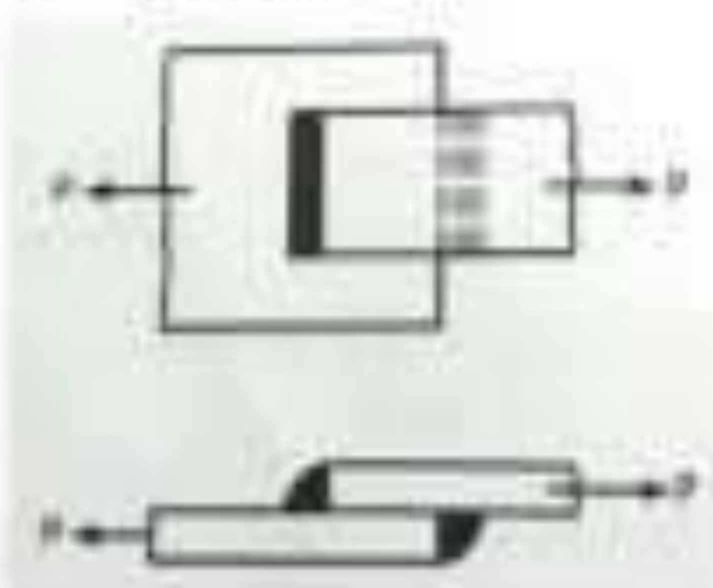
❑ Types of Welded Joints:

a. Lap joint or fillet joints
b. Butt joint

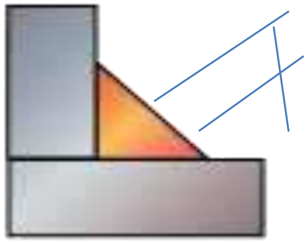
(a) Types of fillet joint:

1)Single transverse 2) Double transverse

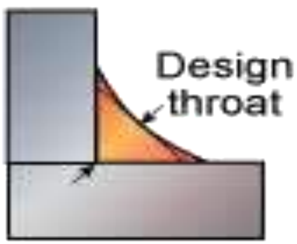




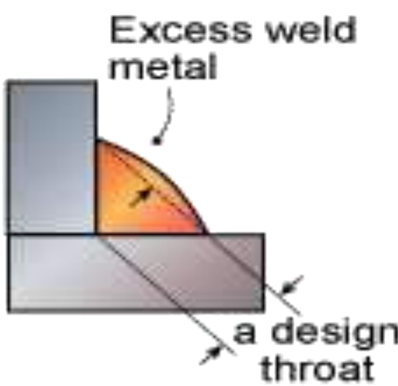
Throat



- Mitre:-
shown as a right angled triangle with a design throat thickness 'a'   a = 0.7072



- Concave:-
as above, but with a reduced design throat thickness.

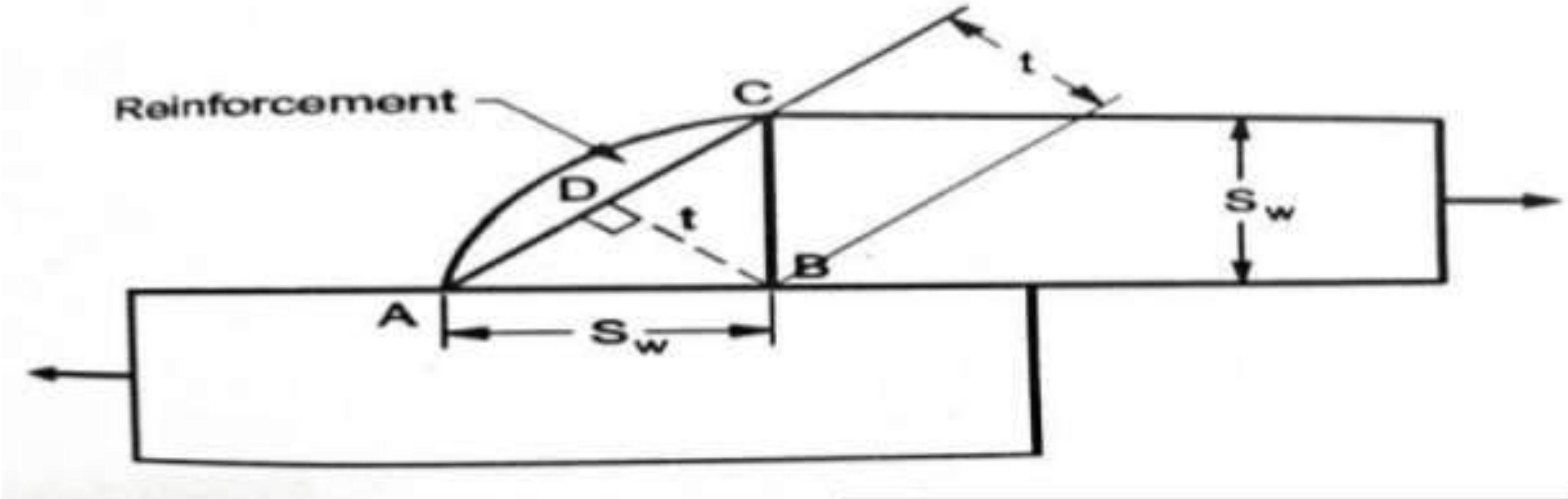


- Convex:-
as for the mitre shape but with an increased design throat thickness and consequential excess weld metal

Understanding of fillet weld

Strength of Transverse Fillet Welded Joint:-

- Consider a single transverse fillet weld.



ΔABC is a right angle isosceles triangle
Let, t=BD=Throat thick. In mm
Sw=AB=BC=size of weld
*L*w=Length of weld in mm
<BAC=<BCA=45°

T= sin45 x h

Objectives of the weld design

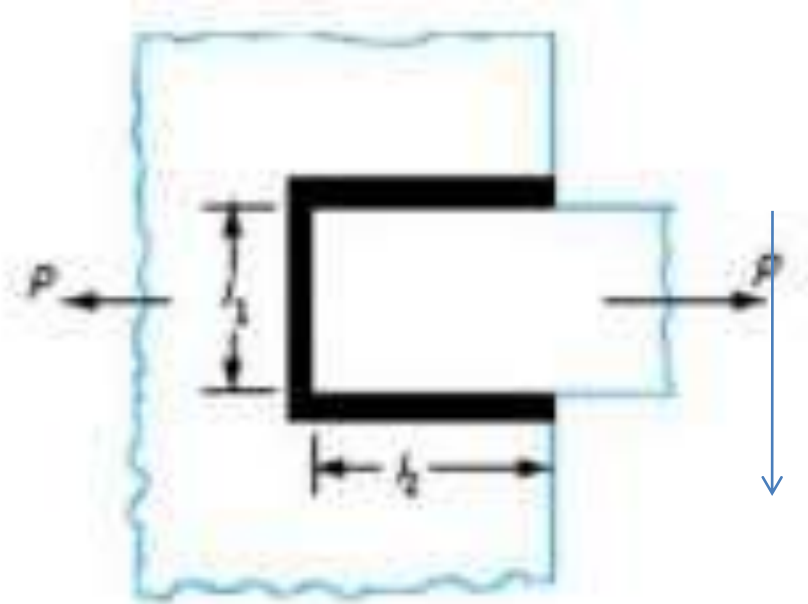
Weld size as " h" or " length"
(both axial load and eccentric load)

Load = area x stress

Area of throat of weld bead or run or fillet

Stress= tensile and shear are major in consideration

bending will also be considered based on the structure and load application

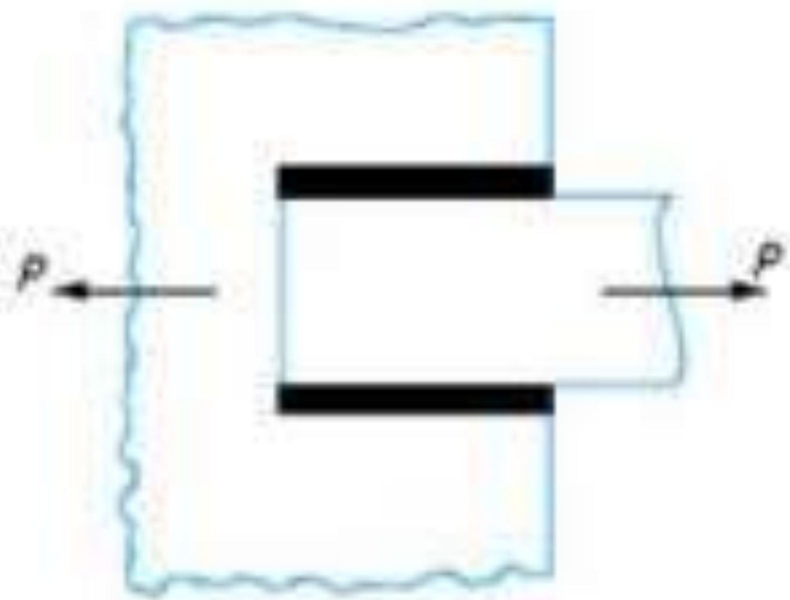


(2) Combination of transverse and parallel fillet weld.

3 fillets

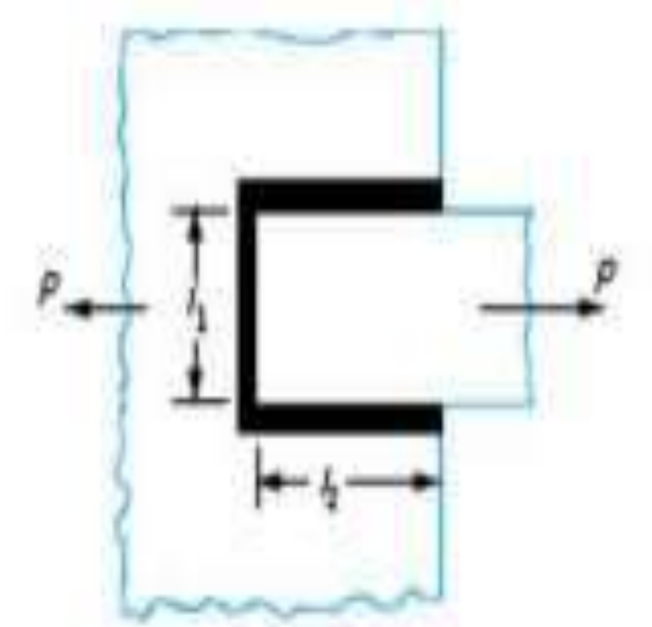
1- set parallel fillet or double parallel Fillet

1. Transverse i fillet



(a) Double parallel fillet weld.

WP1 A plate 75 mm wide and 12.5 mm thick is joined with another plate as shown in figure The maximum tensile and shear stresses are 70 MPa and 56 MPa respectively.



(3) Combination of transverse and parallel fillet weld

Data

Width plate = 75 mm

H= 12.5 weld size

L1= Width of plate =75 mm

L2=?

Total load by the weld joint **(Tensile Load)**

Load by L1 by P1 single

$$P_1 = \text{Throat area} \times \text{Allowable shear stress} = 0.707\, s \times l \times \tau$$

Load by L2 by P2 double

$$P_2 = \text{Throat area} \times \text{Allowable shear stress} = 0.707\, s \times l \times \tau$$

Step1: Cal. Of total load

P= stress x area

Stress= 70 Mpa

Area= 75 x 12.5

P= 65 625 N

Step2: cal. Of Load P1

P1= 0.707 x h x l1 x σt

= 0.707 x h x 75 x 70/56

h= plate thickness= 12.5

P1= 46397 N

~~38 664 N~~

Step3: cal of L2

Use P2= 1.414 h x L2 x τ (2 Single fillet = 2 x 0.707 h)

= 1.414 x h x 56

= 990L2 N

Now P= P1 + P2

65 625 = 46397 ~~38 664~~ + 990 L2

L2=19.43 ~~27.2~~ + weld size

Weld size = 12.5 L2= ~~39.7~~ 31.93 mm

Important pages to be used from DDB

11.3

11.4

11.5

11.6

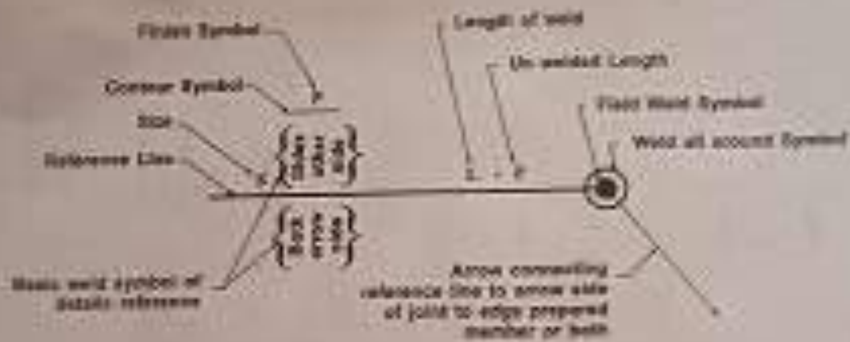
General information

11.1 & 11.2

WELDED JOINTS

TYPE OF WELDS AND SYMBOLS

Form of weld	Sectional representation	Appropriate symbol	Form of weld	Sectional representation	Appropriate symbol
Fillet			Plug or slot		
Square butt					
Single-V butt			Backing strip		
Double-V butt			Spot		
Single-U butt					
Double-U butt			Seam		
Single bevel butt			Mashed Seam		
Double bevel butt			Stitch		
Single-J butt			Mashed Stitch		
Double-J butt					
Stud			Projection		
Bead (edge or seal)			Flash		
Sealing run			Butt Resistance or Pressure (Upset)		



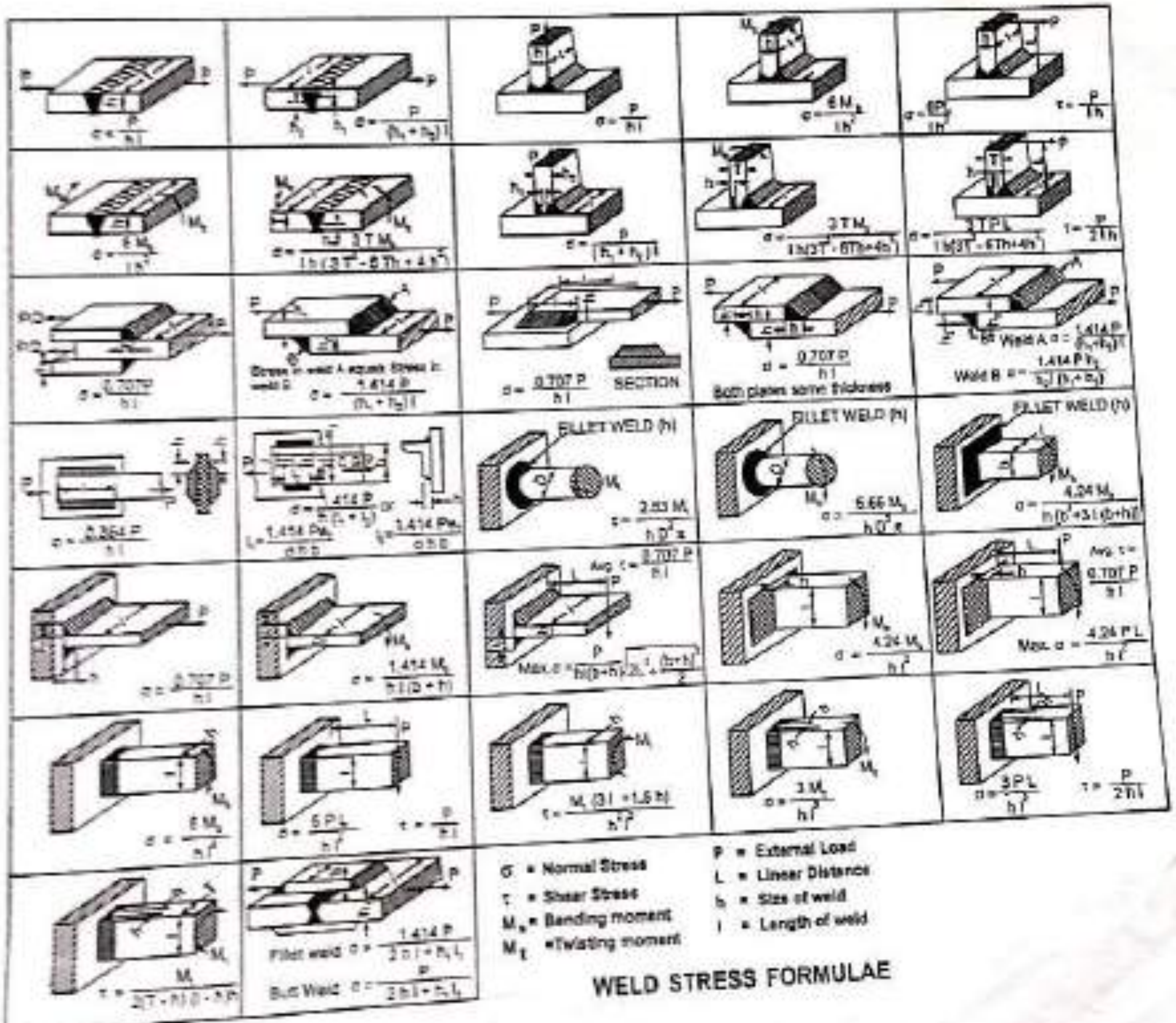
STANDARD LOCATION OF ELEMENTS OF A WELDING SYMBOL

Special Instructions	Drawing Representation	Symbol
Weld all-round		○
Site weld (Field weld)		●
Flush contour		—
Convex contour		⌒
Concave contour		◡
Grinding finish		G
Machining finish		M
Chipping finish		C

WELDS TO BE MADE ON THE ARROW SIDE OF A JOINT

WELDS TO BE MADE ON THE OTHER SIDE OF A JOINT

WELDS TO BE MADE ON BOTH SIDE OF A JOINT



σ = Normal Stress

τ = Shear Stress

M_b = Bending moment

M_t = Twisting moment

P = External Load

L = Linear Distance

h = Size of weld

l = Length of weld

WELD STRESS FORMULAE

PLATE THICKNESS AND WELD SIZE

Plate thickness, mm	3 to 5	6 to 8	10 to 16	18 to 24	26 to 35	over 38
Weld size, mm	3	5	6	10	14	20

* Permissible static load per cm length : (Mild Steel Fillet welds)

Weld size mm	Permissible static load, kgf / cm			
	Bare Electrode		Covered Electrode	
	in tension	in shear	in tension	in shear
3	170	135	210	170
4	225	180	280	225
5	280	225	350	280
6	335	270	420	335
8	450	360	560	450
10	560	450	700	560
12	670	540	840	670
14	785	630	980	785
16	900	720	1120	900
20	1120	900	1400	1120

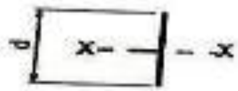
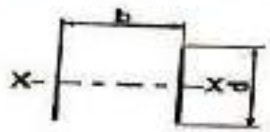
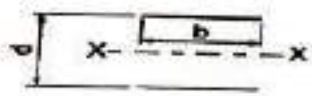
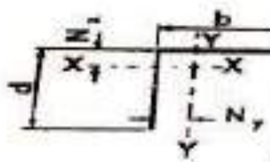
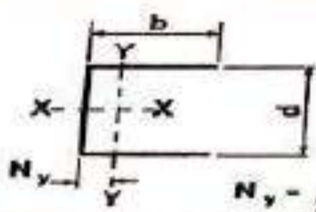
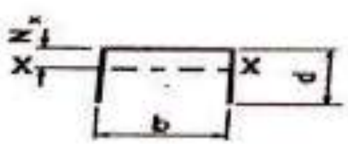
*For machine welding increase the permissible load by 25 to 30 %

DESIGN STRESSES FOR WELDED JOINTS, kgf / cm^2

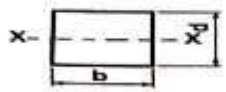
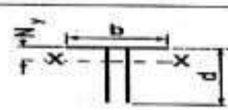
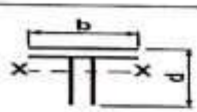
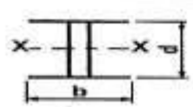
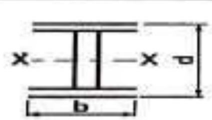
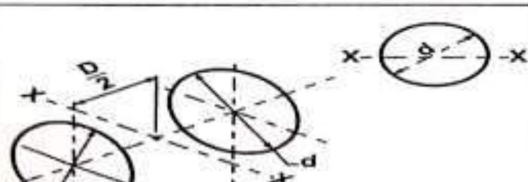
Kind of weld and stresses	Bare Electrodes		Covered Electrodes		Fatigue Design Stresses			
					Structural Steels		Alloy Steels	
	Steady load	Reversed load	Steady load	Reversed load	2×10^6 cycles	10^5 cycles	2×10^6 cycles	10^5 cycles
Butt Welds								
Tension	900	350	1100	550	$\frac{1100}{1 - 0.8r}$	$\frac{1250}{1 - 0.5r}$	$\frac{1150}{1 - 0.8r}$	$\frac{2300}{1 - 0.6r}$
Compression	1000	350	1250	550	$\frac{1250}{1 - 0.5r}$	$\frac{1250}{1 - 0.5r}$	-	-
Shear	550	210	700	350	$\frac{625}{1 - 0.5r}$	$\frac{900}{1 - 0.5r}$	-	-
Fillet Welds								
All	790	210	950	350	-	-	-	-

$r = \frac{F_{min}}{F_{max}}$

PROPERTIES OF WELD TREATED AS A LINE

Outline of welded joint	BENDING about horizondal axis XX, Z_w	TWISTING about centroidal axis, J_w
	$\frac{d^2}{6}$	$\frac{d^3}{12}$
	$\frac{d^2}{3}$	$\frac{d(3b^2+d^2)}{6}$
	$b\,d$	$\frac{b^3+3bd^2}{6}$
 $N_y = \frac{b^2}{2(b+d)}$ $N_x = \frac{d^2}{2(b+d)}$	$\frac{4bd+d^2}{6}$, top $\frac{d^2(4bd+d)}{6(2b+d)}$, bottom	$\frac{(b+d)^4-6b^2d^2}{12(b+d)}$
 $N_y = \frac{b^2}{2b+d}$	$b\,d + \frac{d^2}{6}$	$\frac{(2b+d)^3}{12} - \frac{b^2(b+d)^2}{2b+d}$
 $N_x = \frac{d^2}{\ldots}$	$\frac{2bd+d^2}{3}$, top $\frac{d^2(2b+d)}{3(b+d)}$, bottom	$\frac{(b+2d)^3}{12} - \frac{d^2(b+d)^2}{b+2d}$

PROPERTIES OF WELD TREATED AS A LINE (contd...)

Outline of welded joint	BENDING about horizondal axis XX, Z_w	TWISTING about centroidal axis, J_w
	$bd + \frac{d^2}{3}$	$\frac{(b+d)^3}{6}$
 $N_y = \frac{d^2}{b+2d}$	$\frac{2bd+d^2}{3}$, top $\frac{d^2(2b+d)}{3(b+d)}$, bottom	$\frac{(b+2d)^3}{12} - \frac{d^2(b+d)^2}{(b+2d)}$
 $N_y = \frac{d^2}{2(b+d)}$	$\frac{4bd+d^2}{3}$, top $\frac{4bd^2+d^3}{6b+3d}$, bottom	$\frac{d^3(4b+d)}{6(b+d)} + \frac{b^3}{6}$
	$bd + \frac{d^2}{3}$	$\frac{b^3+3bd^2+d^3}{6}$
	$\frac{2bd+d^2}{3}$	$\frac{2b^3+6bd^2+d^3}{6}$
	$\frac{\pi d^2}{4}$ $\frac{\pi d^2}{2} + \pi D^2$	$\frac{\pi d^3}{4}$

Points to identify

1. How many fillets are there

2. Static load

3. Fatigue load --- Kt to be used to find the weld size.

4. Eccentric loaded – weld bead are treated with line
5. In pg 11.5 & 11.6 , the given formula to be “x “ by ‘t’

Design Procedure
[Welding → Eccentric load]

Step 1: Find & locate "G" - Centre of Gravity.

Step 2: Introduce P_1 & P_2 at failure Point "A"

Step 3: Find τ_1 - direct shear stress $\tau_1 = ?$ $\tau_1 = \frac{P}{A}$

Step 4: Find τ_2 → secondary shear stress

$\tau_2 = ?$

$\tau_2 = \frac{P \times e \cdot r_2}{J}$

$r_2 = ?$

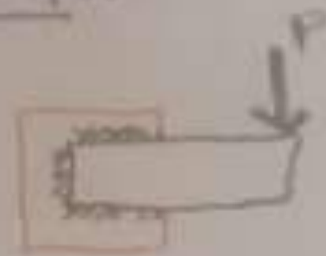
$\cos\theta = \frac{r_1}{r_2}$

Step 5: $\tau_R = ?$

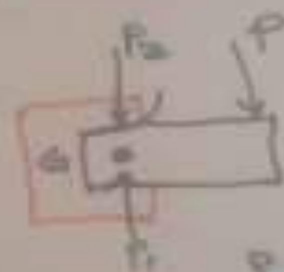
$\tau_R = [\tau]$, $\tau_R = \sqrt{\tau_1^2 + \tau_2^2 + 2\tau_1 \cdot \tau_2 \cos\theta}$

Eccentric Load Concept:

1. Find "G" – CG.
2. Introduce $P_1 + P_2$

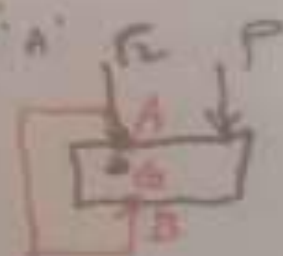


3.



$P_1 = P_2 = P$.

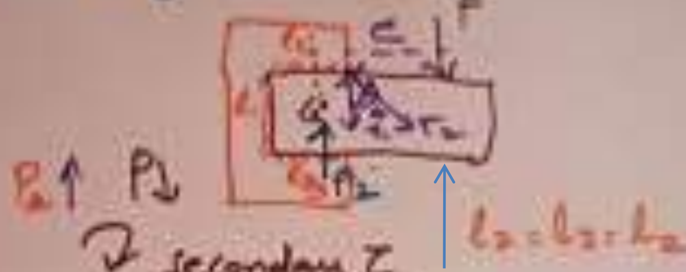
3. Excecute the failure pt "A"



4- direct shear stress

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

4. Secondary shearstress "τ_2"

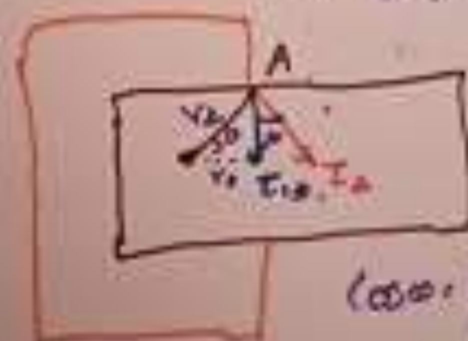


↻ secondary τ "τ_2"
which makes angle θ (acw)

$$\tau_2 = \frac{T \cdot r_2}{J}$$

J = Polar M.I

r_2 = radius between G & A.



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

$$\theta = \cos^{-1}\frac{r_1}{r_2}$$

5. Resultant τ. [: 1/[illegible] Load]

$$\tau_R = \sqrt{(\tau_1)^2 + (\tau_2)^2 \mp 2\tau_1 \cdot \tau_2 \cdot \cos\theta}$$

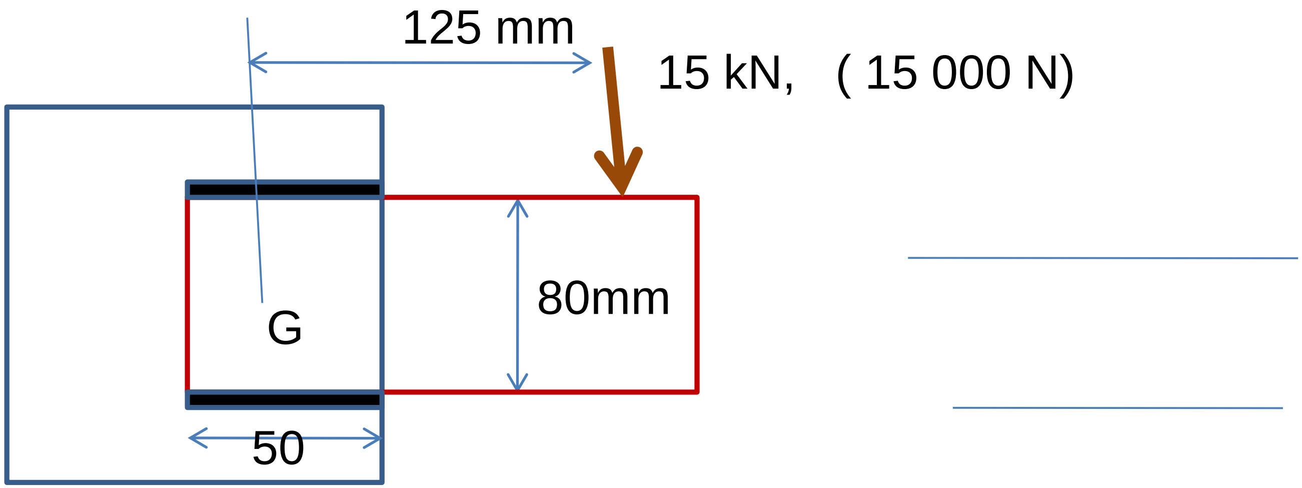
$$\cos\theta = \frac{r_1}{r_2}$$

find "h" of weld bead

$$\tau_R = [\tau].$$

WP2.A bracket carrying a load of 15 KN is to be welded as shown in figure

Find the size of the weld required if the allowable shear stress is 80 Mpa.



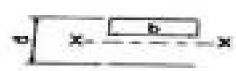
Data :

Load = 15000 N
plate width= 80 mm
e- eccentrical distance = 125 mm
[Ʈ]= 80 Mpa = 80 N/mm^2

DATA			
Load- P	15000	N	
[T]	80	N/mm^2	
width plate -d	80	mm	
eccentric. Dist.	125	mm	
length of weld- l	50	mm	
sin45	0.707		

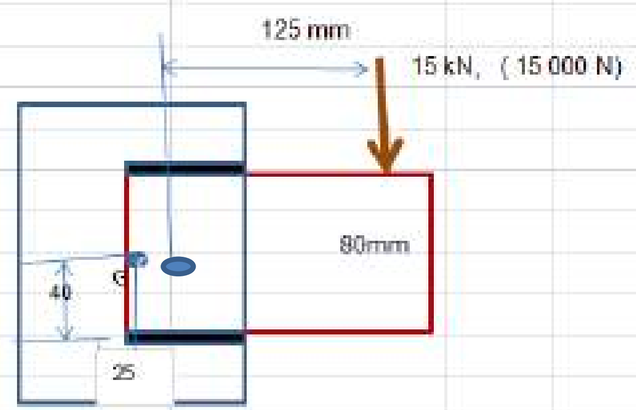
Step1 Find and locate " G" of the given weld arrangement

Refer Pg 11.5/DDB



b=	50	mm
d=	80	mm

G=?



the strucrture is symentrical about X-X

(x,y)

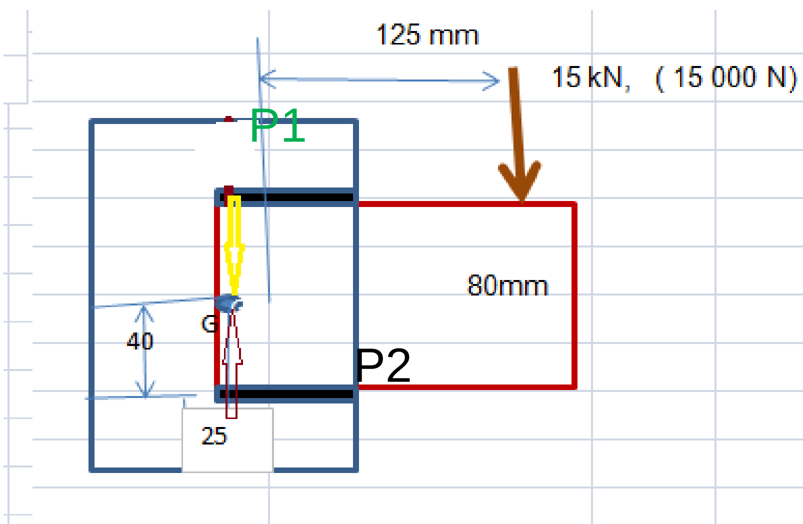
G(x,y)	x=	b/2	25
	y=	d/2	40

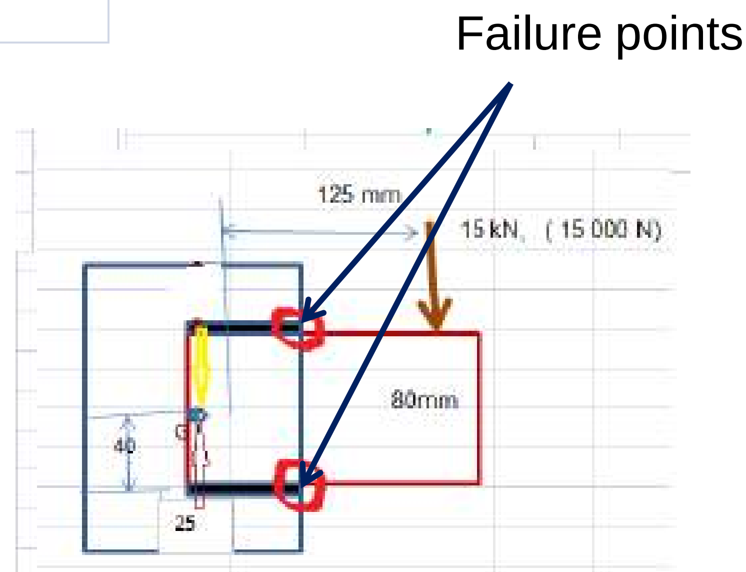
center of gravity G is (25 ,40)

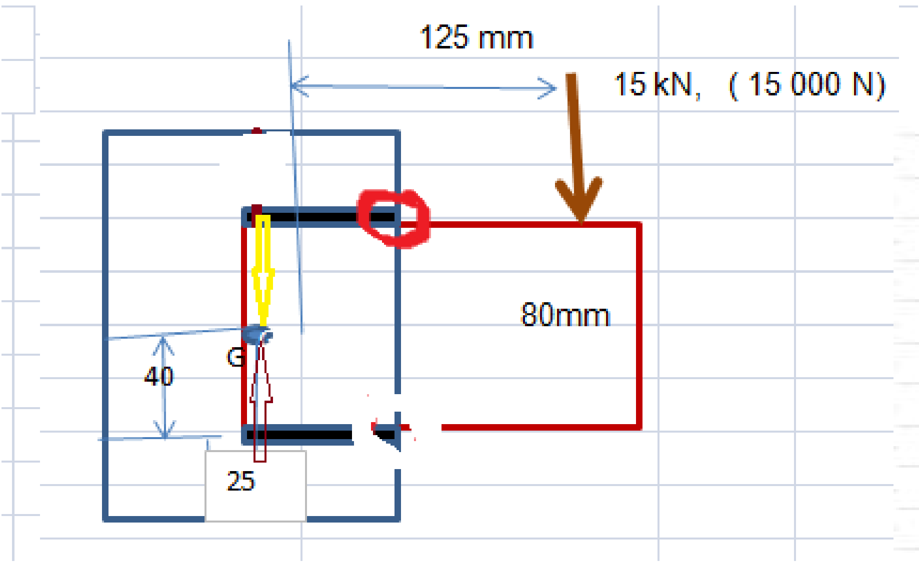
Step2: Introduce p1 & P2 and prepare the stresses induced

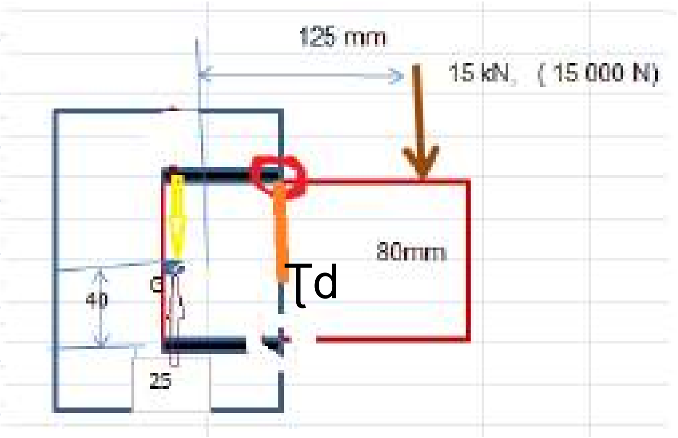
Two stresses will be 1. direct shear stress

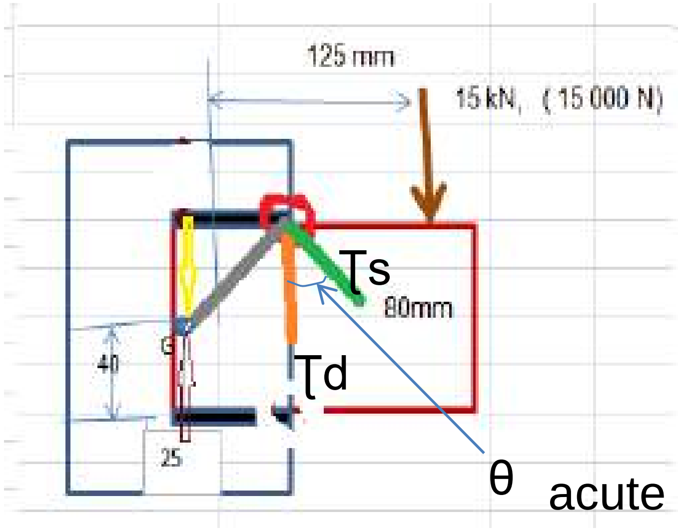
2. Secodary shear stress











Angle between the two forces
Is acute, hence Resultant is given
Parallelogram law of forces

step3. Cal. Direct stress				
stress = load /A				
Load= 15000 N				
A= lt for (single fillet)				
A=2 x l x hx sin45	(for 2 fillet)			
A=	70.7	h		
Ʈd=	load/Area			
	212.1640736	(1/h)		

Step3: cal. Of secondary shearstress "Ʈs"

Use of torsion theory refer pg 7.1 DDB

T/J= Ʈs/r

J= polar M.o.I

T= twisting moment = P x e

r= r2 from the digram

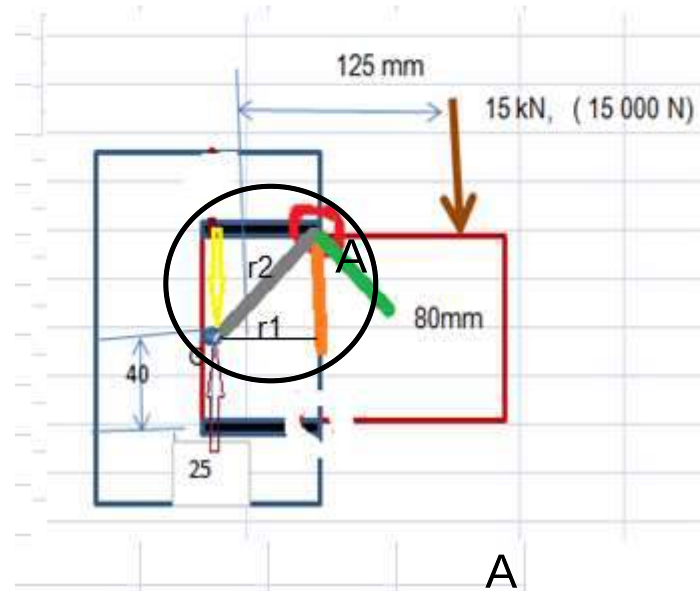
$$Ʈs = \frac{T \times r}{J}$$

From Pg. 11.5, J=?

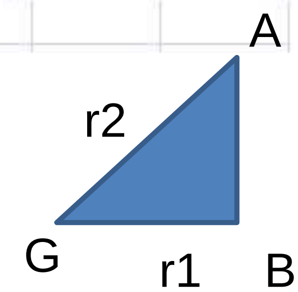
$$\left[\frac{b^3 + 3bd^2}{6}\right] \times t$$

	t=0.707h		
T=	1875000	N-mm	
J=	127849.1667	h	
r2=	47.16990566	mm	use of trignometry-Phythagras theorem

now	
Ts =	691.8 (1/h)



r1=25=GB
AB=40



$$r_2 = \sqrt{(AB)^2 + (GB)^2}$$

Step5 Cal.of " h" (Ʈr by parallegram law of forces)

ƮR	=	$\sqrt{Ʈd^2+Ʈs^2+2.Ʈd.Ʈs.\cos\theta}$		
	Ʈd^2	45014	h^2	
	Ʈs^2	5E+05	(1/h^2)	478587.24*(1/h^2)
	cosθ= r1/r2	0.532		
where	r1= 25			
	r2=47	[from triangle]		
	2.Ʈd.Ʈs.cosθ	2E+05	(1/h^2)	
ƮR	=	824.5	(1/h)	
80	=	824.5	(1/h)	
	h=	10.31	mm	11.299mm

Design procedure for closed system of welds

Step1 Find weld areas

Step2: Find direct shear stress

Step3: Find either bending or tensile stress according to the weld structure.

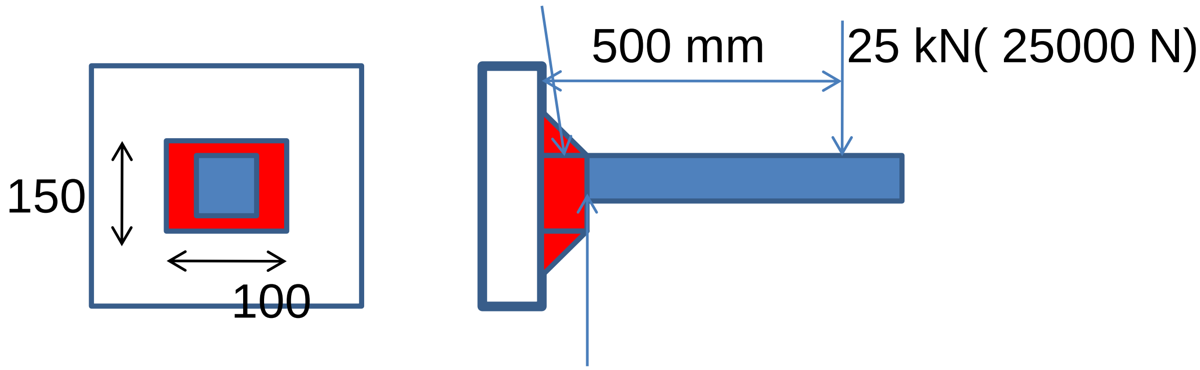
Step4: Find h

1. **apply max.normal stress theory – (When tensile failure is available)**

2 Apply max shear stress theory- (when shear stress is available)

wP3

A shaft of rectangular cross section is welded to a support by means of fillet welds as shown in figure below. Determine the size of weld, if the permissible stress in the weld is limited to 75 MPa



Data

p= 25 000 N

e= 500 mm

[Ţ]= 75 N/mm^2

h=?

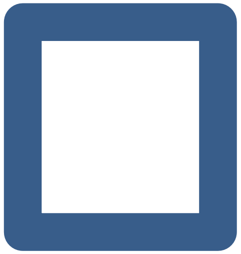
This is a closed system of weld

1. Max normal stress theory (case1)
2. max. Shear stress theory (case 2)

Since , shear stress only given work for “2” case

Step1 Total weld area

4 –sides are there
weld length   l= 2(b+d)



d= 150

b= 100

A= t x l

A= 0.707h x l
A= 0.707 h x [2 b+ 2d]

Area = 353.5 h

Step2: find direct shear stress

τ = P/A

= 25000/(353.5h)

= 70.72/h

Step 3 Find bending stress

σb= M/Z

M= p x e
= 25 000 x 500
= 125 x 10^5 N-mm

σb= 785.8/h

Now to find Z Ref. Pg.11.6/DDB

Z = [bd + (d^2/3)] x t

= [bd + (d^2/3)] x 0.707 h

Z= 15907.5h

Step 4: find h

here , as shear stress value only given

choose, max shear stress theory

$$\tau_{max} = [\tau] = 1/2\sqrt{(-\sigma b)^2 + 4.\ \tau^2}$$

Take [tensile]= [120] N/mm^2

$$[75] = 1/2\sqrt{(785.8/h)^2 + 4.\ (70.72/h)^2}$$

h= 399.2/75

h= 5.32 mm

Closed system concept

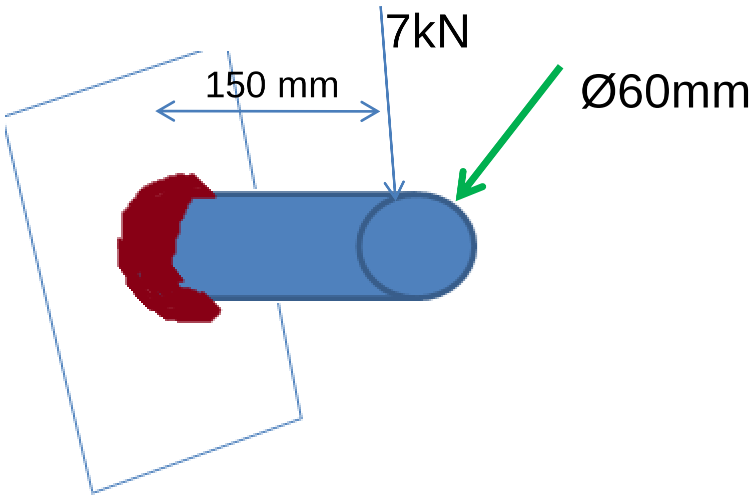
1. **Normal stress theory (if allowable tensile is given)**
2. **max shear stress theory (if allowableshear stress alone given)**

Steps

1. Find direct stress

2. Find beniding or shear stress
3. Find weld size "h"

   Apply **max shear stress theory/ Normal stress theory**



WP4

A circular shaft of dia 60 mm is welded to a support by means of a fillet weld as shown in figure. Determine the size of the weld if the permissible shear stress is limited to 85 MPa

Step1

direct stress   Shear = load/A

A= pi x d x t

= 3.141 x 0.707h

= 133.27h

P= 7000 N

Ƭd=52.52/h

Step2 bending stress

$\sigma b = M/Z$

M= 7000 x 150

= 105 x 10^4 N.mm

Z= pg.no.11.6/DDB

Z= [(pi x d^4)/4] x 0.707 h

= 1999h

$\sigma b = M/Z = 525.26/h$

Step4 :

H=3.15 mm

Unit V

Rolling contact bearing- ball bearings takes both radial and axial load

Sliding contact bearing - Journal bearings- sleeve and shaft – radial loads

Journal bearing
*** shaft and sleeve arrangement***

INTRODUCTION

In journal bearing sliding action is along the circumference of circle or an arc of circle and carrying radial loads. [1]

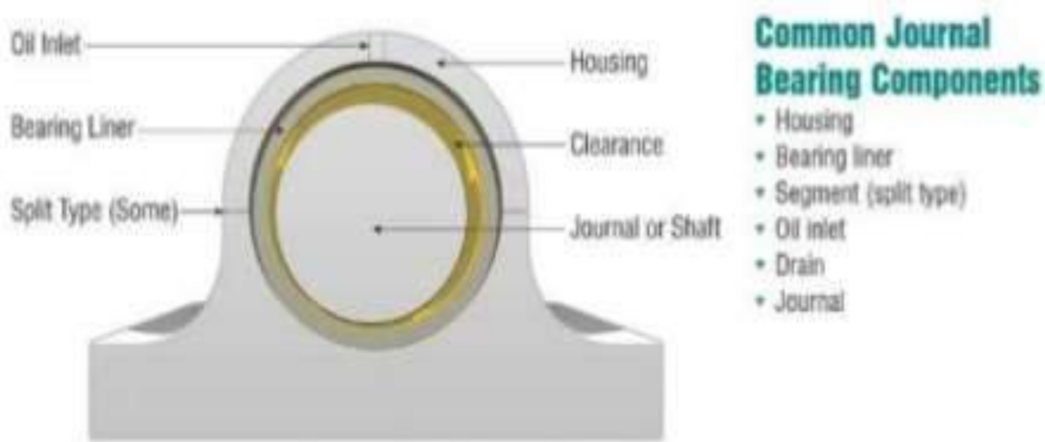
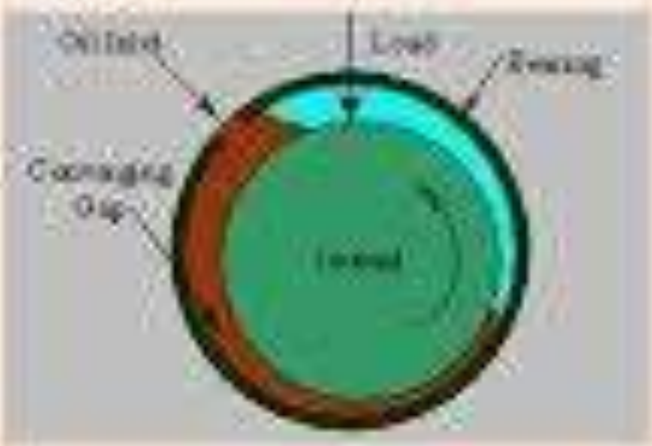
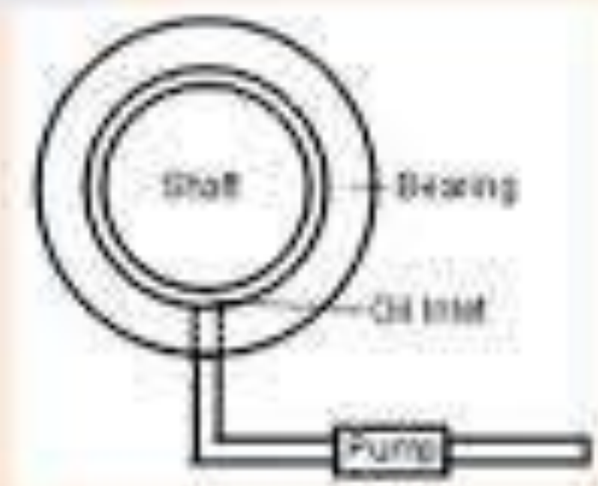


Figure 2. Plain Bearings (Journal Bearings)

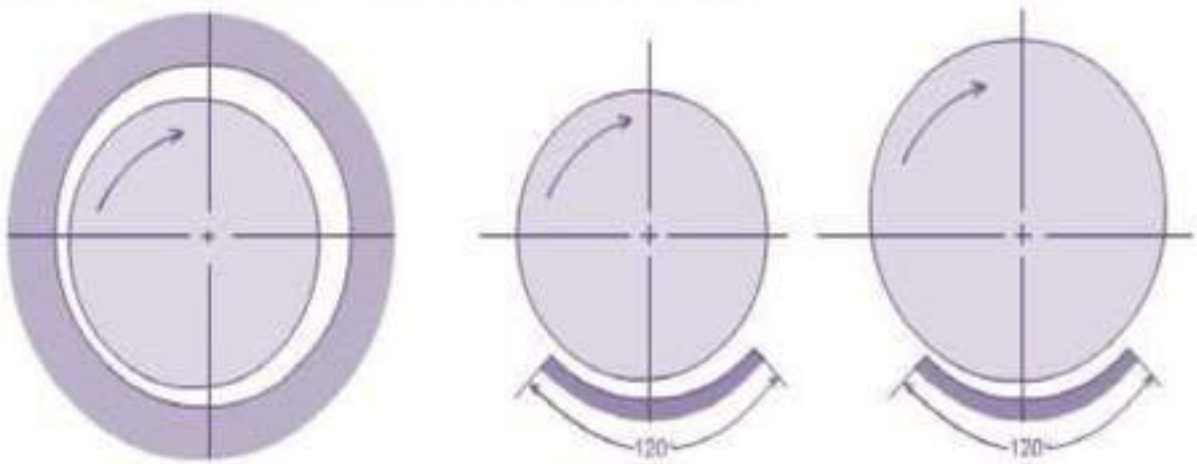
Journal bearings

- Dry
- Hydrodynamic
- Hydrostatic
- Squeeze Film





Bearing Classification



(a) Full (b) Partial (c) Fitted

When the angle of contact of the bearing with the journal is 360° as shown in (a), then the bearing is called a full journal bearing.

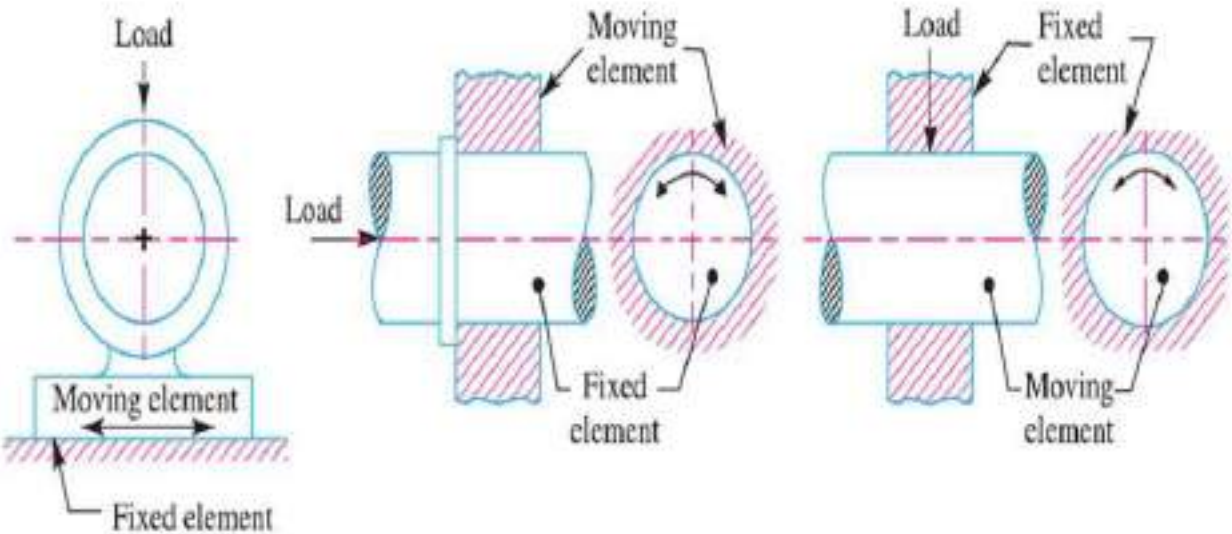
When the angle of contact of the bearing with the journal is 120°, as shown in Fig *(b), then* the bearing is said to be *partial journal bearing.*

the diameters of the journal and bearing are equal, then the bearing is called a *fitted bearing, as shown in Fig. (c).*

Classification of Bearings

1. **Depending upon the direction of load to be supported.**

   a) Radial bearings, and (b) Thrust bearings.



(*a*) Radial bearing. (*b*) Radial bearing. (*c*) Thrust bearing.

Design procedure for sliding contact bearing- Journal bearing

Basic data needed

Load= W

Journal Diameter or Length= D or L

RPM = speed

Step1: find the application chose L/D ration ref.Page N0 7.31/DDB

Step2: Find L & D of the journal bearing & List the values/propertes

Step3: Check for bearing pressure < [pressure] Pg.731

$P = W/(L x D)$

Step4: selection of SAE oil refer. Pg.No.7.31 & 7.41

Step 5: Find coefficient of friction using Mc kees eqn. Pg.No.7.34

Step6: Find heat generation Hg? Pg. 7.34

Step 7 : Find Hd heat dissipation pg. 7.34

Step8: check for Artificial cooling required or not Hg<Hd or Hg> Hd

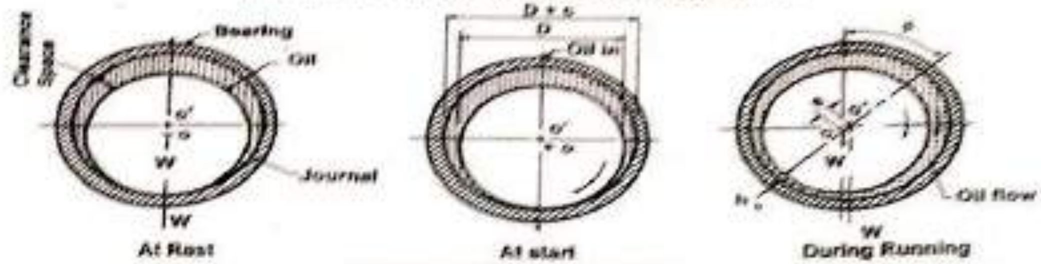
DESIGN PRACTICES – JOURNAL BEARING

MACHINERY	BEARING	L / D	BEARING PRESSURES ALLOWABLE	LUBRICANT	
			kgf / cm^2	Z	Z_n / P min
Stationary High speed steam Engines	Main	1.5 - 3.0	17.50	15	355.6
	Crank pin	0.9 - 1.5	42.00	30	85.3
	Wrist pin	1.3 - 1.7	126.00	25	71.1
Gas and Oil Engines (Four Stroke)	Main	0.6 - 2.0	49 - 84	20 - 65	284.5
	Crank pin	0.6 - 1.5	108 - 126		142.2
	Wrist pin	1.5 - 2.0	125 - 154		71.1
Gas and Oil Engines (Two Stroke)	Main	0.6 - 2.0	35 - 125	20 - 65	355.6
	Crank pin	0.6 - 1.5	70 - 105		170.7
	Wrist pin	1.5 - 2.2	84 - 125		142.2
Aircraft & Automobile Engine	Main	0.8 - 1.8	56 - 119	8	213.3
	Crank pin	0.7 - 1.4	105 - 245		142.2
	Wrist pin	1.5 - 2.2	161 - 350		113.8
Reciprocating Compressors and Pumps	Main	1.0 - 2.2	17.5	30 - 80	426.7
	Crank pin	0.9 - 1.7	42		284.5
	Wrist pin	1.5 - 2.0	70		142.2
Centrifugal Pump, Motors and Generators	Rotor	1.0 - 2.0	7 - 14	25	2844.5
Machine Tools	Main	1.0 - 4.0	21	40	14.2
Steam Turbines	Main	1.0 - 2.0	7 - 20	2-16	1422.3
Railway Cars	Axle	1.9	35	100	711.2
Marine Steam Engines	Main	0.7 - 1.5	35	30	284.5
	Crank pin	0.7 - 1.2	42	40	213.3
	Wrist pin	1.2 - 1.7	105	30	142.2
Transmissions	Light,Fixed	2.0 - 3.0	1.8	25	1422.3
Gyroscopes	Rotor	-	60	30	782.3
Shafting	Self Aligning	2.5 - 4.0	11	60	426.7
	Heavy	2.0 - 3.0	11	60	426.7
Cotton Mills	Spindle	-	0.07	2	142231
Punching and Shearing Machines	Main	1.0 - 2.0	280	100	-
	Crank Pin	1.0 - 2.0	560	100	-
Rolling Mills	Main	1.0 - 1.5	210	50	142.2

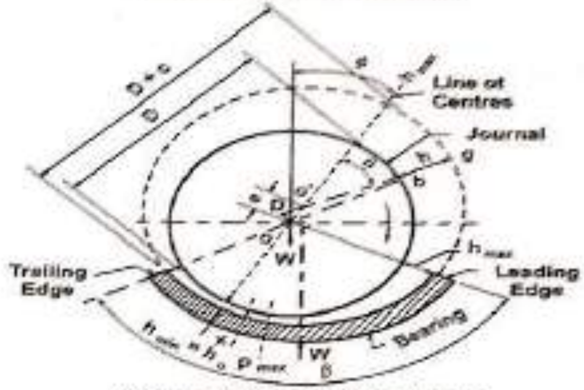
Z, absolute viscosity, centipoises   n, speed, rpm   P, pressure, kgf / cm^2

Pg.7.31

HYDRODYNAMIC JOURNAL BEARINGS



MECHANISM OF LUBRICATION

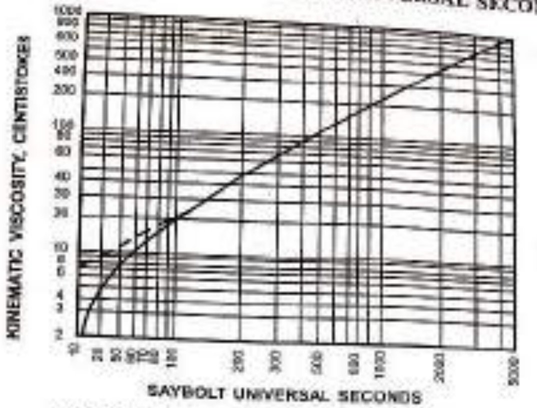


GEOMETRIC RELATION FOR ANY JOURNAL BEARING
(Shown here partial clearance bearing)

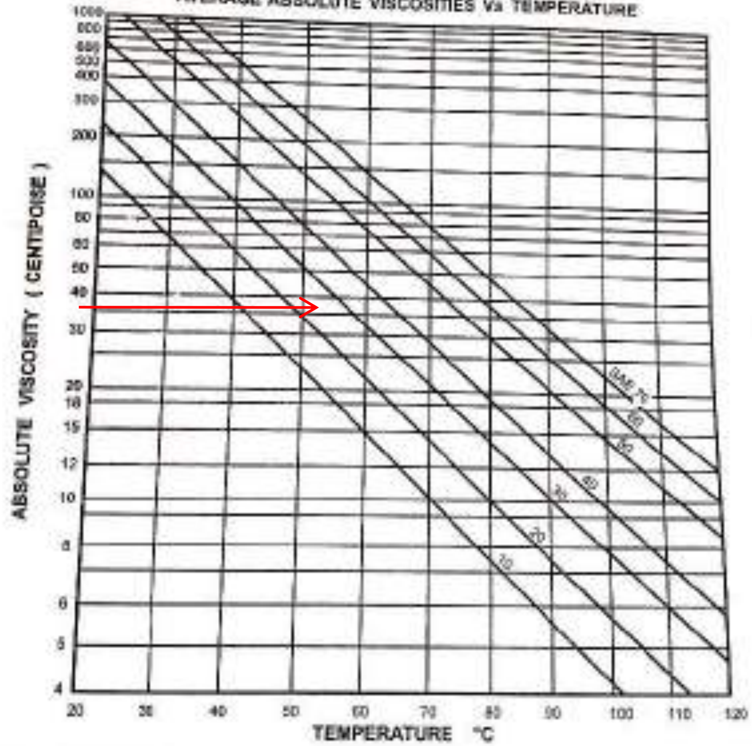
GEOMETRIC RELATIONS FOR A CLEARANCE BEARING

Clearance ratio, C / D	D journal diameter, cm
Eccentricity, $e = \frac{C}{2} - h_o$	C diametral clearance, cm
	h_o minimum film thickness, cm
Eccentricity factor or	ϵ attitude, dimension less
	ϕ attitude, angle
	L Length of bearing, cm
Attitude, $\epsilon = \frac{2e}{C} = 1 - \frac{2h_o}{C}$	W Load, kgf
	P bearing pressure on projected area, kgf / cm^2
Film thickness at any angle θ, $h = \frac{C}{2}(\epsilon \cos\theta + 1)$	e eccentricity, cm
	n speed of journal, rpm
Minimum film thickness, $h_o = \frac{C}{2}(1 - \epsilon)$	n' speed of journal, rps
	Z absolute viscosity of the oil, centipoises

KINEMATIC VISCOSITY - SAYBOLT UNIVERSAL SECONDS



AVERAGE ABSOLUTE VISCOSITIES Vs TEMPERATURE



Pg.7.41

DIMENSIONLESS PERFORMANCE PARAMETERS

Coefficient of friction Variable,	$\mu \dfrac{D}{C}$
Flow variable,	$4q / D C n' L$
Flow ratio,	q_s / q
Pressure ratio,	p / p_{max}
Temperature rise variable,	$\dfrac{\rho C' \Delta t_o}{p}$

FLOW THROUGH THE BEARING

Axial flow in a 360° bearing, pressure fed, through a cylindrical hole at the centre

$$q_{s1} = \frac{C^3 p_1}{24 Z'} \left(\tan^{-1} \frac{\pi D}{L} \right) \left(1 + 1.5\, \epsilon^2\right)$$

Flow is 2 to 3 times greater when the feed hole is located in a longitudinal groove

With the feed hole in a central circumferential groove

$$q_{s1} = \frac{\pi D C^3 p_1}{24 Z' L} \left(1 + 1.5\, \epsilon^2\right)$$

Energy increase of the oil $= q \rho\ C' \Delta t_o$

AVERAGE TEMPERATURE RISE

Oil ring bearings (still air),	$\Delta t_s = 2 \Delta t$
Oil bath bearings (still air),	$\Delta t_s = 1.3 \Delta t$
Waste packed bearings (still air),	$\Delta t_s = 2.5 \Delta t$

Design oil film temperature, 60 - 95 °C

FRICTIONAL LOSS IN THE CAP

Average film thickness in the cap

$$h_{av} = \frac{C}{2} \left(1 + 0.74\, \epsilon^2\right)$$

$$F = \frac{Z' A v}{0.6\, h_{av}}$$

$$U_f = F v$$

- Δt increase in bearing surface temperature from ambient temperature, °C
- k constant for Mckees equation
- v surface speed of Journal m/min
- K constant for heat dissipation, 437 for heavy construction, well ventilated. 775 for light construction in still air
- h_{av} average film thickness, cm
- h film thickness at any angle, θ, cm
- F frictional force on the cap, kgf
- U_f frictional loss in the cap, kgf m/min
- A sliding area of the cap, cm²
- Δt_s difference in temperature of oil and ambient temperature, °C

Sommer feld number, $S = \left(\frac{Z'n'}{P}\right)\left(\frac{D}{C}\right)^2$

$S = (2+\epsilon^2)\sqrt{1-\epsilon^2}/12\pi^2\epsilon$ for $L/D = \alpha$

υ	Kinematic viscosity centistokes
Z	absolute viscosity, kgf sec / cm^2 = $\frac{Z}{9.81 \times 10^7}$
P	hydro dynamic pressure developed at any angles θ, kgf / cm^2
p_i	inlet pressure, kgf / cm^2
p_{max}	maximum pressure, kgf / cm^2
ρ	density of the oil, kgf / cm^3
p_0	Pressure at $\theta = 0°$

Coefficient of friction

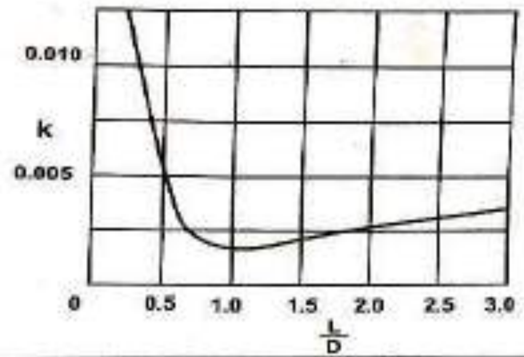
McKEES EQUATION

$$\mu = \frac{33.25}{10^{10}}\left(\frac{Z_0}{P}\right)\left(\frac{D}{C}\right) + k$$

PETROFF'S EQUATION

$$\mu = 2\pi^2\left(\frac{Z'n'}{P}\right)\left(\frac{D}{C}\right)$$

for lightly loaded bearings $S > 0.15$



$$\mu = \frac{C}{D}\,\frac{(1+2\epsilon^2)}{3\epsilon}$$

for medium or heavily loaded bearings $S < 0.15$

Reynolds' equation for pressure distribution at any angle θ

$$p - p_0 = \frac{20\, Z'\, \upsilon D}{C^2}\left[\frac{\epsilon(2+\epsilon\cos\theta)\sin\theta}{(2+\epsilon^2)(1+\epsilon\cos\theta)^2}\right]$$

Load capacity

$$W = 20 Z'\,\upsilon\, L\left(\frac{D}{C}\right)^2\left[\frac{\pi\epsilon}{(2+\epsilon^2)\sqrt{1-\epsilon^2}}\right]$$

Heat generated, $H_g = \mu W v$ kgf m / min

Heat dissipated, $H_d = \frac{(\Delta t + 18)^2 L D}{K}$ kgf m / min

S	Sommer feld number, dimensionless
μ	coefficient of friction
q	oil flow through the bearing, cm^3 / sec
q_a	axial flow of oil, cm^3 / sec
q_{pi}	axial flow of oil due to inlet pressure cm^3 / sec
C'	specific heat of oil 17100 kgf cm / kgf °C
Δt_o	temperature increase of the oil °C

JBP1

Design a full journal bearing for a railway car, running at 600 rpm. Diameter of the journal is 200 mm and load on the bearing is 230 KN.

Data

W= load= 230 000 N

D= diameter= 200 mm

Speed= n= 600 rpm

Step1.find the application chose L/D ratio ref.Page N0 7.31/DDB

Application: railway car

From Pg.7.31

chose L/D ratio, **L/D=1.9**

[pb]= 35 kgf/cm^2 ---- 35 x 10 N/cm^2----35 x 10/10^2 N/mm^2

[Pb]= [3.5] N/mm^2

Z= 100 centipoise

Zn/P= 711.2

Step2: Find L of the journal bearing

L/D=1.9

L= 1.9 x D

L= 1.9 x 200 =380 mm

Step3; Check for Pind< [Pb]

P= w/(L x D)

= 230 x 10^3/(380 x 200)

= 3.026 N/mm^2

3.026 < [3.5]

Pind < [Pb] design safe.

Step4: find SAE oil Pg. 7.41

Assume Operating temperature : 40º - 120 º if not given in problem

Assume 70º C

Z= in cp?

from step 1,

Zn/P= 711.2

Z=? 711.2 x 30.26/(600)

Z=35.87 cP.

Now use :

z= 35.87 cP, to= 70º C

SAE50 oil selected

Refer. Pg.No:7.41/ddb from chart,

To=70º C and Z= 35.87 --- 45 cP

oil selected is **SAE 50**

Step5: find μ

Refer Pg.7.34/DDB

μ = 33.25/10^10 x (zn/P) x (D/C) + k

z=45/36 cP
n= 600 rpm
Pind= 30.26 kgf/cm^2
D/C= 1000 (in general clearance ratio C/D= 1/1000)

To find k
Pg.no. 7.34/DDB
for L/D= 1.9,

k= 0.0025

μ= 0.00546

= 0.0026 for z=36cP

Step 5’ find Hg?

$Hg = \mu \times W \times v$

$\mu = 0.00546$

$W = 23000$ kgf

$V = \pi \times d \times n/1000$

$d = 200$mm, $n = 600$ rpm,

$V = 376.99$ m/min

Now

Hg= 47342.5 kgf m/min

Step6 Find Hd=?

Pg. No. 7.34/DDB

$Hd = (\Delta t+18)^2 \times L \times d/K$

$2\Delta t = \Delta ta$ Pg.No. 7.35/ddB

$\Delta t = 1/2 \times \Delta ta$

$= ½ \times (to-ta)$

$= ½ \times (70 - 30)$

$= 20$

L= 380 mm == 38 cm

D= 200 mm ==20 cm

k= 437 or 775 (pg.no: 7.35/ddB)

Assume k= 437- heavy construction

Now **Hd=2511.3 kgf m/min**

Step7

Hg > Hd

Artificial cooling arrangement is required

A3-3 Design a journal bearing for the following data

Diameter of journal=75 mm
Load on journal=3500N
Length of journal =75 mm
Speed=400 rpm
Minimum film thickness= 0.02 mm
Take operating temperature 60º C

Rolling Contact Bearing

Structure of ball bearing

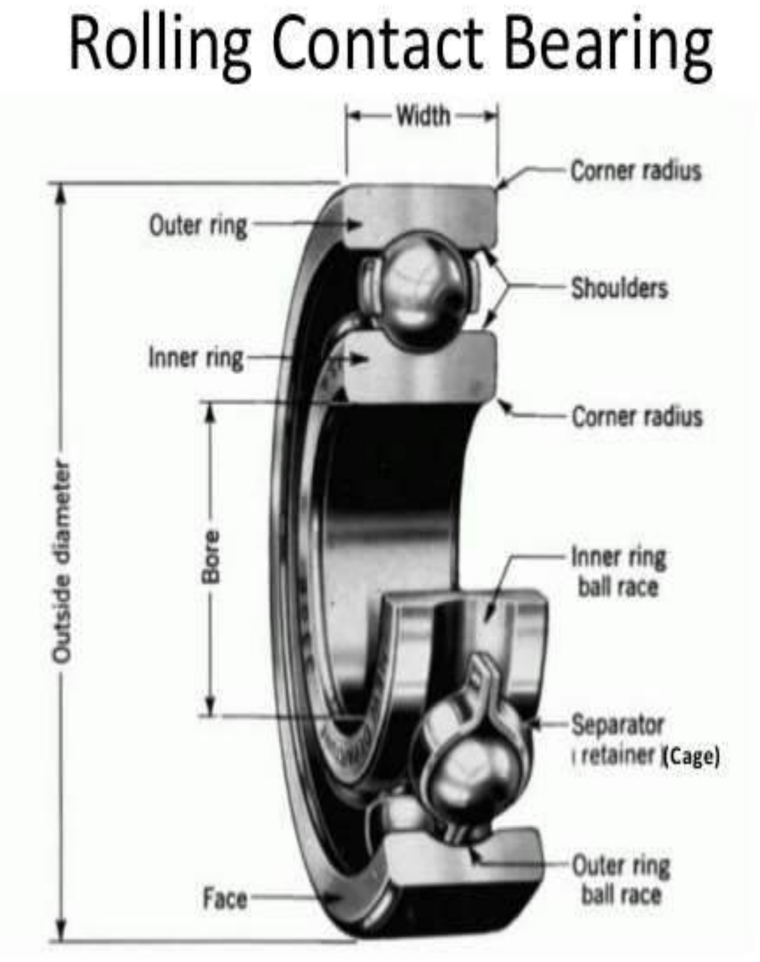
Point of contact

Also called ‘ antifriction bearing”

Loads

1. **Axial**
2. **Radial load**

Needs
Static capacity
Dynamic capacity



Advantages and Disadvantages of Rolling Contact Bearings Over Sliding Contact Bearings

Advantages

1. Low starting and running friction except at very high speeds.
2. Ability to withstand momentary shock loads.
3. Accuracy of shaft alignment.
4. Low cost of maintenance, as no lubrication is required while in service.
5. Small overall dimensions.
6. Reliability of service.
7. Easy to mount and erect.
8. Cleanliness.

Disadvantages

1. More noisy at very high speeds.
2. Low resistance to shock loading.
3. More initial cost.
4. Design of bearing housing complicated.

Types of Rolling Contact Bearings

Following are the two types of rolling contact bearings:

1. Ball bearings; and 2. Roller bearings.

Why need of rolling contact bearings

- Available in many sizes and cross sections
- Minimal lubricant supply needed
- Low driving torque required
- High load carrying capacity
- Accurate positional capability
- Wide temperature operating range (with solid lubricants)
- Wide speed range capability
- Many analytical programs available

95TR29

DDB Pgs

4.1
4.2
4.4
4.8
4.9

Selection of bearings
4.12 to 4.36

DESIGN PROCEDURE

1 FIND Fa/Fr=e, select factors X & Y Pg.4.14/DDB

2 Find equivalent load P Pg.4.2/DDB

3 Find dynamic capacity “ C” of Bearing Pg.4.2/DDB

4 Select suitable rolling contact bearing based on “C”

state bearing designation

5. Rated life of the bearing at 90%, 95% or given percentage