UNIT-IV

			BL	CO	Marks
8	a)	Write the differences between self-energizing and self-locking brakes.	L2	CO4	5 M
	b)	Discuss the factors affecting the selection of material for clutch and brake linings.	L2	CO4	5 M
		OR			
9		Design a closed coil helical spring for a boiler safety valve which is required to blow off steam at pressure of 1.5 MPa. The diameter of the valve is 50 mm. The initial compression of the spring is 40 mm and the lift is limited to 20 mm. The maximum shear stress in the material of the wire is limited to 500 MPa. The modulus of rigidity for the spring material is 80 kN/mm^2.	L4	CO4	10 M

UNIT-V

		BL	CO	Marks
10	Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 rpm for an average life of 5 years at 10 hours per day. Assume uniform and steady load.	L4	CO5	10 M
	OR			
11	Design a gear drive to transmit 22 kW at 1000 rpm. Speed reduction is 2.5. The center distance between the shafts is 350 mm. The materials are pinion-C45; Gear wheel CI Grade 30. Design the drive using Lewis and Buckingham equations.	L4	CO5	10 M

Code: 23ME3503

III B.Tech - I Semester - Regular Examinations - NOVEMBER 2025

DESIGN OF MACHINE ELEMENTS
(MECHANICAL ENGINEERING)

Duration: 3 hours Max. Marks: 70

Note: 1. This question paper contains two Parts A and B.
2. Part-A contains 10 short answer questions. Each Question carries 2 Marks.
3. Part-B contains 5 essay questions with an internal choice from each unit. Each Question carries 10 marks.
4. All parts of Question paper must be answered in one place.

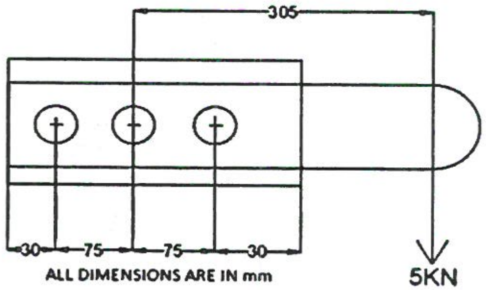
BL – Blooms Level CO – Course Outcome

**** Design Data Book is Allowed ****

PART – A

		BL	CO
1.a)	What is factor of safety? Why is it necessary?	L2	CO1
1.b)	Differentiate endurance strength and endurance limit.	L2	CO1
1.c)	What are the advantages of welded joints?	L2	CO2
1.d)	Define Eccentric loading in bolted joints.	L1	CO2
1.e)	What is equivalent bending moment and equivalent torque?	L1	CO3
1.f)	Define Rigid couplings.	L1	CO3
1.g)	Write load deflection equation for helical spring.	L2	CO4
1.h)	State any two functions of Clutch.	L2	CO4
1.i)	State the law of gearing.	L1	CO5
1.j)	What is full journal bearing?	L1	CO5

PART – B

			BL	CO	Max. Marks
UNIT-I					
2		Discuss about the following theories of failure. (i) Maximum Principal stress theory. (ii) Maximum shear stress theory. (iii) Distortion Energy theory.	L2	CO1	10 M
OR					
3		Discuss about the basic Procedure of Machine Design.	L2	CO1	10 M
UNIT-II					
4		A steel plate is subjected to a force of 5 kN and fixed to a column by means of three identical bolts as shown in figure. The bolts are made of plain carbon steel of yield strength 380 N/mm^2 and FOS is 3. Specify the size of the bolt. 	L3	CO2	10 M
OR					
5	a)	Sketch any three basic types of welded joints.	L2	CO2	3 M

	b)	A plate of 150 mm wide and 10 mm thick is to be welded with another plate by means of transverse welds at the ends. The plates are subjected to a load of 90 kN, find the size of the weld for static load. The permissible tensile stress should not exceed 70 MPa.	L3	CO2	7 M
UNIT-III					
6		The shaft of an axial flow rotary compressor is subjected to a maximum torque of 2 kN-m and maximum bending moment of a 4 kN-m. The combined shear and fatigue factors in torsion and bending may be taken as 1.5 and 2.0 respectively. Determine the diameter of the shaft, the shear stress in shaft should not exceed 50 MN/m^2. Design a hallow shaft for the above compressor taking the ratio of inner diameter to outer diameter as 0.5. Calculate the percentage saving in material.	L3	CO3	10 M
OR					
7		Design a muff coupling to connect two shafts transmitting 40 kW at 120 r.p.m. The permissible shear and crushing stress for the shaft and key material (mild steel) are 30 MPa and 80 MPa respectively. The material of muff is cast iron with permissible shear stress of 15 MPa. Assume that the maximum torque transmitted is 25 percent greater than the mean torque.	L4	CO3	10 M

III B.Tech – I Sem – Regular Examinations – Nov 2025
Design of Machine Elements (23ME3503)

Scheme of evaluation

PART – A

1. a) Definition or Formula of factor of safety – 1m
Necessity of factor of safety – 1m

b) endurance strength – 1m
endurance limit – 1m

c) Any two advantages of welded joints – 2m

d) Definition or diagramatic representation of electric loading in bolted joints – 2m

e) equivalent bending moment (definition or formula) – 1m
equivalent twisting moment (definition or formula) – 1m

f) Definition or example of rigid coupling – 2m

g) Load deflection equation of spring – 2m

h) Function of a clutch – 2m

i) law of gearing statement – 2m

j) Definition or diagram of full journal bearing – 2m

PART – B

Unit – I

2. Formulae of given theories of failures – 4m
explanation of given theories of failures – 6M

(OR)

3. Flow chart of design procedure – 5m
explanation of ~~all~~ phases of design – 5m

Unit – II

4. Given data – 2m
primary shear ~~stress~~ – 3m

Secondary shear ~~force~~ – 3 m
Resultant shear force – 2 m
Size of the bolt – 1 m

(OR)

5. a) sketches of any three welded joints – 3 m

b) Given data – 3 m
length of the weld – 4 m

Unit – III

6. Given data – 2 m
Design of Solid shaft – 3 m
Design of hollow shaft – 3 m
percentage saving of material – 2 m

(OR)

7. Given data – 2 m
Design of shaft – 3 m
Design of muff – 3 m
Design of Key – 2 m

Unit – IV

8. a) Definition of Self-energizing brake – 2½ m
Definition of self-locking brake – 2½ m

b) Any 3 factors affecting material selection – 5 m

(OR)

9. Given data – 2 m (spring index may be as per student choice)
Wire diameter of Spring – 3 m
Mean diameter of coil – 2 m
Number of coils – 2 m
pitch of the Spring – 1 m

Unit – V

10. Given data – 2m
    Basic dynamic equivalent load – 4m
    Basic dynamic load rating – 4m

(OR)

11. Given data – 2m
    Material properties – 2m
    module of gear tooth – 3m
    Dynamic load, endurance strength and wear load – 3m

III B.Tech – I Sem – Regular Examinations – Nov 2025
Design of Machine Elements (23ME3503)

KEY

PART – A

1. a) The ratio of maximum stress to working stress is known as factor of safety.

$$\text{Factor of Safety} = \frac{\text{maximum stress}}{\text{working stress}}$$

When factor of safety is large, the probability of failure is reduced.

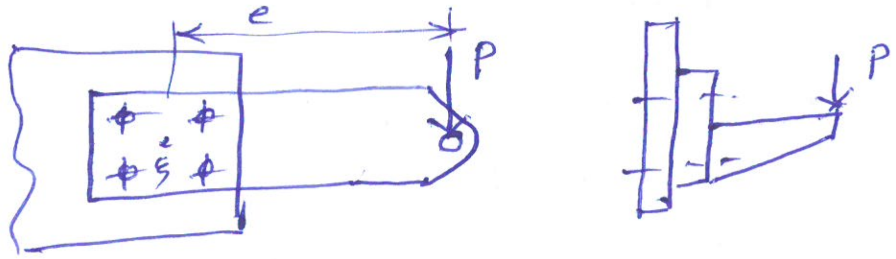
b) The endurance strength is the maximum completely reversed stress under which a material will fail after it has experienced the stress for a specified number of cycles.

The endurance limit is defined as the maximum value of completely reversed stress that the standard specimen can sustain for an unlimited number of cycles without fatigue failure

c) Advantages of welded joints:

> lighter in weight
> maximum efficiency
> smooth in appearance
> greater strength
> provide rigid joint
> welding process takes less time

d) Eccentric loading in bolted joints: occurs when an applied load is not centred, causing a combination of direct and indirect forces.



e) Equivalent Bending moment: It is defined as the bending moment, which when acting alone, will produce the same bending stress in the shaft as under the combined action of bending moment (M) and torsional moment (T) under fluctuating loads.

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]$$

Equivalent Torque: It is defined as the ~~bending~~ torsional moment, which when acting alone, will produce the same torsional shear stress in the shaft as under the combined action of bending moment (M) and torsional moment (T) under fluctuating loads.

$$T_e = \sqrt{M^2 + T^2}$$

f) Rigid coupling cannot tolerate misalignment between the axes of the shafts. It can be used only when there is precise alignment between two shafts.

g) load deflection equation for helical spring

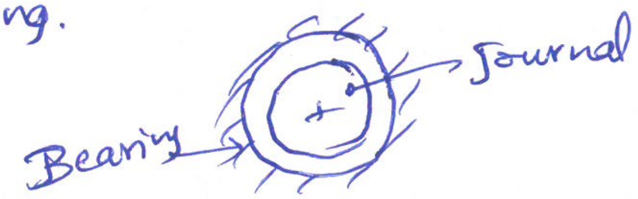
$$y = \frac{8 P c^3 n}{G d}$$

y = deflection; P = load; c = Spring index;
n = no. of coils; G = Modulus of rigidity;
d = diameter of wire

h) clutch is used to keep the driving and driven member moving together.

i) Law of gearing: The common normal to the tooth profiles of two meshing gears must pass through the fixed point called pitch point, which lies on the line of centres.

j) Full journal bearing: when the angle of contact of the bearing with the journal is 360°, then the bearing is called a full journal bearing.



PART - B

Unit - I

2. i) Maximum principal stress theory:-

The theory states that " the failure of the mechanical component subjected to biaxial stresses occurs when the maximum normal stress reaches the limiting strength of the material in a simple tension test".

$$\sigma_1 = \frac{\sigma_y}{F.S}$$

ii) Maximum shear stress theory:-

According to this theory, "failure of the member takes place when the maximum shear stress in a biaxial stress system equals to the shear stress produced during simple tension test".

$$\tau_{max} = \frac{\tau_y}{F.S}$$

iii) Distortion energy theory:-

According to this theory, " failure of the member takes place if the shear strain energy produced due to complex loading equals to the shear strain energy in simple tension test".

$$\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2 = \left(\frac{\sigma_y}{F.S}\right)^2$$

(OR)

3. Basic procedure of Machine Design.

Market Survey

↓

Define Specifications of product

↓

Study alternative mechanisms for product and select proper mechanism

↓

prepare general layout of Configuration and select joining methods between individual components of product

↓

Design individual Components

↓

prepare assembly and detail drawings and modify drawings after testing prototype model

· Explanation about each phase.

Unit – II

4. Given $P = 5000\,N$; $\sigma_y = 380\ N/mm^2$; $F.S = 3$

permissible shear stress, $\tau = \dfrac{\sigma_y}{2 \times F.S} = \dfrac{380}{2 \times 3} = 63.3\ N/mm^2$

primary shear forces,

$$P_1' = P_2' = P_3' = \frac{P}{n} = \frac{5000}{3} = 1666.67\ N$$

Secondary shear forces,

$$P_1'' = P_3'' = \frac{(Pe)(r_1)}{(r_1^2 + r_3^2)}$$

$$= \frac{(5000 \times 305)(75)}{(75^2 + 75^2)}$$

$$= 10166.67\ N$$

Resultant shear force,

$$P_3 = P_3' + P_3'' = 1666.67 + 10166.67$$

$$= 11833.34\ N$$

Size of the bolts,

$$\tau = \frac{P_3}{A} \Rightarrow 63.33 = \frac{11833.34}{\frac{\pi}{4} d_c^2}$$

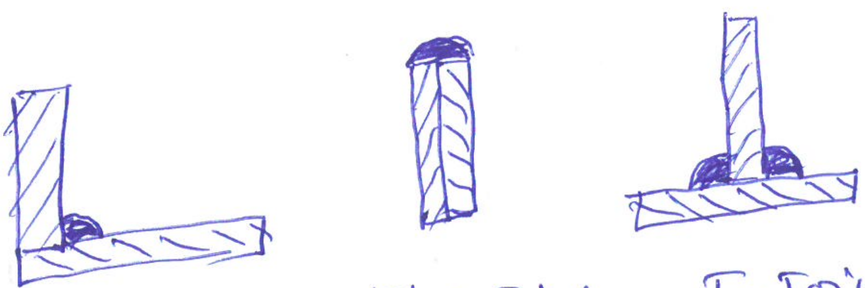
$$\Rightarrow d_c = 15.42\ mm$$

$$\therefore d = \frac{d_c}{0.8} = 19.28 \quad \text{or} \quad 20\ mm.$$

(OR)

5. a) Basic types of welded joints

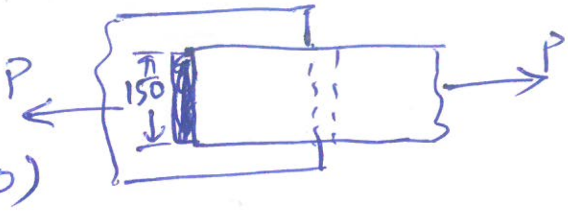
Lap Joint   Butt Joint



Corner Joint   Edge Joint   T-Joint

b) Given: $b = 150\,mm$; $s = 10\,mm$; $P = 90000\,N$

$\sigma = 70\,N/mm^2$



$P = 1.414\, s\, l\, \sigma$

$90000 = 1.414 (10)\, l\, (70)$

$\Rightarrow l = 90.9\,mm$

Adding 12.5 mm for start and stop of weld run,

$l = 90.9 + 12.5 = 103.4\,mm$

Unit - III

6. Given: $T = 2\,KN\text{-}m$; $M = 4\,KN\text{-}m$; $K_m = 2.0$;

$K_t = 1.5$; $\tau = 50\,MN/m^2$

Solid shaft

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

$= \sqrt{(2 \times 4)^2 + (1.5 \times 2)^2} = 8.54\,KN\text{-}m$

$= 8.54 \times 10^6\,N\text{-}mm$

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

$8.54 \times 10^6 = \frac{\pi}{16} \times 50 \times d^3 \Rightarrow d = 95.5\,mm$

Hollow shaft

Given $K = \frac{d_i}{d_o} = 0.5$

$T_e = \frac{\pi}{16} \tau d_o^3 (1 - K^4) \Rightarrow d_o = 97.5\,mm$

$\Rightarrow d_i = 48.75\,mm$

percentage saving of material $= \frac{Area_{(solid)} - Area_{(hollow)}}{Area_{(solid)}}$

$= 16.73\%$

(OR)

7. Given: $P = 40\ KW = 40 \times 10^3 W$; $N = 120\ rpm$;

$\tau_s = 30\ MPa$; $\sigma_c = 80\ MPa$; $\tau_c = 15\ MPa$

Design of shaft:

$$T = \frac{P \times 60}{2\pi N} = \frac{40 \times 10^3 \times 60}{2\pi \times 120} = 3183.1\ N\text{-}m$$

$$= 3183.1 \times 10^3\ N\text{-}mm$$

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

$$3183.1 \times 10^3 = \frac{\pi}{16} \times 30 \times d^3 \Rightarrow d = 81.5\ mm$$

$$\simeq 85\ mm$$

Design of muff:

outer diameter, $D = 2d + 13 = (2 \times 85) + 13$

$= 183\ mm$

$\simeq 185\ mm$

Length of the muff, $L = 3.5d = 297.5\ mm$

$\simeq 300\ mm$

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

$$3183.1 \times 10^3 = \frac{\pi}{16} \times \tau_c \times \left(\frac{185 - 85}{185}\right)$$

$$\Rightarrow \tau_c = 2.68\ N/mm^2$$ ($< 15\ N/mm^2$, design is safe)

Design of Key:

From data book, for $d = 85\ mm$; $w = 25\ mm$

Since σ_c is twice the τ value, square key may be used

$\therefore t = w = 25\ mm$

length of the key, $l = \frac{L}{2} = \frac{300}{2} = 150\ mm$

for shearing of the key,

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

$$3183.1 \times 10^3 = 150 \times 25 \times \tau_s \times \frac{85}{2}$$

$$\Rightarrow \tau_s = 20 \text{ N/mm}^2 \quad (< 30 \text{ N/mm}^2\text{, design is safe})$$

For crushing of the key,

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

$$3183.1 \times 10^3 = 150 \times \frac{25}{2} \times \sigma_c \times \frac{85}{2}$$

$$\Rightarrow \sigma_c = 40 \text{ N/mm}^2 \quad (< 60 \text{ N/mm}^2\text{, design is safe})$$

Unit - IV

8. a) Self-energizing Brake - The frictional force helps to apply the brake along with the external force. Such types of brakes are said to be self-energizing brakes

Self-locking Brake - when the frictional force is great enough to apply the brake without any external force, then the brakes are said to be self-locking brakes.

b) The factors affecting the selection of material for clutch and brake linings

i) It should have high coefficient of friction
ii) The coefficient of friction should be constant over the entire range of temperatures in operation
iii) It should have high thermal conductivity.
iv) It should have high wear resistance
v) It should remain unaffected by environmental conditions

9. Given: $p = 1.5\ N/mm^2$ (OR) ; $D_{valve} = 50\,mm$; $y_1 = 40\,mm$;
$y_2 = 20\,mm$; $\tau = 500\,N/mm^2$; $G = 80\times10^5\,N/mm^2$

Load, P = Pressure × Area

$$= 1.5 \times \frac{\pi}{4} \times 50^2 = 2945\,N$$

Deflection of the spring, $y = y_1 + y_2 = 40 + 20 = 60\,mm$

Maximum load on the spring, $P_{max} = \frac{2945}{40} \times 60$

$$= 4417.8\,N$$

Diameter of the spring wire:-

Wahl's stress factor, $k = \frac{4C-1}{4C-4} + \frac{0.615}{C}$

[Take $C=5$] $= \frac{4(5)-1}{4(5)-4} + \frac{0.615}{5} = 1.31$

$$\tau = \frac{8 P_{max} C}{\pi d^2} \times K \Rightarrow 500 = \frac{8 \times 4417.8 \times 5}{\pi \times d^2} \times 1.31$$

$$\Rightarrow d = \underline{12.14\,mm}$$

Diameter of the Coil:-

$$C = \frac{D}{d} \Rightarrow 5 = \frac{D}{12.14} \Rightarrow D = \underline{60.7\,mm}$$

Number of coils:-

$$y = \frac{8 P_{max} C^3 \cdot n}{G d} \Rightarrow 60 = \frac{8 \times 4417.8 \times (5)^3 \times n}{80 \times 10^3 \times 12.14}$$

$$\Rightarrow n = 13.19 \simeq \underline{14}$$

Total No. of coils, $n' = 14 + 2 = 16$

Pitch of the spring:- $= \frac{\text{Free length}}{n'-1} = \frac{319.96}{14-1} = \underline{24.6\,mm}$

Free length, $l_f = 1.5y + y + n'.d$
$= 1.5(60) + 60 + 14(12.14) = 319.96\,mm$

Unit - IV

10. Given: $F_r = 4000N$; $F_a = 5000N$; $N = 1600 rpm$

Basic dynamic equivalent load,

$$P = (V \times F_r + Y F_a) S$$

$$= [(1 \times 0.56 \times 4000) + (1 \times 5000)] \times 1$$

$$= 7240 N$$

$\frac{F_a}{F_r} = \frac{5000}{4000} = 1.25$ $(> e)$

$\frac{F_a}{C_o} = 0.5$ (Assume

$\Rightarrow X = 0.56$
$Y = 1$

Also $V = 1$, $S = 1$

$K = 3$, for ball bearing

$L_H = 5 \times 10 \times 300 = 15000 h$

$L = 60 N \times L_H$
$= 1440 \times 10^6$ rev

Basic dynamic load Rating,

$$C = P\left(\frac{L}{10^6}\right)^{1/K}$$

$$= 7240\left(\frac{1440 \times 10^6}{10^6}\right)^{1/3}$$

$$= 81760 N$$

From data Book, SKF bearing 6315 is selected

$C_o = 71100 N$, $C = 89280 N$

$$\therefore \frac{F_a}{C_o} = \frac{5000}{71100} = 0.07.$$

for which $X = 0.56$ and $Y = 1.6$

So, $P = [(1 \times 0.56 \times 4000) + (1.6 \times 5000)] \times 1$

$$= 10240 N$$

and $C = 10240\left[\frac{1440 \times 10^6}{10^6}\right]^{1/3}$

$$= 115635 N.$$

(OR)

4. $P = 22.5 kW = 22500 W$; $N = 1000 rpm$; $a = 350 mm$

$V.R = i = 2.5$

$$a = \frac{D_P}{2} + \frac{D_G}{2} = \frac{D_P}{2} + \frac{2.5D_P}{2}$$

$$\Rightarrow 350 = \frac{3.5 D_P}{2} \Rightarrow D_P = 200 mm$$

$$\Rightarrow D_G = 2.5 \times 200 = 500 mm$$

Material properties	σ_y	σ_e	BHN
pinion - C45	330	270	175-215
gear (C.I) - grade 30	350	160	207-241

Since the strength of gear material is less, the design is based upon the gear.

$$v = \frac{\pi D_G N_G}{60} = \frac{\pi \times 500 \times 400}{60 \times 1000} = 10.47 m/s$$

$$C_v = \frac{3}{3+V} = 0.223$$

$$y_G = 0.54 - \frac{0.912}{D_G} = 0.154 - 0.002m \qquad \because m = \frac{D_G}{T_G}$$

$$F_t = \frac{P}{V} \times C_s \qquad C_s = 1$$

$$= \frac{22500}{10.47} \times 1 = 2149 N$$

$$F_t = \sigma_{bG}\, b\, \pi m\, y_G$$

$$\Rightarrow 2149 = \left(\frac{350}{3} \times 0.223\right)(10m)(\pi m)(0.154 - 0.002m)$$

$$\Rightarrow m = 6. mm$$

Face width, $b = 10m = 60 mm$

$$T_G = \frac{D_G}{m} = \frac{500}{6} = 84$$

$$T_P = \frac{D_P}{m} = \frac{200}{6} = 34$$

Checking the gears for dynamic and wear load

$$F_D = F_t + \frac{21v(bC + F_t)}{21v + \sqrt{(bC + F_t)}} \qquad \text{Take } C = 80$$

$$= 2149 + \frac{21(10.47)(60\times80 + 2149)}{21(10.47) + \sqrt{60\times80 + 2149}}$$

$$= 2149 + 503.86$$

$$= 2652.86 \text{ N}$$

Form factor of tooth for gear,

$$y_G = 0.154 - 0.002(6) = 0.142$$

Endurance strength of the tooth

$$F_S = \sigma_e \, b \, \pi m \, y$$

$$= 160 \times 60 \times \pi \times 6 \times 0.142$$

$$= 25695.7 \text{ N}$$

Ratio factor, $Q = \frac{2 \times V.R}{V.R + 1} = \frac{2 \times 2.5}{2.5 + 1} = 1.428$

Maximum limiting load for wear,

$$F_W = D_G \cdot b \cdot Q \cdot K = 500 \times 60 \times 1.428 \times 1.4$$

$$= 59976 \text{ N}$$

Since F_S and F_W are greater than F_D, the design is safe