Swami Keshvanand Institute of Technology, Management & Gramothan,
Ramnagaria, Jagatpura, Jaipur-302017, INDIA
Approved by AICTE, Ministry of HRD, Government of India
Recognized by UGC under Section 2(f) of the UGC Act, 1956
Tel. : +91-0141- 5160400 Fax: +91-0141-2759555
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A

Course File

On

Design of Machine Elements-I: 5ME4-04

Programme: B.Tech. Mechanical Engineering

Semester: V

Session: 2022-2023

Dr. Prem Singh
Associate Professor
Mechanical Engineering Department

Contents

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Institute Vision/Mission/Quality Policy

Institute Vision

"To promote higher learning in advanced technology and industrial research to make our country a global player."

Mission

"To promote quality education, training and research in the field of Engineering by establishing effective interface with industry and to encourage faculty to undertake industry sponsored projects for students. "

Quality Policy

We are committed to 'achievement of quality' as an integral part of our institutional policy by continuous self-evaluation and striving to improve ourselves.

Institute would pursue quality in

- All its endeavours like admissions, teaching- learning processes, examinations, extra and co-curricular activities, industry institution interaction, research & development, continuing education, and consultancy.
- Functional areas like teaching departments, Training & Placement Cell, library, administrative office, accounts office, hostels, canteen, security services, transport, maintenance section and all other services."

Departmental Vision/Mission

Departmental Vision

Our Vision is to be a:

" To become a nationally visible mechanical engineering department with excellence in teaching-learning, research and development, entrepreneurship and industry outreach activities.

."

Departmental Mission

Our Mission is:

M1. To provide facilities and environment conducive to high quality education and research and development in the field of mechanical engineering.

M2. To inculcate technical, professional and communication skills in students, staff and faculty members.

M3. To instill innovative skills, critical thinking, leadership & teamwork in students through various teaching-learning activities and industry linkages.

M4. To inculcate strong ethical qualities in the students and faculty for realizing lifelong learning and serving the society and nation at large.

"

RTU Scheme & Syllabus

5ME4-04: Design of Machine Elements – I

3rd Year - V Semester: B.Tech.: Mechanical Engineering

Credit: 3	Max. Marks: 150 (IA:30, ETE:120)
3L+0T+0P	End Term Exam: 3 Hours

S.No.	Contents	Hours
1	Introduction: Objective, scope and outcome of the course.	1
2	Materials: Mechanical Properties and IS coding of various materials, Selection of material from properties and economic aspects.	3
	Manufacturing Considerations in Design: Standardization, Interchangeability, limits, fits tolerances and surface roughness, BIS codes, Design consideration for cast, forged and machined parts. Design for assembly.	4
3	Design for Strength: Modes of failure, Strength and Stiffness considerations, Allowable stresses, factor of safety, Stress concentration: causes and mitigation, fatigue failures.	4
	Design of Members subjected to direct stress: pin, cotter and keyed joints.	5
4	Design of Members in Bending: Beams, levers and laminated springs. Design for stiffness of beam: Use of maximum deflection formula for various end conditions for beam design.	7
5	Design of Members in Torsion Shaft and Keys: Design for strength, rigidity. Solid and hollow shafts. Shafts under combined loading. Sunk keys.	5
	Couplings: Design of muff coupling, flanged couplings: rigid and flexible.	3
6	Design of Threaded fasteners: Bolt of uniform strength, Preloading of bolts: Effect of initial tension and applied loads, Eccentric loading.	4
	Power screws like lead screw, screw jack.	2
	Design of members which are curved like crane hook, body of C-clamp, machine frame etc.	3

Prerequisite of Course

Students should have the knowledge of following subjects

1. Engineering mechanics and mechanics of solids
2. Material science and engineering
3. Machine drawing practice
4. Theory of machines

List of Text and Reference Books

Text Books

S. No.	Title of Book	Author	Publisher
1.	Design of Machine Elements	V.B. Bhandari	Tata Mcgraw Hill
2.	Design Data Hand Book	Mahadevan and Reddy	CBS

Reference Books

S. No.	Title of Book	Author	Publisher
1.	Machine Design	Robert L. Norton	Pearson
2.	Fundamentals of Machine Component Design	Juvinall and Marshek	Wily

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Time Table (2022-23)

Day Time	8:00-9:00	9:00-10:00	10:00-11:00		11:30-12:30	12:30-1:30	1:30-2:30
MON	FEA Lab VII-A2 (Cad Lab)			Break			
TUE	MDP-I V-B2 (2T1)						
WED		DME-1 V-B			FEA Lab VII-A1 (Cad Lab)		
THU	MDP-I V-B1 (2T1)						
FRI						DME-1 V-B	
SAT	DME-1 V-B						

Syllabus Deployment: Course Plan & Coverage

Lect.	TOPICS SUB-TOPICS	Methods and Media
	UNIT 1: Introduction	
L1	Objective, scope and outcome of the course.	Chalk Board
L2	Materials properties and selection: Mechanical Properties, stress strain diagram of different materials such as plain carbon steel, cast Iron, Aluminium, Definition of strengths.	Chalk Board
L3	Types of steels, IS coding of steels, alloy steels, Aluminium, Cast Iron	Chalk Board
L4	Selection of material from properties and economic aspects	Chalk Board
L5	Manufacturing considerations in Design: Standardization, Interchangeability, limits, tolerances	Chalk Board
L6	Fits, BIS Codes, Use of Tables	Chalk Board
L7	Numerical Problems based on limit and fits	Chalk Board
L8	Selective assembly, Surface roughness	Chalk Board
L9	Design consideration for casting, forging and machining parts	Chalk Board
	UNIT 2: Design for Strength	
L10	Design for Strength: Modes of failure, Strength and Stiffness considerations, Allowable stresses, factor of safety, selecting FOS values.	Chalk Board
L11	Stress concentration: causes and mitigation	Chalk Board
L12	Fatigue failures, Notch Sensitivity, Numerical problems	Chalk Board
L13	Applications & Design of Members subjected to direct stress: Direct tensile, compressive, bending, crushing, bearing and shear stresses, numerical problems.	Chalk Board
L14	Design of knuckle joints, Applications, numerical problems.	Chalk Board
L15	Cotter joint and its types, Design of spigot-socket, sleeve-cotter and gib-cotter joints.	Chalk Board
L16	Numerical problems on cotter joints	Chalk Board
L17	**UNIT TEST-1**	
	UNIT 3: Design of Member in Bending	
L18	Design of Members in Bending: Beams, design of beams	Chalk Board
L19	Levers. Types of levers. Design procedure for bell crank lever, Numerical Problems	Chalk Board

L20	Design of foot lever, and hand levers, Numerical problems	Chalk Board
L21	Laminated/ leaf spring, Design procedure of laminated springs, nipping	Chalk Board
L22	Numerical problems based on laminated spring	Chalk Board
L23	Design of beam based on rigidity criteria, use of maximum deflection formula for various end conditions for beam design, numerical problems	Chalk Board
UNIT 4: Design of Members in Torsion		
L24	Design of Members in Torsion: Shafts, Design of solid and hollow shafts based on strength, numerical problems	Chalk Board
L25	Shafts under combined loading. ASME criteria. Numerical problems	Chalk Board
L26	Design of solid and hollow shafts based on rigidity, Numerical problems	Chalk Board
L27	Key, types of keys, Design of Sunk keys, Numerical problems	Chalk Board
L28	Couplings: Types and uses. Design of muff coupling (rigid and clamped), Numerical Problems	Chalk Board
L29	Design of rigid flanged couplings, Numerical Problems,	Chalk Board
L30	Numerical Problem on rigid flanged couplings	Chalk Board
L31	Design of flexible flanged couplings, Numerical Problems	Chalk Board
L32	Numerical Problem on flexible flanged couplings	Chalk Board
L33	**UNIT TEST-2**	
UNIT 5: Design of Threaded fasteners, Power screw and Curved beams		
L34	Design of Threaded fasteners: Types of threads, thread- terminology, failure of threads. Bolt loading types, Bolt of uniform strength	Chalk Board
L35	Preloading of bolts: Effect of initial tension and applied loads+ numerical	Chalk Board
L36	Eccentric loading of bolts in plane, Numerical problems	Chalk Board
L37	Numerical problems based on eccentric loading	Chalk Board
L38	Power screws like lead screw & screw jack. Design procedures.	Chalk Board
L39	Numerical Problems based on power screws	Chalk Board
L40	Design of curve members, curved beam formulae, Numerical problems	Chalk Board
L41	Numerical problems on Curve members	Chalk Board
L42	Numerical problems on Curve members	Chalk Board

PO/PSO-Indicator-Competency

PO 1: Engineering knowledge: Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization for the solution of complex engineering problems.

Competency	Indicator
1.1 Demonstrate competence in mathematical modelling	1.1.1 Apply mathematical techniques such as calculus, linear algebra, and statistics to solve problems 1.1.2 Apply advanced mathematical techniques to model and solve mechanical engineering problems
1.2 Demonstrate competence in basic sciences	1.2.1 Apply laws of natural science to an engineering problem
1.3 Demonstrate competence in engineering fundamentals	1.3.1 Apply fundamental engineering concepts to solve engineering problems
1.4 Demonstrate competence in specialized engineering knowledge to the program	1.4.1 Apply Mechanical engineering concepts to solve engineering problems.

PO 2: Problem analysis: Identify, formulate, research literature, and analyze complex engineering problems reaching substantiated conclusions using first principles of mathematics, natural sciences, and engineering sciences.

Competency	Indicator
2.1 Demonstrate an ability to identify and formulate complex engineering problem	2.1.1 Articulate problem statements and identify objectives 2.1.2 Identify engineering systems, variables, and parameters to solve the problems 2.1.3 Identify the mathematical, engineering and other relevant knowledge that applies to a given problem
2.2 Demonstrate an ability to formulate a solution plan and methodology for an engineering problem	2.2.1 Reframe complex problems into interconnected sub- problems 2.2.2 Identify, assemble and evaluate information and resources. 2.2.3 Identify existing processes/solution methods for solving the problem, including forming justified approximations and assumptions 2.2.4 Compare and contrast alternative solution processes to select the best process.
2.3 Demonstrate an ability to formulate and interpret a model	2.3.1 Combine scientific principles and engineering concepts to formulate model/s (mathematical or otherwise) of a system or process that is appropriate in terms of applicability and required

	accuracy. 2.3.2 Identify assumptions (mathematical and physical) necessary to allow modeling of a system at the level of accuracy required.
2.4 Demonstrate an ability to execute a solution process and analyze results	2.4.1 Apply engineering mathematics and computations to solve mathematical models 2.4.2 Produce and validate results through skillful use of contemporary engineering tools and models 2.4.3 Identify sources of error in the solution process, and limitations of the solution. 2.4.4 Extract desired understanding and conclusions consistent with objectives and limitations of the analysis
PO3 Design/development of solutions: Design solutions for complex engineering problems and design system components or processes that meet the specified needs with appropriate consideration for the public health and safety, and the cultural, societal, and environmental considerations.	
Competency	**Indicator**
3.1 Demonstrate an ability to define a complex / open-ended problem in engineering terms	3.1.1 Recognize that need analysis is key to good problem definition 3.1.2 Elicit and document, engineering requirements from stakeholders 3.1.3 Synthesize engineering requirements from a review of the state-of-the-art 3.1.4 Extract engineering requirements from relevant engineering Codes and Standards such as ASME, ASTM, BIS, ISO and ASHRAE. 3.1.5 Explore and synthesize engineering requirements considering health, safety risks, environmental, cultural and societal issues 3.1.6 Determine design objectives, functional requirements and arrive at specifications
3.2 Demonstrate an ability to generate a diverse set of alternative design solutions	3.2.1 Apply formal idea generation tools to develop multiple engineering design solutions 3.2.2 Build models/prototypes to develop diverse set of design solutions 3.2.3 Identify suitable criteria for evaluation of alternate design solutions
3.3 Demonstrate an ability to select optimal design scheme for further	3.3.1 Apply formal decision making tools to select optimal engineering design solutions for further development

development	3.3.2 Consult with domain experts and stakeholders to select candidate engineering design solution for further development
3.4 Demonstrate an ability to advance an engineering design to defined end state	3.4.1 Refine a conceptual design into a detailed design within the existing constraints (of the resources) 3.4.2 Generate information through appropriate tests to improve or revise design
PO 6: The engineer and society: Apply reasoning informed by the contextual knowledge to assess societal, health, safety, legal, and cultural issues and the consequent responsibilities relevant to the professional engineering practice.	
6.1 Demonstrate an ability to describe engineering roles in a broader context, e.g. pertaining to the environment, health, safety, legal and public welfare	6.1.1 Identify and describe various engineering roles; particularly as pertains to protection of the public and public interest at global, regional and local level
6.2 Demonstrate an understanding of professional engineering regulations, legislation and standards	6.2.1 Interpret legislation, regulations, codes, and standards relevant to your discipline and explain its contribution to the protection of the public
PO14/PSO2: Participate and succeed in competitive examinations	
Competency	**Indicator**
14.1 Demonstrate an ability to success in competitive exams	14.1.1 Solve the objective and comprehensive questions from syllabus of competitive exams 14.1.2 Able answer the questions orally during interview

COs Competency Level

CO1	Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.
Competency	1.1 Demonstrate competence in Indian Standard codes and fundamentals of material section 1.2 Demonstrate competence in manufacturing consideration in design
CO2	Identify the factors for engineering components and analyze various members subjected to direct stress.
Competency	2.1 Demonstrate competence factors for engineering components. 2.2 Demonstrate an ability to analyse the members subjected to direct stresses and improve results.
CO3	Design various members such as beams, levers, laminated springs for bending and stiffness.
Competency	3.1 Demonstrate an ability to design the members under bending and stiffness and improve the design.
CO4	Design various machine components under torsion such as shafts, shaft couplings, and keys.
Competency	4.1 Demonstrate an ability to design the members under torsion and improve the design.
CO5	Design various threaded fasteners, power screws and curved machine components.
Competency	5.1 Demonstrate an ability to design the threaded members and improve the design.

CO-PO-PSO Mapping Using Performance Indicators (PIs)

	COs				
POs	**CO1**	**CO2**	**CO3**	**CO4**	**CO5**
PO1	1.3.1, 1.4.1		1.3.1, 1.4.1	1.3.1, 1.4.1	1.3.1, 1.4.1
PO2		2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2	2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2	2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2	2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2
PO3			3.1.4, 3.1.6 3.2.1,3.2.3 3.4.1	3.1.4, 3.1.6 3.2.1, 3.2.3 3.4.1	3.1.4, 3.1.6 3.2.1, 3.2.3 3.4.1
PO6			6.1.1	6.1.1	6.1.1
PO14/PSO2	14.1.1, 14.1.2	14.1.1, 14.1.2	14.1.1, 14.1.2	14.1.1, 4.1.2	14.1.1, 1 4.1.2

CO-PO-PSO Mapping: Formulation & Justification

CO	Course Outcomes	Bloom's Level	PO Indicators	PSO Indicators
5ME4-04.1:	Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing considerations in design.	L2	1.3.1, 1.4.1	2.1.1, 2.1.2
5ME4-04.2:	Identify the factors for engineering components design and analyze various members subjected to direct stress.	L4	2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2	2.1.1, 2.1.2
5ME4-04.3:	Design various members such as beams, levers, laminated springs for bending and stiffness.	L4	1.3.1, 1.4.1, 2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2, 3.1.4, 3.1.6, 3.2.1,3.2.3, 3.4.1, 6.1.1	2.1.1, 2.1.2
5ME4-04.4:	Design various machine components under torsion such as shafts, shaft couplings, and keys.	L4	1.3.1, 1.4.1, 2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2, 3.1.4, 3.1.6, 3.2.1,3.2.3, 3.4.1, 6.1.1	2.1.1, 2.1.2
5ME4-04.5:	Design various threaded fasteners, power screws and curved machine components.	L4	1.3.1, 1.4.1, 2.1.2,2.1.3, 2.2.3, 2.3.2, 2.4.1, 2.4.2, 3.1.4, 3.1.6, 3.2.1,3.2.3, 3.4.1, 6.1.1	2.1.1, 2.1.2

The CO-PO/PSO mapping is based on the correlation of course outcome (CO) with Program Outcome Indicators. These indicators are the breakup statements of broad Program Outcome statement.
The correlation is calculated as number of correlated indicators of a PO/PSO mapped with CO divided by total indicators of a PO/PSO. The calculated value represents the correlation level between a CO & PO/PSO.
Detailed formulation and mathematical representation can be seen below in equation 1:

Input: ***CO_i:*** *The i^{th} course outcome of the course*

PO_j: *The j^{th} Program Outcome*

Ij_k: *The k^{th} indicator of the j^{th} Program Outcome*

α (I_{jk}, CO_i): *level of CO-PO mapping*

$$=1,\ \ if,\ 0<\alpha<0.33$$

$$=2,\ \ if,\ 0.33\geq\alpha<0.66$$

$$=3,\ \ if,\ 0.66\geq\alpha<1$$

$$\alpha(I_{jk}, CO_i) = \frac{count(\ \lambda(I_{jk}, CO_i))}{count(I_k, PO_j)}$$

λ: *Degree of correlation*

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CO-PO/PSO Mapping

COs	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO 1	PSO 2
5ME4-04.1:	2	-	-	-	-	-	-	-	-	-	-	-	-	3
5ME4-04.2:	-	2	-	-	-	-	-	-	-	-	-	-	-	3
5ME4-04.3:	2	2	2	-	-	2	-	-	-	-	-	-	-	3
5ME4-04.4:	2	2	2	-	-	2	-	-	-	-	-	-	-	3
5ME4-04.5:	2	2	2	-	-	2	-	-	-	-	-	-	-	3
5ME4-04	2	2	2	-	-	2	-	-	-	-	-	-	-	3

Attainment Level (Internal Assessment)

Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur	
B.Tech III Year V Semester (Session 2022-2023)	
CO's Attainment (Theory Mid Term : I)	Department: Mechanical Engineering
Faculty Name: Dr. Prem Singh	Course Name with CODE: DME-I(5ME4-04)
Upon successful completion of this course, students will be able to:	
CO1: Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.	
CO2: Identify the factors for engineering components and analyze various members subjected to direct stress.	
CO3: Design various members such as beams, levers, laminated springs for bending and stiffness.	
CO4: Design various machine components under torsion such as shafts, shaft couplings, and keys.	
CO5: Design various threaded fasteners, power screws and curved machine components	

MID TERM EVALUATION											Section-B		
S. NO	ROLL NO	PART →	A			B			C		Total (20)	Assignment (10)	Total (30)
		Note→	Attempt All			Attempt Any Two			Attempt Any One				
		QUESTION NO. →	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8			
		COURSE OUTCOME(S) SATISFIED →	CO 1	CO 1	CO 2	CO 1	CO 1	CO 2	CO 2	CO 2			
		MAXIMUM MARKS →	2	2	2	4	4	4	6	6			
		MINIMUM QUALIFYING MARKS (50%) →	1	1	1	2	2	2	3	3			
		NAME OF STUDENT ↓											
1	20ESKME049	Kavita Sharma	0	1	2		4				7	8	15
2	20ESKME050	Keshav Singh	0	1.5	2		3.5	0	3		10	9	19
3	20ESKME051	Kiran Kumar	1.5	1.5	2	3		3	1		12	9	21
4	20ESKME052	Kunal Krishnani	1.5	1	2	2.5	3				10	8	18
5	20ESKME053	Lav Kumar	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	DB +NS
6	20ESKME054	Lavesh Jain	2	1	2	3	3			2	13	9	22

7	20ESKME055	Madan Lal Prajapat	2	1	2	2	3			3	13	9	22
8	20ESKME056	Madhurum Verma		1	2	3	2			3	11	9	20
9	20ESKME058	Mayank Mittal	2	1.5	2		3	4		5.5	18	10	28
10	20ESKME059	Mohit Pareek	2	2	2	4		3.5	5.5		19	10	29
11	20ESKME060	Naveen Kumar	2	1	1	3	3		3		13	9	22
12	20ESKME061	Navneet Sagar	2	1	2		4	2		2	13	8	21
13	20ESKME062	Nishant Tomar	2	1.5	1	3.5	3.5			2.5	14	8	22
14	20ESKME063	Nitin Sharma	2	1.5	2	4		3.5		4	17	10	27
15	20ESKME064	Nitish S Chauhan	2	1.5	2	3.5				3	12	9	21
16	20ESKME066	Parth Bhatt	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	DB +NS
17	20ESKME067	Pawan Bora					4			4	8	9	17
18	20ESKME068	Priyadarshini Singh Shekhawat	2	1	1		3.5	2	3.5		13	9	22
19	20ESKME069	Priyanshu Goyal	2	1	1		4	1		2	11	8	19
20	20ESKME070	Priyanshu Jangid	2	2	1	3.5	3.5	3			15	9	24
21	20ESKME071	Puru Raj Singh	2	1	1	3.5	3.5	3			14	9	23
22	20ESKME072	Rahul Patra	1	1	2		3.5	3.5			11	9	20
23	20ESKME073	Rahul Singh Gurjar	2	1	2	1	3.5		3.5		13	9	22
24	20ESKME074	Rajat Sharma	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	DB +NS
25	20ESKME075	Rajdeep Mathuria	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	DB +NS
26	20ESKME076	Rajvardhan Gupta	1	1.5	1.5		3.5	3.5		3	14	9	23
27	20ESKME077	Rohan Singh	1	1	1		3.5	3.5		3	13	8	21
28	20ESKME078	Sachin Sharma		1	1					2	4	9	13
29	20ESKME079	Samyak Jain	1	1	1	3	3.5			1.5	11	8	19

30	20ESKME080	Saurav Kumar Jha		1	1	3.5	3.5			4	13	9	22
31	20ESKME081	Shailendra Chauhan	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	DB +NS
32	20ESKME082	Shivam	2	1.5	2	2	3.5		2		13	9	22
33	20ESKME083	Shreshth Singh	2	1	2		3.5	2	1.5		12	9	21
34	20ESKME084	Shubh			2	3.5	3.5				9	8	17
35	20ESKME085	Shyam Kumar			2	3.5	3			1.5	10	9	19
36	20ESKME087	Sumit Khandal	2	1	2		3.5			2.5	11	9	20
37	20ESKME088	Suraj Jaimini	2	1.5	2	3.5	3.5			3.5	16	8	24
38	20ESKME089	Tehsin Khan	2	2	2	4	4			4	18	10	28
39	20ESKME090	Vaibhav Vyas	2	1.5	2		3.5	2		4	15	7	22
40	20ESKME091	Vikram Singh Rathore		1	2	3	3			2	11	8	19
41	20ESKME092	Vishnu Kumar			2		2	1		3	8	9	17
42	20ESKME093	Yaffee Gulzar	AB	AB	AB	AB	AB	AB	AB	AB	AB	NS	AB +NS
43	20ESKME094	Yash Goyal	2	2	2	4	4		3		17	10	27
44	20ESKME095	Yash R Thakur	2	1	1	3	3		2		12	8	20
45	20ESKME096	Yashi	2	2	1	3		2		1	11	9	20
46	20ESKME097	Yashvardhan Singh	DB	DB	DB	DB	DB	DB	DB	DB	DB	7	DB +7
47	20ESKME300	Yuvraj Singh Tanwar	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	DB +NS
48	21ESKME200	Mali Shailesh Rajendra	2	1	2	3.5	3.5		4		16	10	26
49	21ESKME201	Niraj Kumar	2	1.5	2	3.5	3.5		3.5		16	10	26
Total No. of DEBARRED (DB)			7	7	7	7	7	7	7	7			
Total No. of ABSENT (AB)			1	1	1	1	1	1	1	1			
Total Students Appreaed for Exam			41	41	41	41	41	41	41	41			
Total Students Attempted the Question (A)			33	37	40	26	35	17	12	23			

No. of Students scored >=50% marks (B)	31	37	40	25	35	14	8	13
Percentage Attainment of Criterion (B/A)	93.94	100.00	100.00	96.15	100.00	82.35	66.67	56.52
CO Attainment Level	**3**	**3**	**3**	**3**	**3**	**3**	**2**	**1**
Attainment of CO-1	**97.52%**							
Attainment of CO-2	**76.39%**							
Attainment of CO-3								
Attainment of CO-4								
Attainment of CO-5								
Criterion of Percentage for CO Attainment Level	**Attainment Level**							
Percentage attainment Below 60%	1							
Percentage attainment 60%-69.99%	2							
Percentage attainment Above and equal to 70%	3							

Dr, Prem Singh
Faculty name with signature

Swami Keshvanand Institute of Technology, Management & Gramothan,
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Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur	
B.Tech III Year V Semester (Session 2022-2023)	
CO's Attainment (Theory Mid Term : II)	**Department: Mechanical Engineering**
Faculty Name: Dr. Prem Singh	**Course Name with CODE: DME-I(5ME4-04)**
Upon successful completion of this course, students will be able to:	
CO1: Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.	
CO2: Identify the factors for engineering components and analyze various members subjected to direct stress.	
CO3: Design various members such as beams, levers, laminated springs for bending and stiffness.	
CO4: Design various machine components under torsion such as shafts, shaft couplings, and keys.	
CO5: Design various threaded fasteners, power screws and curved machine components	

MID TERM EVALUATION											**Section-B**		
S.NO.	**ROLL NO**	**PART →**	**A**			**B**			**C**		**Total (20)**	**Assignment (10)**	**Total (30)**
		Note→	**Attempt All**			**Attempt Any Two**			**Attempt Any One**				
		QUESTION NO. →	**Q1**	**Q2**	**Q3**	**Q4**	**Q5**	**Q6**	**Q7**	**Q8**			
		COURSE OUTCOME(S) SATISFIED →	**CO 5**	**CO 5**	**CO 5**	**CO 4**	**CO 4**	**CO 4**	**CO 3**	**CO 3**			
		MAXIMUM MARKS →	**2**	**2**	**2**	**4**	**4**	**4**	**6**	**6**			
		MINIMUM QUALIFYING MARKS (50%) →	**1**	**1**	**1**	**2**	**2**	**2**	**3**	**3**			
		NAME OF STUDENT ↓											
1	20ESKME049	Kavita Sharma	1	1	1	2					5	8	13
2	20ESKME050	Keshav Singh	1	1	1			2	3		8	8	16
3	20ESKME051	Kiran Kumar	1	1.5	2		2	1.5		4	12	9	21
4	20ESKME052	Kunal Krishnani	1	1.5		1	2		2		8	7	15
5	20ESKME053	Lav Kumar	DB	DB	DB	DB	DB	DB	DB	DB	DB	8	8
6	20ESKME054	Lavesh Jain	2	1.5	1		2	2.5	4		13	9	22
7	20ESKME055	Madan Lal Prajapat	2	1.5	1	2	2		3.5		12	8	20

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8	20ESKME056	Madhurum Verma	1	1	1.5	3	2.5		1		10	9	19
9	20ESKME058	Mayank Mittal	2	1.5	1.5	4	3		3		15	10	25
10	20ESKME059	Mohit Pareek	2	1.5	1.5	4	3		3		15	10	25
11	20ESKME060	Naveen Kumar	1	1.5	1.5	3	3				10	10	20
12	20ESKME061	Navneet Sagar	1	1.5	1.5	3					7	9	16
13	20ESKME062	Nishant Tomar	2	1.5	1.5	2			3		10	9	19
14	20ESKME063	Nitin Sharma	2	1.5	1.5	2	2			3	12	10	22
15	20ESKME064	Nitish S Chauhan	2	1.5	1.5					3	8	8	16
16	20ESKME066	Parth Bhatt	1	1	1						3	NS	3
17	20ESKME067	Pawan Bora	1.5	1.5			3			2	8	9	17
18	20ESKME068	Priyadarshini Singh Shekhawat	1.5	1.5	2		2	2		3	12	9	21
19	20ESKME069	Priyanshu Goyal	1	1	1.5	2		2.5			8	9	17
20	20ESKME070	Priyanshu Jangid	1	1	1.5			3.5	3		10	9	19
21	20ESKME071	Puru Raj Singh	1	1	1.5		2	3.5			9	8	17
22	20ESKME072	Rahul Patra	2	1.5	1.5	2	2				9	9	18
23	20ESKME073	Rahul Singh Gurjar	1	1.5	1.5		2	2			8	8	16
24	20ESKME074	Rajat Sharma	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	0
25	20ESKME075	Rajdeep Mathuria	1.5	1.5	1		2	2			8	9	17
26	20ESKME076	Rajvardhan Gupta	1.5	1.5	1.5			2.5	2		9	9	18
27	20ESKME077	Rohan Singh	1.5	1.5	1.5			2	2		9	9	18
28	20ESKME078	Sachin Sharma	1.5	1.5	1.5		1.5	2			8	9	17

29	20ESKME079	Samyak Jain	1.5	1	1.5	2		2	1		9	9	18
30	20ESKME080	Saurav Kumar Jha	1.5	1.5	1.5	2		2	1		10	9	19
31	20ESKME081	Shailendra Chauhan	1.5	1	1.5						4	NS	4
32	20ESKME082	Shivam	1	1.5	1.5				3		7	9	16
33	20ESKME083	Shreshth Singh	1.5	1.5	1.5		2	2.5			9	7	16
34	20ESKME084	Shubh	1	1.5	1.5		2	2	3		11	7	18
35	20ESKME085	Shyam Kumar	1.5	1	1.5		2	2	3		11	10	21
36	20ESKME087	Sumit Khandal	1.5	1				4	1.5		8	9	17
37	20ESKME088	Suraj Jaimini	1.5	1.5	1.5		2	4	4.5		15	10	25
38	20ESKME089	Tehsin Khan	1.5	1.5	1.5	2.5		4	3		14	10	24
39	20ESKME090	Vaibhav Vyas	1.5	1.5	1.5		2	3	3.5		13	10	23
40	20ESKME091	Vikram Singh Rathore	1.5	1.5	1.5	2		2		3	12	8	20
41	20ESKME092	Vishnu Kumar	1.5	1.5	1.5		2.5	3	4		14	10	24
42	20ESKME093	Yaffee Gulzar	AB	AB	AB	AB	AB	AB	AB	AB	AB	NS	0
43	20ESKME094	Yash Goyal	1.5	1.5	1.5	2	2		1.5		10	10	20
44	20ESKME095	Yash R Thakur	1.5	1.5	1.5	2	1.5			3	11	9	20
45	20ESKME096	Yashi	1.5	1.5	1		2	1		2	9	9	18
46	20ESKME097	Yashvardhan Singh	AB	AB	AB	AB	AB	AB	AB	AB	AB	NS	0
47	20ESKME300	Yuvraj Singh Tanwar	DB	DB	DB	DB	DB	DB	DB	DB	DB	NS	0
48	21ESKME200	Mali Shailesh Rajendra	1.5	1.5	1.5		2	2.5		4	13	8	21
49	21ESKME201	Niraj Kumar	1.5	1.5	1.5	2	2			3.5	12	10	22

Total No. of DEBARRED (DB)	3	3	3	3	3	3	3	3
Total No. of ABSENT (AB)	2	2	2	2	2	2	2	2
Total Students Appreaed for Exam	44	44	44	44	44	44	44	44
Total Students Attempted the Question (A)	44	44	41	19	27	25	22	10
No. of Students scored >=50% marks (B)	44	44	41	18	25	23	14	8
Percentage Attainment of Criterion (B/A)	100.00	100.00	100.00	94.74	92.59	92.00	63.64	80.00
CO Attainment Level	**3**	**3**	**3**	**3**	**3**	**3**	**2**	**3**
Attainment of CO-1								
Attainment of CO-2								
Attainment of CO-3	**71.82%**							
Attainment of CO-4	**93.11%**							
Attainment of CO-5	**100.00%**							
Criterion of Percentage for CO Attainment Level	**Attainment Level**							
Percentage attainment Below 60%	1							
Percentage attainment 60%-69.99%	2							
Percentage attainment Above and equal to 70%	3							

Dr, Prem Singh
Faculty name with signature

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Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur
B.Tech III Year V Semester (Session 2021-2022)

CO's Attainment (Theory Mid Term-I) | **Mechanical Engineering** | **Date:28/10/2021**

Faculty Name: Dr. Prem Singh | **Design of Machine Elements-I (5ME4-04)**

Upon successful completion of this course, students will be able to:

CO1: Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.
CO2: Identify the factors for engineering components and analyze various members subjected to direct stress.
CO3: Design various members such as beams, levers, laminated springs for bending and stiffness.
CO4: Design various machine components under torsion such as shafts, shaft couplings, and keys.
CO5: Design various threaded fasteners, power screws and curved machine components

FIRST MID TERM EVALUATION

		QUESTION NO.	**Q1**	**Q2**	**Q3**	**Q4**	**Q5**	**Mid term Marks (50)**	**Scaled Mid-term Marks (24)**	**Assignment (6)**	**Total (30)**
		COURSE OUTCOME(S)	**CO 1**	**CO 1**	**CO 1**	**CO 2**	**CO 2**				
		MAXIMUM MARKS	**10**	**10**	**10**	**10**	**10**				
		MINIMUM QUALIFYING MARKS (50%)	**5**	**5**	**5**	**5**	**5**				
S.N.	**Roll No.**	**NAME OF STUDENT**	**10**	**10**	**10**	**10**	**10**				
1	19ESKME057	DEVASHEESH SHARMA	8	9	10	9	8	44	22	5	27
2	19ESKME058	DEVASHISH MUNDRA	8	9	10	9	8	44	22	4	26
3	19ESKME059	DEVESH SHRIMAL	9	8	10	8	8	43	21	5	26
4	19ESKME060	DHARMA RAM JAT	8	5	10	9	8	40	20	5	25
5	19ESKME061	DIPESH GOYAL	8	8	9	8	8	41	20	5	25
6	19ESKME062	DIVYA ARORA	7	8	10	9	8	42	21	5	26
7	19ESKME063	DIVYANSH CHATURVEDI	8	7	7	9	8	39	19	5	24
8	19ESKME064	GAURAV DUBEY	9	8	10	9	8	44	22	5	27
9	19ESKME065	GAURAV JAIN	9	7	10	9	8	43	21	5	26
10	19ESKME066	GAURAV KUMAR	9	5	10	9	8	41	20	NS	20
11	19ESKME067	GAURAV MISHRA	9	7	10	9	8	43	21	5	26
12	19ESKME069	HARBEET	10	6	10	9	8	43	21	NS	21
13	19ESKME070	HARI OM MAHIYA	9	6	10	9	8	42	21	5	26
14	19ESKME071	HARSH MAHESHWARI	9	7	10	9	8	43	21	5	26
15	19ESKME072	HARSH SHARMA	9	8	10	9	8	44	22	6	28
16	19ESKME073	HARSH SINGH GAHLOT	9	7	10	9	8	43	21	NS	21
17	19ESKME074	HARSHIKA KUMARI	AB	AB	AB	AB	AB	AB	AB	5	AB+05

18	19ESKME075	HEMANT YOGI	9	8	10	9	8	44	22	5	27
19	19ESKME076	HITESH THADANI	9	7	10	9	8	43	21	4	25
20	19ESKME077	JALAJ PRAKASH GUPTA	9	9	8	10	8	44	22	5	27
21	19ESKME078	JANCY C JOSHWA	9	9	10	8	9	45	22	6	28
22	19ESKME079	JATIN CHAUDHARY	8	8	8	8	9	41	20	5	25
23	19ESKME080	JATIN KUMAR YADAV	8	9	9	8	8	42	21	NS	21
24	19ESKME081	KAELIN	9	9	9	8	8	43	21	5	26
25	19ESKME082	KAPIL SIDDHU	9	9	8	8	8	42	21	NS	21
26	19ESKME083	KARAN SONI	8	8	8	8	8	40	20	4	24
27	19ESKME084	KHUSHANK SHARMA	8	6	8	8	8	38	19	NS	19
28	19ESKME085	KOMAL YADAV	7	8	8	8	8	39	19	5	24
29	19ESKME086	KSHITIJ KALRA	9	9	8	10	8	44	22	5	27
30	19ESKME087	KULDEEP SHARMA	8	9	8	8	8	41	20	5	25
31	19ESKME088	KUNAL COLIN WILLIAMS	8	9	8	8	8	41	20	6	26
32	19ESKME089	KUNAL MANIWAL	8	8	8	10	8	42	21	5	26
33	19ESKME090	KUNAL MITTAL	7	7	7	8	8	37	18	5	23
34	19ESKME091	LABHANSH SHARMA	8	7	7	8	8	38	19	4	23
35	19ESKME092	LALIT SHARMA	7	8	7	8	8	38	19	4	23
36	19ESKME093	LAVANSHU GARG	7	8	7	8	8	38	19	4	23
37	19ESKME094	LILADHAR GADWAL	9	7	8	8	8	40	20	4	24
38	19ESKME095	LOVENASH SINGHAL	9	9	8	9	8	43	21	6	27
39	19ESKME096	MADHUR KALA	8	10	7	7	6	38	19	5	24
40	19ESKME098	MANURADITYA SINGH HADA	9	10	8	6	6	39	19	5	24
41	19ESKME099	MOHD MATEEN JOAD	9	10	8	7	8	42	21	NS	21
42	19ESKME101	MOHIT OLA	6	10	5	6	6	33	16	4	20
43	19ESKME102	MUKUL DANGAYACH	9	10	0	6	6	31	15	NS	15
44	19ESKME103	MUKUL GARG	9	10	8	9	9	45	22	5	27
45	19ESKME104	MUSKAN RANGREJ	9	10	9	8	8	44	22	5	27
46	19ESKME106	NAVAL TRIPATHI	9	10	8	8	9	44	22	4	26
47	19ESKME107	NAVEEN CHOUDHARY	9	10	8	7	8	42	21	5	26
48	19ESKME108	NIKHIL PANDEY	9	10	9	8	9	45	22	6	28
49	19ESKME110	OJASVEE SHARMA	9	10	8	7	8	42	21	5	26
50	19ESKME111	PARTH SHARMA	9	10	8	8	9	44	22	5	27
51	19ESKME112	PAWAN PAREEK	8	10	9	7	8	42	21	5	26
52	19ESKME113	PAWAS BANSAL	9	10	8	7	8	42	21	5	26

53	19ESKME114	PRADEEP KUMAR PRAJAPATI	9	10	8	0	8	35	17	5	22
54	19ESKME115	PRADYUM SHARMA	9	10	8	7	9	43	21	4	25
55	19ESKME116	PRAKHAR GAUR	9	10	9	7	9	44	22	4	26
Total Students Eligible for Exam			55	55	55	55	55				
Total Students Attempted the Question (A)			54	54	54	54	54				
No. of Students scored >=50% marks (B)			54	54	53	53	54				
Percentage Attainment of Criterion Marks (B/A)			**100.00**	**100.00**	**98.15**	**98.15**	**100.00**				
CO			CO 1	CO 2	CO 3	CO 4	CO 5				
Percentage Attainment of COs			99.38	99.07	-	-	-				
***CO Attainment Level (AL)**			3	3	-	-	-				

Criterion of Percentage for CO Attainment Level	**Attainment Level**
Percentage attainment Below 60%	1
Percentage attainment 60%-69.99%	2
Percentage attainment Above and equal to 70%	3

Dr. Prem Singh
Faculty name with signature

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Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur

B.Tech III Year V Semester (Session 2021-2022)

CO's Attainment (Theory Mid Term : II)

Department: Mechanical Engineering
Date:20/12/2021

Faculty Name: Dr. Prem Singh

Course Name with CODE: Design of Machine Elements-I (5ME4-04)

Upon successful completion of this course, students will be able to:

CO1: Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.

CO2: Identify the factors for engineering components and analyze various members subjected to direct stress.

CO3: Design various members such as beams, levers, laminated springs for bending and stiffness.

CO4: Design various machine components under torsion such as shafts, shaft couplings, and keys.

CO5: Design various threaded fasteners, power screws and curved machine components

MID TERM EVALUATION											(Section-B)		
S. N.	ROLL NO	PART →	A				B			C		Mid (24)	Assignment (6)	Total (30)
		Note→	Attempt All				Attempt Any Two			Attempt Any One				
		QUESTION NO. →	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8	Q9			
		COURSE OUTCOME(S) SATISFIED →	CO 2	CO 5	CO 5	CO 5	CO 3	CO 4	CO 4	CO 3	CO 4			
		MAXIMUM MARKS →	2	2	2	2	4	4	4	8	8			
		MINIMUM QUALIFYING MARKS (50%) →	1	1	1	1	2	2	2	4	4			
		NAME OF STUDENT ↓												
1	19ESKME057	DEVASHEESH SHARMA	1	0	1	1	3.5	2.5		6.5		16	4	20
2	19ESKME058	DEVASHISH MUNDRA	1	0.5	1	1	2	3.5		7.5		17	5	22
3	19ESKME059	DEVESH SHRIMAL	1.5	1	2	1	2	2		7.5		17	5	22
4	19ESKME060	DHARMA RAM JAT	1	1	1	2		3	2	2		12	4	16
5	19ESKME061	DIPESH GOYAL	1.5	0.5	1	1	2		2		1	9	6	15
6	19ESKME062	DIVYA ARORA	1.5	0	1	2	2	0				7	3	10
7	19ESKME063	DIVYANSH CHATURVEDI	1.5	1	0.5	2		1	3	1		10	6	16
8	19ESKME064	GAURAV DUBEY	1.5	1	1	1		4	3	4		16	6	22
9	19ESKME065	GAURAV JAIN	1.5	1	2	0.5		4	3	2		14	5	19
10	19ESKME066	GAURAV KUMAR	0.5	0	1	0	0		1	0		3	3	6
11	19ESKME067	GAURAV MISHRA	1.5	0.5	1	0.5	3	1	7			15	6	21
12	19ESKME069	HARBEET	1	1	1	0		2	2	4		11	4	15
13	19ESKME070	HARI OM MAHIYA	1.5	0	0	0	0		2	1		5	3	8
14	19ESKME071	HARSH MAHESHWARI	1.5	0	1		1		2	1		7	3	10
15	19ESKME072	HARSH SHARMA	2	1	1	2		4	3	7.5		21	6	27

16	19ESKME073	HARSH SINGH GAHLOT	2	0.5	2	0.5		3.5	3			12	5	17
17	19ESKME074	HARSHIKA KUMARI	1.5	0.5	1			3	2	2		10	3	13
18	19ESKME075	HEMANT YOGI	2	1	1	2	0		1	1		8	4	12
19	19ESKME076	HITESH THADANI	0.5	0.5	1	1	2		1		1	7	NS	7
20	19ESKME077	JALAJ PRAKASH GUPTA	2	1	2	2	3	2.5		7		20	6	26
21	19ESKME078	JANCY C JOSHWA	2	2	2	2	2.5	2		1		14	5	19
22	19ESKME079	JATIN CHAUDHARY	1	0	1	1	2.5	1		2		9	3	12
23	19ESKME080	JATIN KUMAR YADAV	1	1	1	2	2				1	8	6	14
24	19ESKME081	KAELIN	1	1	1	0.5	2	2				8	6	14
25	19ESKME082	KAPIL SIDDHU	AB	AB	AB	AB	AB	AB	AB	AB	AB	AB	NS	AB
26	19ESKME083	KARAN SONI	2	1	2	0.5		2.5	2	1		11	3	14
27	19ESKME084	KHUSHANK SHARMA	1	1	2	2	2.5					9	NS	9
28	19ESKME085	KOMAL YADAV	1	1	1	0.5	1					5	6	11
29	19ESKME086	KSHITIJ KALRA	2	1	1	2	1	1		2		10	5	15
30	19ESKME087	KULDEEP SHARMA	1.5	0.5	1		1		1	1		6	5	11
31	19ESKME088	KUNAL COLIN WILLIAMS	1	1	1	2		1	1	2		9	6	15
32	19ESKME089	KUNAL MANIWAL	1	1	1	2	2		0	0		7	6	13
33	19ESKME090	KUNAL MITTAL	2	0			0		1		1	4	5	9
34	19ESKME091	LABHANSH SHARMA	2			2		0	0	0		4	5	9
35	19ESKME092	LALIT SHARMA	0		2		1		2	8		13	4	17
36	19ESKME093	LAVANSHU GARG	0.5		1		0		2	5.5		9	4	13
37	19ESKME094	LILADHAR GADWAL	2		2	1		2	2	6		15	5	20
38	19ESKME095	LOVENASH SINGHAL	1.5	0.5	1	0	3		3	7.5		17	5	22
39	19ESKME096	MADHUR KALA	2	1	2	2				5		12	4	16
40	19ESKME098	MANURADITYA SINGH HADA	2	1	1			1	1	1		7	5	12
41	19ESKME099	MOHD MATEEN JOAD	2	2	2	2	1	1			1	11	4	15
42	19ESKME101	MOHIT OLA	0	1	1	2			3			7	3	10
43	19ESKME102	MUKUL DANGAYACH	0.5	0.5	2	2	2					7	NS	7
44	19ESKME103	MUKUL GARG	1				2		3	5		11	5	16
45	19ESKME104	MUSKAN RANGREJ	1.5	1	1	2		1	1	6.5		14	6	20
46	19ESKME106	NAVAL TRIPATHI	1	1	1	0.5	2		2	2.5		10	5	15
47	19ESKME107	NAVEEN CHOUDHARY	1.5	1	1	0.5	1		1	4		10	4	14
48	19ESKME108	NIKHIL PANDEY	1.5		2	2	2		3	2		13	6	19
49	19ESKME110	OJASVEE SHARMA	2	2	1	0.5	2	1			1	10	6	16
50	19ESKME111	PARTH SHARMA	2	1	1		1		1	1		7	4	11

51	19ESKME112	PAWAN PAREEK	2	1	1	2	0.5		1.5		1	9	6	15
52	19ESKME113	PAWAS BANSAL	2	1	1	2	1	0		1		8	4	12
53	19ESKME114	PRADEEP KUMAR PRAJAPATI	2	1	1	2		2	2	3		13	6	19
54	19ESKME115	PRADYUM SHARMA	1	1	1	2	2	2		3		12	4	16
55	19ESKME116	PRAKHAR GAUR	1.5			2	1			1		6	5	11
Total No. of DEBARRED (DB)			0	0	0	0	0	0	0	0	0			
Total No. of ABSENT (AB)			1	1	1	1	1	1	1	1	1			
Total Students Appreaed for Exam			54	54	54	54	54	54	54	54	54			
Total Students Attempted the Question (A)			54	47	50	45	37	29	35	39	7			
No. of Students scored >=50% marks (B)			48	31	48	32	21	17	22	15	0			
Percentage Attainment of Criterion (B/A)			88.89	65.96	96.00	71.11	56.76	58.62	62.86	38.46	0.00			
Cos			CO1	CO2	CO3	CO4	CO5							
Percentage Attainment of COs			-	88.89	47.61	40.49	77.69							
CO Attainment Level			-	3.00	1.00	1.00	3.00							

Criterion of Percentage for CO Attainment Level	**Attainment Level**
Percentage attainment Below 60%	1
Percentage attainment 60%-69.99%	2
Percentage attainment Above and equal to 70%	3

Dr. Prem Singh

Faculty name with signature

Swami Keshvanand Institute of Technology, Management & Gramothan, Ramnagaria, Jagatpura, Jaipur-302017, INDIA
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Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur
B.Tech III Year V Semester (Session 2020-2021)

CO's Attainment (Theory Mid Term-I) — **Mechanical Engineering** **Date:01/10/2020**

Faculty Name: Dr. Prem Singh — **Design of Machine Elements-I (5ME4-04)**

Upon successful completion of this course, students will be able to:

CO1: Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.
CO2: Identify the factors for engineering components and analyze various members subjected to direct stress.
CO3: Design various members such as beams, levers, laminated springs for bending and stiffness.
CO4: Design various machine components under torsion such as shafts, shaft couplings, and keys.
CO4: Design various threaded fasteners, power screws and curved machine components

FIRST MID TERM EVALUATION

		QUESTION NO.	Q1	Q2	Q3	Q4	Q5	Mid term Marks (50)	Scaled Mid-term Marks (24)	Assignment (6)	Total marks (30)
		COURSE OUTCOME (S)	CO 1	CO 2	CO 2	CO 3	CO 3				
		MAXIMUM MARKS	10	10	10	10	10				
		MINIMUM QUALIFYING MARKS (50%)	5	5	5	5	5				
S.No.	Roll No.	NAME OF STUDENT	10	10	10	10	10				
1	18ESKME054	Jayesh Verma	10	8	2		9.5	30	15	4	19
2	18ESKME055	Juber Khan	7	8	6	10		31	15		15
3	18ESKME056	Kalpit Arya	9	9	4		9	31	15	6	21
4	18ESKME057	Kapil Raj Tanwar	9	9	4		9	31	16	6	22
5	18ESKME059	Karan Chawda	8	8	10		7.5	34	17	6	23
6	18ESKME060	Kartik Singhal	10	7			8.5	26	13		13
7	18ESKME061	Kashish Nawal	10	7			10	27	13	5	18
8	18ESKME062	Khushal Singh Panwar	8	4	8		7	27	13	6	19
9	18ESKME063	Kishlay Thakur	7	6	5	10	6	34	17	5	22
10	18ESKME065	Kshitiz Anurag	8	4	6	10	4	32	16	6	22
11	18ESKME066	Love Kumar	10		6	10	8	34	17	6	23
12	18ESKME067	Luckey Sharma	10	8	5	10	4	37	18	6	24

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13	18ESKME068	Mannat Mehta	7	4	5	10	10	36	18	6	24
14	18ESKME069	Manoj Kumar Sahu	10	9	9	10	8	46	23	5	28
15	18ESKME070	Mohammed Danish	8	9	8	10	8	43	21	4	25
16	18ESKME071	Mohit Gautam	7		5	10	10	32	16		16
17	18ESKME072	Mohit Kumar Meena	8		4	10	8	30	15		15
18	18ESKME073	Mohit Tolani	10		6	10	10	36	18	6	24
19	18ESKME074	Naincy Kamthan	10	7		10	3	30	15		15
20	18ESKME075	Naitik Popli	8	5		10	10	33	16		16
21	18ESKME076	Navdeep Singh Rathore	8	8	9	6	3	34	17	6	23
22	18ESKME077	Naveen Pareek	8	9	9	6		32	16	6	22
23	18ESKME078	Neeraj Choudhary	9	6	8	5		28	14	4	18
24	18ESKME079	Nikhil Bhatia	8	9	8	5		30	15	6	21
25	18ESKME080	Omprakash Dhakar	8	8	8	5		29	14	5	19
26	18ESKME081	Palash Madhukar	8	8	7	5		28	14		14
27	18ESKME082	Pankaj Yadav	8	6	4	5		23	12	6	18
28	18ESKME083	Panna Dan Rav	9	9	6	7	4	35	17	4	21
29	18ESKME084	Parth Sharma	8	9	7	5		29	14	5	19
30	18ESKME085	Pawan Kumar	8	9	7	4	2	30	15	6	21
31	18ESKME086	Pimpale Dipesh Gajanan	8	9	7	5	4	33	16	6	22
32	18ESKME088	Pradeep Jangid	8	9	7	5	2	31	15	5	20
33	18ESKME089	Pradyuman	8	9	7	5	2	31	15	3	18
34	18ESKME090	Prakhar Bhardwaj	8	9	10	8	10	45	22	5	27
35	18ESKME091	Prateek Kumar	9	8	10	8	10	45	22		22
36	18ESKME092	Praveen Kathat	9	8	8	10	10	45	22		22
37	18ESKME093	Priyanka Soni	8	9	9	10	10	46	23	4	27
38	18ESKME094	Puneet Kumawat	8	9	10	9	10	46	23	5	28
39	18ESKME095	Raghuraj Singh Shekhawat	8	9	8	9	10	44	22		22

40	18ESKME096	Rajan Sharma	9	8	8	10	10	45	22		22
41	18ESKME097	Rajat Gagar	9	9	10	10	10	48	24	5	29
42	18ESKME098	Rajat Gautam	9	8	8	10	10	45	22	4	26
43	18ESKME099	Rajat Kumar Saini	9	9	10	10	10	48	24	6	30
44	18ESKME101	Ravi Shankar Suthar	8	8	8	4	10	38	19	5	24
45	18ESKME102	Rhythm Purohit	8	9	10	10	10	47	23	6	29
46	18ESKME103	Rishabh Shrivastava	9	10	9	10	10	48	24	6	30

Total Students Appreaed for Exam			46	46	46	46	46
Total Students Attempted the Question			46	42	42	39	38
No. of Students scored >=50% marks			46	39	37	37	29
Percentage Attainment of COs			**1.00**	**0.85**	**0.80**	**0.80**	**0.63**
CO Attainment Level			**3.00**	**3.00**	**3.00**	**3.00**	**2.00**

Attainment of CO-1	**100 %**
Attainment of CO-2	**83%**
Attainment of CO-3	**72%**

Faculty name with signature

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B.Tech III Year V Semester (Session 2020-2021)

CO's Attainment (Theory Mid Term-II)

Mechanical Engineering
Date:02/12/2020

Faculty Name: Dr. Prem Singh

Design of Machine Elements-I (5ME4-04)

Upon successful completion of this course, students will be able to:

CO1: Apply the knowledge of Indian Standard codes and engineering fundamentals of material selection and manufacturing consideration in design.

CO2: Identify the factors for engineering components and analyze various members subjected to direct stress.

CO3: Design various members such as beams, levers, laminated springs for bending and stiffness.

CO4: Design various machine components under torsion such as shafts, shaft couplings, and keys.

CO4: Design various threaded fasteners, power screws and curved machine components

FIRST MID TERM EVALUATION

		QUESTION NO.	**Q1**	**Q2**	**Q3**	**Q4**	**Q5**	**Mid term Marks (50)**	**Scaled Mid-term Marks (24)**	**Assignment (6)**	**Total marks (30)**
		COURSE OUTCOME (S)	**CO 1**	**CO 4**	**CO 4**	**CO 4**	**CO 5**				
		MAXIMUM MARKS	**10**	**10**	**10**	**10**	**10**				
		MINIMUM QUALIFYING MARKS (50%)	**5**	**5**	**5**	**5**	**5**				
S.No.	**Roll No.**	**NAME OF STUDENT**	**10**	**10**	**10**	**10**	**10**				
1	18ESKME054	Jayesh Verma	10	10	10	10	4	44	22	5	27
2	18ESKME055	Juber Khan	9	10	10	10	6	45	22	0	22
3	18ESKME056	Kalpit Arya	9	10	10	10	8	47	23	6	29
4	18ESKME057	Kapil Raj Tanwar	9	10	10	10	9	48	24	5	29
5	18ESKME059	Karan Chawda	9	10	10	10	7	46	23	5	28
6	18ESKME060	Kartik Singhal	8	10	10	10	7	45	22	NS	22
7	18ESKME061	Kashish Nawal	8	10	10	10	8	46	23	6	29
8	18ESKME062	Khushal Singh Panwar	9	10	8	10	9	46	23	6	29
9	18ESKME063	Kishlay Thakur	9	10	10	10	8	47	23	6	29
10	18ESKME065	Kshitiz Anurag	9	10	10	10	9	48	24	5	29
11	18ESKME066	Love Kumar	6	10	10	10	9	45	22	NS	22
12	18ESKME067	Luckey Sharma	9	10	10	10	9	48	24	6	30

13	18ESKME068	Mannat Mehta	9	10	10	9	2	40	20	6	26
14	18ESKME069	Manoj Kumar Sahu	10	10		10	9	39	19	6	25
15	18ESKME070	Mohammed Danish	10	8		9	9	36	18	6	24
16	18ESKME071	Mohit Gautam	8				8	16	8	NS	8
17	18ESKME072	Mohit Kumar Meena	10			2	9	21	11	6	17
18	18ESKME073	Mohit Tolani	10		10	8	9	37	18	5	23
19	18ESKME074	Naincy Kamthan	10		9	8	8	35	17	NS	17
20	18ESKME075	Naitik Popli	9			8	8	25	13	NS	13
21	18ESKME076	Navdeep Singh Rathore	9		9	9	8	35	17	6	23
22	18ESKME077	Naveen Pareek	9		9	9	9	36	18	5	23
23	18ESKME078	Neeraj Choudhary	10		9	8	9	36	18	NS	18
24	18ESKME079	Nikhil Bhatia	10			8	9	27	13	6	19
25	18ESKME080	Omprakash Dhakar	10	6	8	2	9	35	17	5	22
26	18ESKME081	Palash Madhukar	8				9	17	9	5	14
27	18ESKME082	Pankaj Yadav	8	8	9	10	8	43	21	5	26
28	18ESKME083	Panna Dan Rav	8	8	10	9	10	45	22	6	28
29	18ESKME084	Parth Sharma	8	10	10	10	10	48	24	6	30
30	18ESKME085	Pawan Kumar	8	6	10	10	10	44	22	6	28
31	18ESKME086	Pimpale Dipesh Gajanan	8	9	10	10	10	47	23	6	29
32	18ESKME088	Pradeep Jangid	8	9	10	10	10	47	23	6	29
33	18ESKME089	Pradyuman	8	8	9	9	10	44	22	6	28
34	18ESKME090	Prakhar Bhardwaj	8	9	10	10	10	47	23	6	29
35	18ESKME091	Prateek Kumar	8	9	10	10	10	47	23	NS	23
36	18ESKME092	Praveen Kathat	8	9	10	10	10	47	23	NS	23
37	18ESKME093	Priyanka Soni	9	9	10	9	10	47	23	5	28
38	18ESKME094	Puneet Kumawat	8	10	10	10	10	48	24	6	30

39	18ESKME095	Raghuraj Singh Shekhawat	6	7	8	8		21	11	NS	11
40	18ESKME096	Rajan Sharma	6	6		8		20	10	6	16
41	18ESKME097	Rajat Gagar	9	10	10	10	10	49	24	6	30
42	18ESKME098	Rajat Gautam	5	5	8	8	3	29	14	6	20
43	18ESKME099	Rajat Kumar Saini	8	8	6	8	3	33	16	6	22
44	18ESKME101	Ravi Shankar Suthar	8	3	3	8		22	11	NS	11
45	18ESKME102	Rhythm Purohit	7	7	3	8	4	29	14	6	20
46	18ESKME103	Rishabh Shrivastava	7	7	8	8	4	34	17	6	23

Total Students Appreaed for Exam			46	46	46	46	46
Total Students Attempted the Question			46	36	38	44	43
No. of Students scored >=50% marks			46	35	36	42	37
Percentage Attainment of COs			**1.00**	**0.76**	**0.78**	**0.91**	**0.80**
CO Attainment Level			**3.00**	**3.00**	**3.00**	**3.00**	**3.00**

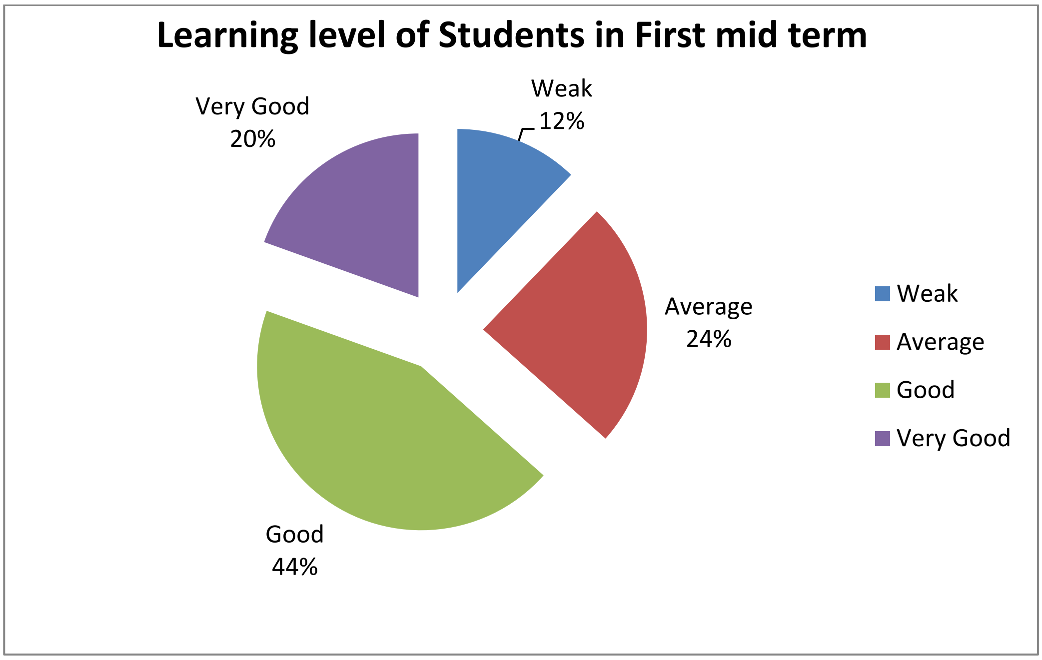
Attainment of CO-1	**100 %**
Attainment of CO-4	**82%**
Attainment of CO-5	**80%**

Faculty name with signature

Learning Levels of Students through Marks Obtained in 1st Mid Term Exam (2022-23)

B.Tech. V Semester, Section-B

Subject: Design of Machine Elements-I



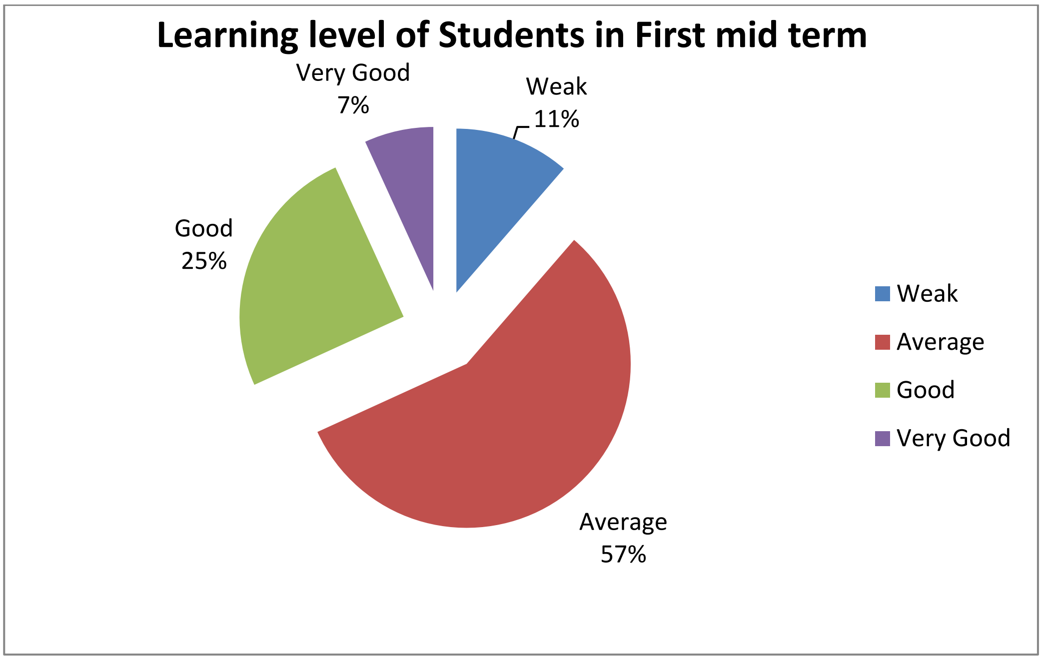
List of weak Students (Below 50% Marks) in 1st Mid term

Sr No.	Roll	Name	Ist Mid-term (20)
1	20ESKME049	Kavita Sharma	7
2	20ESKME067	Pawan Bora	8
3	20ESKME078	Sachin Sharma	4
4	20ESKME084	Shubh	9
5	20ESKME092	Vishnu Kumar	8

Learning Levels of Students through Marks Obtained in 2nd Mid Term Exam (2022-23)

B.Tech. V Semester, Section-B

Subject: Design of Machine Elements-I



List of weak Students (Below 50% Marks) in 1st Mid term

Sr No.	Roll	Name	Ist Mid-term (20)
1	20ESKME049	Kavita Sharma	5
2	20ESKME061	Navneet Sagar	7
3	20ESKME066	Parth Bhatt	3
4	20ESKME081	Shailendra Chauhan	4
5	20ESKME082	Shivam	7

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Planning for Remedial Classes for Average/Below Average Students

SWAMI KESHVANAND INSTITUTE OF TECHNOLOGY, MANAGEMENT, AND GRAMOTHAN
DEPARTMENT OF MECHANICAL ENGINEERING

Notice

Date: April 21, 2022

Faculty members are hereby informed to take remedial classes as per the given schedule for the identified weak students. A proper record is to be maintained in this regard at Dy. Head office.

Time Table (Remedial Classes)

Period / Day	IV Sem 2:30PM -3:30PM	VI Sem 2:30 PM – 3:30 PM	VIII Sem 02:30 PM-3:30PM
Date of Commencement	May 15, 2022	May 02, 2022	April 25, 2022
Monday	DA (Namita Soni) ME 101	DME-II (Dr. Prem Singh) ME 102	EM (Dinesh Kr. Sharma) Me 103
Tuesday	DE (Pramod Jain) ME 101	RAC (Nikhil Sharma) ME 102	SOM (Dr. Achin Srivastava) ME 103
Wednesday	TOM (Sandeep Kr. Bhaskar) ME 101	MM (Yogesh Sharma) ME 102	****
Thursday	MP (Sushil Surana) ME 101	QM (Monika Khurana) ME 102	****
Friday	FMM (Brij Mohan Sharma) ME 101	CIMS (Sunil Kumar) ME 102	****
Saturday	TC (Dr. Shikha Agarwal) ME 101	MV (Md. Suhaib Ansari) ME 102	****

Dheeraj Joshi 21/4/2022
Prof. Dheeraj Joshi
Head, Mech. Engg. Deptt.

Copy to:

- *ME Notice Board*
- *All faculty members, ME Deptt.*

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SWAMI KESHVANAND INSTITUTE OF TECHNOLOGY, MANAGEMENT, AND GRAMOTHAN
DEPARTMENT OF MECHANICAL ENGINEERING

Notice

Date: Nov 11, 2021

Sub: Remedial Classes Time Table (Odd Semester, 2021-22)

Faculty members are hereby informed to take remedial classes as per the given schedule for the identified weak students with effect from Nov 15, 2021. A proper record of student attendance is to be maintained in this regard.

Time Table (Remedial Classes)

Period / Day	III Sem 2:30PM -3:30PM	V Sem 2:30 PM – 3:30 PM	VII Sem 12:30 PM-1:30PM
Monday	MS (Namita Soni) ME 101	DME-I (Dr. Prem Singh) ME 102	****
Tuesday	EM (Dr. Manu Augustine) ME 101	HT (Nikhil Sharma) ME 102	****
Wednesday	ET (Dinesh K Sharma) ME 101	AE (Yogesh Sharma) ME 102	****
Thursday	MOS (Md. Suhaib) ME 101	MT (Vinay S Marwal) ME 102	****
Friday	MEFA (Nitin Goyal) ME 101	POM (Madhukar Kumar) ME 102	ICE (Dr. Ashish Nayyar) Me 103
Saturday	AEM (Dr. Jyoti Arora) ME 101	MECHA (Sudesh Garg) ME 102	NDET (Vikash Gautam) ME 103

11/11/2021.
Prof. Dheeraj Joshi
Head, Mech. Engg. Deptt.

Copy to:

- *Deptt. Notice Boards*
- *All ME faculty members*

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Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur
Department of Mechanical Engineering

Date: 23.10.2020

NOTICE

EXTRA CLASSES SCHEDULE

V Semester, Session: 2020-21

Date of Commencement: 2nd November, 2020

DAY	TIME	COURSE	FACULTY
MONDAYS	3:00-4:00 PM	5ME4-02: Heat Transfer	Mr. Nikhil Kumar Sharma /Mr. Suresh Choudhary
TUESDAYS	3:00-4:00 PM	5ME4-02: Heat Transfer	Mr. Nikhil Kumar Sharma /Mr. Suresh Choudhary
WEDNESDAYS	3:00-4:00 PM	5ME4-04: Design of Machine Elements	Dr. Prem Singh/ Mr. Trivendra Kumar Sharma
THURSDAYS	3:00-4:00 PM	5ME4-04: Design of Machine Elements	Dr. Prem Singh/ Mr. Trivendra Kumar Sharma

Note:

1. **These classes are primarily intended to help the students to learn the concepts of the course (already taught in regular classes) and clear their doubts, if any.**
2. The mode of these classes is 'online' till physical classroom teaching is not possible due to COVID pandemic.
3. Faculty members can also utilize these classes for focussing on numerical or application part of the course (based on RTU question papers) in a more detailed manner.
4. Attendance of these classes have to be properly recorded and to be submitted at the office of Dy. Head periodically.

Dr. Dheeraj Joshi
Head, Mech. Engg. Deptt.

Copy to:

- *All faculty members (ME Deptt.)*
- *Deptt. Notice Board*

Teaching-Learning Methodology

- Power point presentation and classes through chalk and board
- Dissemination of notes and PPT through Google classroom and email
- Dissemination of video lectures through ERP system
- Assignments and quizzes through Google classroom and email

RTU Papers (Previous Years)

0219751

5E1324

Roll No. ____________ [Total No. of Pages : 4

5E1324

B.Tech. V Semester (Main) Examination, Nov. - 2019
PCC/PEC Mechanical Engg.
5ME4-04 Design of Machine Elements I
Common For ME, AE

Time : 3 Hours **Maximum Marks : 120**
Min. Passing Marks : 42

Instructions to Candidates:

*Attempt **all ten** questions from **Part A**, **five** questions out of **Seven** from **Part B** and **Four** questions out of **Five** from **Part C**.*

Schematic diagrams must be shown wherever necessary. Any data you feel missing suitably be assumed and stated clearly. Units of quantities used/calculated must be stated clearly.

Part - A

(Answer should be given up to **25** words only)

All questions are **compulsory** **(10×2=20)**

1. What are the economic aspects form the selection of material?
2. What is meant by Aesthetic consideration in design?
3. Define the term interchange ability and standardization.
4. Which theories of failure are applicable for shaft?
5. What are the methods of reducing stress concentration?
6. Define the terms lever and the displacement ratio.
7. What is the critical speed of shaft?
8. What is the purpose of rubber bush in pin type flexible coupling?
9. What is meant by bolts of uniform strength?
10. How will you designate the ISO metric coarse thread?

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Part - B

(Analytical/Problem solving questions)

Attempt any **five** questions (5×8=40)

1. Explain the various mechanical properties of engineering materials.

2. Explain the design considerations of casting process used in manufacturing?

3. Design the cotter against the failure under bending and express the bending stress induced.

4. The standard cross section for a flat key, which is fitted on a 50 mm diameter shaft, is 16×10mm. The key is transmitting 475 N-m torque from the shaft to the hub. The key is made of commercial steel ($S_{yt} = S_{yc} = 230$ N/mm²). Determine the length of the key, if the FOS is 3.

5. A laminated leaf spring is to carry a load of 3400 N with a deflection of about 31 mm. The spring must be supported at ends, the distance between the supports being 650 mm and is loaded at the centre. Allow a maximum stress of 420 N/mm². Take $E = 2\times10^5$. Find

   a) The stress which will be induced if the load comes down with a shock, deflecting the spring 75 mm.

   b) The magnitude of impact energy which the spring will absorb in this case.

6. The cylinder of a stationary engine is 0.12 m in diameter and is held to the crank case by M12×1.75 c, nickel steel bolts having core diameter 9.853 mm. the maximum gas pressure in the cylinder is 3.5 N/mm². Assume the ultimate strength of this steel to be 800 N/mm² and the yield stress to be 600 N/mm². Determine the number of bolts required. Take FOS = 2.

7. What is self - locking of the power screw? What is the condition for self locking?

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Part - C

(Descriptive/Analytical/Problem Solving/Design Questions)

Attempt any **Four** questions **(4×15=60)**

1. It is required to design a knuckle joint to connect two circular rods subjected to an axial tensile force of 50 kN. The rods are coaxial and a small amount of angular movement between their axes is permissible. On strength basis, the material of two rods and pin is selected as plain carbon steel of Grade 30C8 (S_{yt} = 400 N/mm^2), a higher FOS of 5 is assumed in present design. Design the joint and specify the dimensions of its component.

2. A right angled bell - crank lever is to designed to raise a load of 5 kN at the short arm end. The length of short and long arm is 100 and 450 mm respectively. The lever and pins are made of steel 30C8 (S_{yt} = 400 N/mm^2) and the FOS is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has rectangular cross - section and ratio of width to thickness is 1.25:1. Calculate

   i) Diameter and length of fulcrum pin;

   ii) Shear stress in the pin

   iii) Dimensions of the boss of the lever at the fulcrum

   iv) Dimension of the cross section of the lever

   Assume that the arm of bending moment on the lever extend up to the axis of the fulcrum.

3. Explain the designing of shaft according to A.S.M.E. code.

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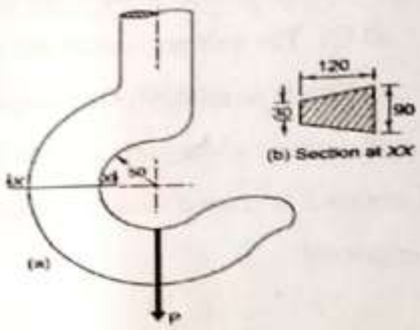
4. A crane hook having an approximate trapezoidal cross section is shown in figure. It is made of plain carbon steel 45C8 (S_{yt} = 380 N/mm^2) and factor of safety is 3.5. Determine the load carrying capacity of the hook.



5. Write a short notes on

   i. Ergonomics

   ii. Allowable stress

   iii. Stiffness of spring

   iv. Beam Column

xxxxxxx

Solution of DME-I:5ME4-04 (RTU-2019)

Q-3

Sol^n Interchangeability -

The term interchangeability is normally employed for the mass production of identical items within the prescribed limits of sizes.

In order to control the size of the finished part with due allowance for error for interchangeable parts is called limit system.

Standardization - Standard products can be manufactured on a mass scale and their production cost can be kept minimum. Standardization leads to cheaper and easier procurement and cost of replacement can also be reduced.

Q.4
Sol^n. 1) Maximum shear stress theory or Guest's theory - ductile material

2) Maximum normal stress theory or (Rankine Theory) - brittle material

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

2) $\sigma_{b\,max} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$

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Q-5 Methods of reducing stress concentration:-

Sol^n
1) Provide a fillet radius so that the cross section may change gradually.
2) Some times an elliptical fillet is also used.
3) If notch is unavoidable, it is better to provide a number of ~~sm~~ small notches rather than a long one.
4) If a projection is unavoidable from design consideration it is preferable to provide a narrow notch than a wide notch.
5) stress relieving grooves are some times provided.

Q-6

Sol^n Lever :- A lever is a rigid rod or bar ~~capat~~ capable of turning about a fixed point called fulcrum. It is used by a machine to ~~lift~~ lift a load by the application of a small effort.

Displacement Ratio :- The ratio of effort arm to load arm, i.e. $\left(\frac{l_1}{l_2}\right)$ is called the leverage.

Q-7

Sol^n **Critical speed of shaft :-**

The critical speed is also known as the whirling speed. A shaft achieves critical speed when the natural frequency becomes equal to applied frequency. When shaft rotates at critical speed it possess heavy vibrations.

Q-8

Sol^n The rubber or leather bushes are used over the pin. The rubber bushes absorbs shocks and vibration during its operations.

Q-9

Sol^n **Bolts of uniform strength :-** The bolt, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts of uniform strength.

Q-10 **ISO metric coarse thread :-**

Sol^n According to Indian standards the complete designation of the screw thread shall include

1) Size designation — The size of the ~~coarse~~ screw is designated by the letter M followed by the diameter and pitch
2) Tolerance designation — 7 for fine, 8 for normal, 9 for coarse grade.

← PART – B →

Solution

Q-1

Sol^n Mechanical properties of Engineering materials:-

materials are characterized by their properties. They may be hard, ductile or heavy. They may be soft, brittle or light. The mechanical properties of the materials are the properties that describe the behaviour of the material under the action of external forces.
The important mechanical properties of materials are following.

1) Strength :- strength is defined as the ability of the material to resist, without rupture, external forces causing various types of stresses.

2) Elasticity : Elasticity is defined as the ability of the material to regain its original shape and size after the deformation, when the external forces are removed.

3) plasticity :- plasticity is defined as the ability of material to retain the deformation produced under the load on permanent basis

(4) Stiffness :- Stiffness or rigidity is defined as the ability of the materials to resist deformation under the action of external load.

(5) Resilience :- Resilience is defined as the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded.

(6) Toughness :- Toughness is defined as the ability of the material to absorb energy before fracture takes place.

7) Malleability :- is defined as the ability of a material to deform to a greater extent before the sign of crack, when it is subjected to compressive load.

8) Dutility

9) Brittleness

10) Hardness

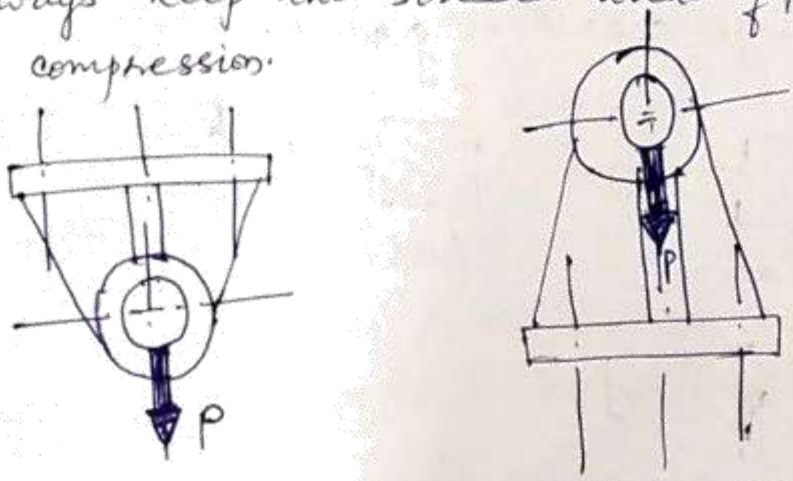
4)

Q-2

Soln DESIGN CONSIDERATION OF CASTING:-

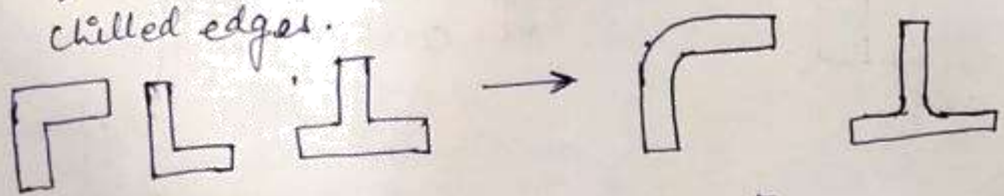
The general principles of for the design of casting are as follows.

⇒ Always keep the stressed areas of the part in compression.



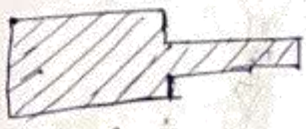
(a) Incorrect (b) Correct.

⇒ Round All the External corners:- It has two advantages - it increases the endurance limit of the component and reduce the formation of brittle chilled edges.



Incorrect Correct

⇒ Wherever possible, the section Thickness throughout should be held as uniform as compatible with overall Design consideration:- Abrupt changes in the Cross section result in high stress concentration. If the thickness is to be varied at all the change should be gradual.



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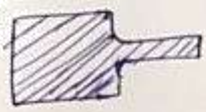
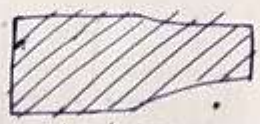
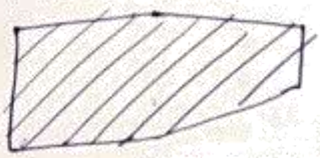


Good

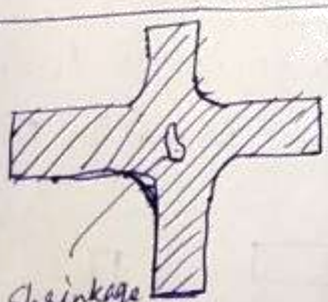


better

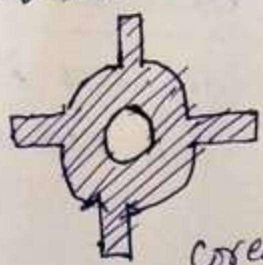


best

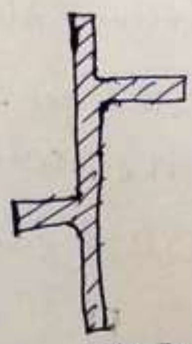
⇒ Avoid Concentration of metal at the junctions:-



Shrinkage Cavity



Cored holes.



Staggered Ribs.

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⇒ Avoid very Thin sections:-

⇒ Shot Blast the parts wherever possible:-

Q-3

Ans. Bending Failure of cotter:-

When the cotter is tight in the socket and spigot, it is subjected to shear stress, when it becomes loose, bending occurs. The forces acting on the cotter are shown in fig. The force P between the cotter and spigot end is assumed as uniformly distributed over the length d2, The force between the socket end and cotter is assumed to be varying linearly from zero to max. with triangular distribution. The cotter is treated as beam as shown in fig!

For Triangular distribution, $x = \frac{1}{3}y = \frac{1}{3}\left(\frac{d_4-d_2}{2}\right) = \left(\frac{d_4-d_2}{6}\right)$

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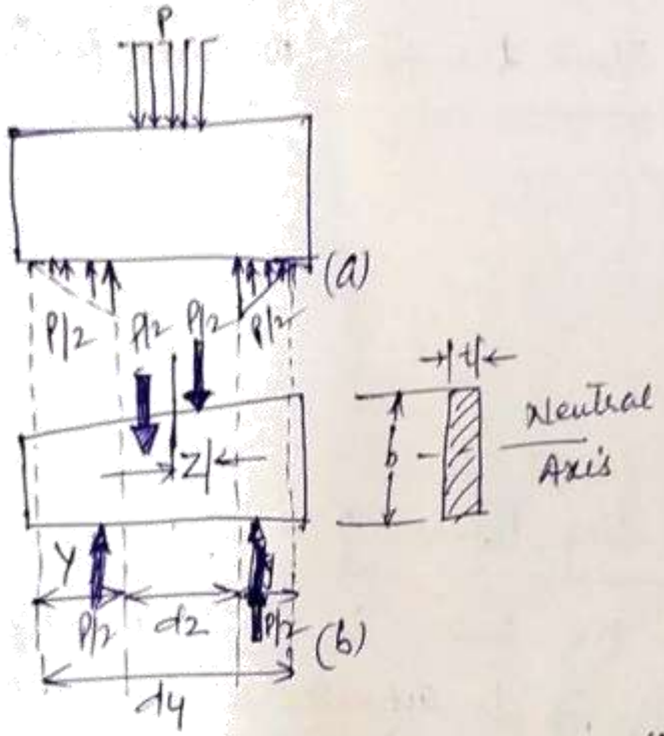

Cotter ~~to~~ treated as Beam. (a) Actual distribution of forces (b) Simplified Diagram of Forces.

The bending moment is max. at the centre, At the central section

$$M_b = \frac{P}{2}\left[\frac{d_2}{2} + x\right] - \frac{P}{2}(z)$$

$$= \frac{P}{2}\left[\frac{d_2}{2} + \frac{d_4-d_2}{6}\right] - \frac{P}{2}\left[\frac{d_2}{4}\right]$$

$$\boxed{M_b = \frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4-d_2}{6}\right]}$$

Also $I = \frac{tb^3}{12}$

$y = \frac{b}{2}$

Therefore :- $\sigma_b = \dfrac{\frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4-d_2}{6}\right] b/2}{tb^3/12}$ and $\sigma_b = \frac{M_b y}{I}$

Ans

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Q-4

Sol^n Given :- Torque $(M_t) = 475 \cdot N\text{-}m$, $S_{yt} = S_{yc} = 230 N/mm^2$

FOS = 3, $d = 50 mm$, $b = 16 mm$, $h = 10 mm$.

Step I :- Permissible compressive and shear stresses :-

$$\sigma_c = \frac{S_{yc}}{fos} = \frac{230}{3} = 76.67 N/mm^2$$

According to max. shear stress theory of failure.

$$S_{sy} = 0.5 S_{yt} = 0.5 (230) = 115 N/mm^2$$

$$t = \frac{S_{sy}}{fos} = \frac{115}{3} = 38.33 N/mm^2$$

Step II :- Key length

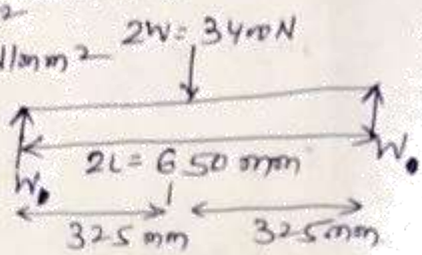
$$l = \frac{2 M_t}{t d b} = \frac{2(475 \times 10^3)}{(38.33)(50)(16)} = \underline{\underline{30.98 mm}} \quad \text{—(i)}$$

$$l = \frac{4 M_t}{\sigma_c d h} = \frac{4(475 \times 10^3)}{(76.67) \times (50) \times (10)} = \underline{\underline{49.56 mm}} \quad \text{—(ii)}$$

From (i) and (ii), the length of the key should be 50mm. Ans.

Q- 5

Sol^n Given:- $2W = 3400N$, $W_0 = \frac{3400}{2} = 1700N$, $\delta = 31mm$,
$2L = 650mm$, $L = 325mm$, $\sigma = 420 N/mm^2$
$E = 2 \times 10^5$.



We know that

$$\sigma = \frac{6WL}{bt^2} \Rightarrow 420 = \frac{6 \times 1700 \times 325}{bt^2} \quad —(i)$$

also,

$$\delta = \frac{4WL^3}{E\,bt^3} \Rightarrow 31 = \frac{4 \times 1700 \times (325)^3}{2 \times 10^5 \times bt^3} \quad —(ii)$$

from eq^n (i) and (ii),

$$\frac{bt^3}{bt^2} = \frac{4 \times 1700 \times (325)^3}{2 \times 10^5 \times 31} \times \frac{420}{6 \times 1700 \times 325}$$

$$t = 4.77 \text{ mm say } 5 \text{ mm}$$

$$\boxed{t = 5mm}$$

from eq^n (i) by putting value of t,

$$420 = \frac{6 \times 1700 \times 325}{b \times (5)^2} \Rightarrow b = \frac{6 \times 1700 \times 325}{420 \times 25} = 315.71 \text{ mm}$$

from eq^n-(ii)

$$31 = \frac{4 \times 1700 \times (325)^3}{2 \times 10^5 \times b \times (5)^3} \Rightarrow b = 301.20mm$$

Sol^n(a)

$$\delta = \frac{4WL^3}{Ebt^3} \Rightarrow 75 = \frac{4 \times W \times (325)^3}{2 \times 10^5 \times (5)^3 \times 315.71} \Rightarrow W = 4311 N \text{ Ans.}$$

$$\sigma = \frac{6 \times 4311 \times 325}{315.71 \times (5)^2} = 213.01 N/mm^2$$

$\sigma = 214 N/$ Ans.

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(b)

Q-6

Soln energy aborb by the spring

$U = \frac{1}{2} W.\delta \Rightarrow \frac{1}{2} \times 4311 \times 75 = 1.61 \times 10^5$ N-mm Ans.

Q-6

Soln Given:- D = 0.12 m = 120 mm. d_c = 9.853 mm.

p = 3.5 N/mm², S_{ut} = 800 N/mm², S_{ys} = 600 N/mm²

FoS = 2

We know that,

$$Fos = \frac{\cancel{S_{ys}}}{} \frac{\cancel{\text{working stress}}}{} = \frac{S_{ys}}{S_{ws}}$$

$$S_{ws} = \frac{600}{2} = 300 \text{ N/mm}^2$$ where (∵ S_{ws} = working stress),

We know that the pressure acting on the cylinder case is

$$P = \frac{\pi}{4} \times D^2 \times p \Rightarrow \frac{\pi}{4} \times (120)^2 \times 3.5 = 39564 \text{ N} \quad \text{--- (i)}$$

Resisting force offered by n. ~~nubee~~ number of bolts

$$P = \frac{\pi}{4} \times (d_c)^2 \times S_{ws} \times n \Rightarrow 39564 = \frac{\pi}{4} \times (9.853)^2 \times 300 \times n$$

$$n = \frac{39564}{22862} = 1.73 \text{ or say } 2 \text{ bolts}$$ Ans

$\boxed{n = 2}$ Ans.

Taking the diameter of the bolt hole (d_1) as 13 mm, we have pitch circle diameter of bolts. assuming t = 10 mm

$$D_p = D + 2t + 3d_1 \Rightarrow 120 + 2 \times 10 + 3 \times 13 = 179 \text{ mm}$$

$$\boxed{\therefore D_p = 179 \text{ mm}}$$

$\therefore$ Circumferential pitch of the bolts is

$$= \pi \times \frac{D_p}{n} = \frac{\pi \times 179}{2} = 281.17 \text{ mm}$$

we know that for a leak proof joint. the circumferential pitch of the ~~bole~~ bolts should lie between $20\sqrt{d_1}$ to $30\sqrt{d_1}$, where d_1 is the diameter of the bolt hole in mm:

$\therefore$ Minimum circumferential pitch of the bolts.

$$= 20\sqrt{d_1} = 20\sqrt{13} = 72.11 \text{ mm.}$$

& Max. circumferential pitch of the bolts —

$$30\sqrt{d_1} = 30\sqrt{13} = 108 \text{ mm}$$

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PART- B

Q-7

Sol^n Self locking of power screws :-

we know that the effort required to lower the load is

$$P = W \tan(\phi - \alpha)$$

and the torque required to lower the load

$$T = P \times \frac{d}{2} = W \tan(\phi - \alpha) \frac{d}{2}$$

"A screw will be self locking if the friction angle is greater than helix angle or coefficient of friction is greater than tangent of helix angle i.e. μ or $\tan\phi > \tan\alpha$."

$$(\phi > \alpha)$$

Efficiency of self locking screws :-

we know that efficiency of screw

$\eta = \frac{\tan\phi}{\tan(\alpha+\phi)}$ and for self locking screws $\phi \geq \alpha$ or $\alpha \leq \phi$

$\therefore$ Efficiency for self locking screws,

$$\eta \leq \frac{\tan\phi}{\tan(\phi+\alpha)} \leq \frac{\tan\phi}{\tan 2\phi} \leq \frac{\tan\phi(1-\tan^2\phi)}{2\tan\phi}$$

$$\leq \frac{1}{2} - \frac{\tan^2\phi}{2}$$

From this expression we see that efficiency of self locking screws is less than $\frac{1}{2}$ or 50%. Ans.

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PART- C

Q- 1

Sol^n Given:- $P = 50 \times 10^3 N$, FOS = 5, Plain carbon steel of Grade 30C8, $S_{yt} = 400 N/mm^2$, $S_{yt} = S_{yc}$

→ Calculation of permissible stresses:—

$$\sigma_t = \frac{S_{yt}}{fos} = \frac{400}{5} = 80 N/mm^2$$

$$\sigma_c = \frac{S_{yc}}{fos} = \frac{S_{yt}}{fos} = \frac{400}{5} = N/mm^2$$

$$\tau = \frac{S_{sy}}{fos} = \frac{0.5\,S_{yt}}{fos} = \frac{0.5(400)}{5} = 40 N/mm^2$$

Calculation of dimensions:— where, D = Diameter of each rod(mm)

Diameter of Rods:-

$$D = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4(50\times10^3)}{\pi(80)}} = 28.21 \text{ or } 30mm.$$

Enlarged diameter of rods (D_1):-

$$D_1 = 1.1D = 1.1(30) = 33 \text{ or } 35 mm.$$

Dimensions a and b

$$a = 0.75D = 0.75\times30 = 22.5 \text{ or } 25mm$$
$$b = 1.25D = 1.25\times30 = 37.5 \text{ or } 40 mm.$$

Diameter of pin:

Where d = diameter of knuckle pin (mm)

$$d = \sqrt{\frac{2P}{\pi \tau}} = \sqrt{\frac{2(50\times10^3)}{\pi(40)}} = 28.21 \text{ or } 30\text{ mm}$$

Also,

$$d = \sqrt[3]{\frac{32}{\pi\sigma_b} \times \frac{P}{2}\left[\frac{b}{4}+\frac{a}{3}\right]}$$

$$= \sqrt[3]{\frac{32}{\pi(80)} \times \frac{50\times10^3}{2}\left[\frac{40}{4}+\frac{25}{3}\right]}$$

$$= 38.79 \text{ or } 40\text{ mm}$$

$\therefore$ $d = 40$ mm

Dimensions d_0 and d_1

Where, d_0 = out side diameter of eye or fork (mm)

d_1 = diameter of pin head (mm)

$$d_0 = 2d = 2(40) = 80\text{ mm}$$

$$d_1 = 1.5d = 1.5\times(40) = 60\text{ mm}$$

Check for stresses in eye.

$$\sigma_t = \frac{P}{b(d_0-d)} = \frac{50\times10^3}{40(80-40)} = 31.25\text{ N/mm}^2$$

$\therefore$ $\sigma_t < 80$ N/mm^2

$$\sigma_c = \frac{P}{bd} = \frac{50\times10^3}{40\times40} = 31.25\text{ N/mm}^2$$

$\therefore$ $\sigma_c < 80$ N/mm^2

$$\tau = \frac{P}{b(d_0-d)} = \frac{50\times10^3}{40(80-40)} = 31.25\text{ N/mm}^2$$

$\tau < 40$ N/mm^2

Check for stresses in fork:- where a = thickness of each eye of fork (mm)

$$\sigma_t = \frac{P}{2a(d_0-d)} = \frac{50\times10^3}{2(25)(80-40)} = 25\,N/mm^2$$

$\therefore \sigma_t < 80\,N/mm^2$

$$\sigma_c = \frac{P}{2ad} = \frac{50\times10^3}{2(25)\times(40)} = 25\,N/mm^2$$

$\therefore \sigma_c < 80\,N/mm^2$

$$\tau = \frac{P}{2a(d_0-d)} = \frac{50\times10^3}{2(25)(80-40)} = 25\,N/mm^2$$

$\tau < 40\,N/mm^2$

It is observed that stresses are within limits.

Q-2

Sol^n Given:- $S_{yt} = 400 N/mm^2$ fos = 5, F = 5KN

For lever, long arm = 450 mm,

Short arm = 100 mm.

For Pin $p_b = 10 N/mm^2$, $\frac{l_1}{d_1} = 1.25$, $\frac{d}{b} = 3$

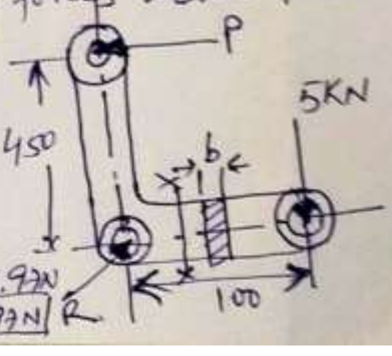
→ Calculation of permissible stresses for the pin and lever.

$$\sigma_t = \frac{S_{yt}}{fos} = \frac{400}{5} = \underline{\cancel{800}\,80 N/mm^2}$$

$$\tau = \frac{S_{sy}}{fos} = \frac{0.5 S_{yt}}{fos} = \frac{0.5 \times 400}{5} = 40 N/mm^2$$

→ Calculation of forces acting on the lever

The forces acting on the lever are shown in fig. Taking moment of forces about the axis of the fulcrum



$5 \times 10^3 \times 100 = P \times 450$

$\boxed{P = 1111.11 N}$

$R = \sqrt{(5000)^2 + (1111.11)^2} = 5121.97N$

$\boxed{R = 5121.97N}$

(i) Diameter and length of fulcrum Pin :-

Considering bearing pressure on the fulcrum pin :-

$R = p \times (\text{projected area of the pin})$

$R = p \times (d_1 \times l_1)$ ——— (1)

where, d_1 = diameter of the fulcrum pin (mm)
l_1 = length of fulcrum pin (mm)

$512.97 = 10 \times (d_1 \times 1.25 d_1)$

$\therefore \boxed{d_1 = 20.24 \text{ mm}}$ Ans.

$\boxed{l_1 = 1.25 d_1 = 1.25(20.24) = 25.30 \text{ mm}}$ Ans.

(ii) shear stress in pin

The pin is subjected to double shear. The shear stress in the pin is given by,

$$\tau = \frac{R}{2\left[\frac{\pi}{4} \times d_1^2\right]} = \frac{512.97}{2\left[\frac{\pi}{4}(20.24)^2\right]} = 7.96 \text{ N/mm}^2$$

$\boxed{\tau = 7.96 \text{ N/mm}^2}$ Ans.

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(iii) Dimensions of the boss:-

The dimensions of the boss of the lever are as follows:-

inner diameter = 21 mm. Ans.
outer diameter = 42 mm. Ans.
length = 26 mm Ans.

iv) Dimensions of the cross-section of lever:-

For the lever

$d = 3b$, $M_b = 5000 \times 100$ N-mm.

therefore

$$\sigma_b = \frac{M_b y}{I} \quad \text{or} \quad 80 = \frac{(5000 \times 100)(1.5b)}{\left[\frac{1}{12}(b)(3b)^3\right]}$$

$$\therefore \boxed{b = 16.09 \text{ mm}} \text{ Ans.}$$

$$d = 3b = 3 \times 16.09 = 48.27 \text{ mm}$$

$$\boxed{d = 48.27 \text{ mm}} \text{ Ans.}$$

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Q-3

Sol^n Designing of shaft according to ASME code.

One important approach of designing a transmission shaft is to use the ASME Code. According to this code, the permissible shear stress τ_{max} for the shaft without keyways is taken as 30% of yield strength in tension or 18% of the ultimate tensile strength of the material, whichever is minimum. Therefore,

$$\tau_{max.} = 0.30\, S_{yt}$$
$$\text{or, } \tau_{max} = 0.18\, S_{ut} \quad \text{(whichever is minimum)} \quad \text{---(i)}$$

If keyways are present, the above values are to be reduced by 25 percent. According to the ASME code, the bending and torsional moments are to be multiplied by factors K_b and K_t respectively, to account for shock and fatigue in operating condition. The ASME code is based on max. shear stress theory. Therefore,

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \quad \text{---(ii)}$$

Q.(4)

Solution – given $S_{yt} = 380\,MPa$ $f(s) = 3.5$, $b_i = 90\,mm$

$b_o = 30\,mm$, $h = 120\,mm$, $R_o = 170\,mm$

$R_i = 50\,mm$

permissible tensile stress :-

$$\sigma_{max} = \frac{S_{yt}}{f(s)} = \frac{380}{3.5} =$$

$$\sigma_{max} = 108.57\ N/mm^2$$

eccentricity (e) for the cross sec. X-X

$$R_N = \frac{\left(\frac{b_i + b_o}{2}\right) h}{\left(\frac{b_i R_o - b_o R_i}{h}\right) \log_e\left(\frac{R_o}{R_i}\right) - (b_i - b_o)}$$

$$R_N = \frac{\left(\frac{90+30}{2}\right)(120)}{\left(\frac{90 \times 170 - 30 \times 50}{120}\right) \log_e\left(\frac{170}{50}\right) - (90-30)}$$

$$R_N = \underline{89.1816\ mm}$$

$$R = R_i + \frac{h(b_i + 2b_o)}{3(b_i + b_o)}$$

$$= 50 + \frac{120(90 + 2 \times 30)}{3(90+30)} = \underline{100\,mm}$$

$$e = R - R_N = 100 - 89.1816$$

$$\underline{e = 10.8184\ mm}$$

Bending stress –

$$h_i = R_N - R_i = 89.1816 - 50$$

$$h_i = 39.1816 \text{ mm}$$

$$A = \frac{1}{2}[h(b_i + b_o)] = \frac{1}{2}[120(90+30)]$$

$$A = 7200 \text{ mm}^2$$

$$M_b = PR = (100P) \text{ N-mm}$$

$$\sigma_{bi} = \frac{M_b \cdot h_i}{A e R_i} = \frac{100P \times 39.1816}{7200 \times 10.8184 \times 50}$$

$$\sigma_{bi} = \frac{(7.2435)P}{(7200)} \text{ N/mm}^2$$

Direct tensile stress –

$$\sigma_t = \frac{P}{A} = \frac{P}{7200} \text{ N/mm}^2$$

Load carrying capacity –

$$\sigma_{bi} + \sigma_t = \sigma_{max}.$$

$$\frac{(7.2435)P}{7200} + \frac{P}{7200} = 108.57$$

~~P = 9004.80~~

$$P = 94827.95 \text{ N}$$

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Q 5.

Soln (i) Ergonomics :- Ergonomics is defined as the relationship between man and machine and the application of anatomical, physiological and psychological principles to solve the problems arising from man-machine relationship. The word ergonomics is coined from two Greek words - 'ergon' which means 'work' and 'nomos' which means 'natural laws'. Ergonomics means natural laws of work.

The aim of ergonomics is to reduce the operational difficulties present in man-machine joint system and thereby reduce the resulting physical and mental stresses.

The shape and dimensions of certain machine elements like levers, cranks and handwheels are decided on the basis of ergonomics studies.

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Sol^n (ii) Allowable stress:- The allowable stress is the stress value which is used in design to determine the dimensions of the component. It is considered as a stress, which the designer expects ~~we~~ will not be exceeded under normal operating conditions. For ductile material the allowable stress σ is obtained by the following relationship

$$\boxed{\sigma = \frac{S_{yt}}{fos}}$$

For brittle materials,

$$\boxed{\sigma = \frac{S_{ut}}{fos}}$$

where S_{yt} and S_{ut} are the yield strength and the ultimate tensile strength of the material respectively

Sol^n (iii) Stiffness of spring(k):- The stiffness of the spring (k) is defined as the force required to produce unit deflection. Therefore,

$$\boxed{K = \frac{P}{\delta}}$$

where, K = stiffness (N/mm)
P = axial spring force (N)
δ = axial deflection (mm)

There are various names for stiffness of spring such as rate of spring, gradient of spring, scale of spring, or simple spring constant. It represents the slope of the load-deflection line

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4E 4143

Roll No. ____________ [Total No. of Pages : 3

4E 4143

B.Tech. IV Semester (Main/Back) Examination, May - 2018
Mechanical Engg.
4ME4 Design of Machine Elements - I
AE,ME,PI

Time : 3 Hours **Maximum Marks : 80**
Min. Passing Marks : 26

Attempt any ***five questions****, selecting* ***one question*** *from* ***each unit****. All Questions carry* ***equal*** *marks. Schematic diagrams must be shown wherever necessary. Any data you feel missing suitably be assumed and stated clearly. Units of quantities used/calculated must be stated clearly.*

Unit - I

1. a) Discuss the following mechanical properties of the material: **(4)**
   i) Hardness
   ii) Toughness
   iii) Creep Strength
   iv) Fatigue strength

   b) Discuss fits, types of fit and tolerance? Also discuss the shaft basis and hole basis system. http://www.rtuonline.com **(12)**

OR

1. Write short note on followings: **(16)**
   a) Design consideration for casting.
   b) Material selection.
   c) Standardization and interchangeability.
   d) BIS designation of Plain carbon steel.

Unit - II

2. a) Why gibs are used in a cotter joint? **(4)**

b) Describe the design procedure of a gib and cotter joint with the help of neat sketch the use of single and double gib. **(12)**

OR

2. Design a knuckle joint for a tie rod of a circular section to sustain a maximum pull of 70 kN. The ultimate tensile strength of the material of the tearing is 420 MPa. The ultimate tensile and shearing strength of the pin material are 510 MPa and 396 MPa respectively. Determine the tie rod section and pin section. Take factor of safety = 6. **(16)**

Unit - III

3. a) What is lever? Discuss the first, second and third type of levers with neat sketch. **(4)**

b) A cranked lever has the following dimensions:

Length of the handle = 300 mm

Length of the lever arm = 400 mm

Overhang of the journal = 100 mm

If the lever is operated by a single person exerting a maximum force of 400 N at a distance of 1/3rd length of the handle from its free end, find:

i) Diameter of the handle,

ii) Cross-section of the lever arm, and

iii) Diameter of the journal.

The permissible bending stress for the lever material may be taken as 50 MPa and shear stress for shaft material as 40 MPa. **(12)**

OR

3. a) Discuss the nipping and camber in the leaf spring. **(4)**

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

Unit - IV

4. A shaft is supported by two bearings placed 1 m apart. A 600 mm diameter pulley is mounted at a distance of 300 mm to the right of left hand bearing and this drives a pulley directly below it with the help of belt having maximum tension of 2.25 kN. Another pulley 400 mm diameter is placed 200 mm to the left of right hand bearing and is driven with the help of electric motor and belt, which is placed horizontally to the right. The angle of contact for both the pulleys is 180° and $\mu = 0.24$. Determine the suitable diameter for a solid shaft, allowing working stress of 63 MPa in tension and 42 MPa in shear for the material of shaft. Assume that the torque on one pulley is equal to that on the other pulley. **(16)**

OR

4. A shaft is supported on bearings A and B, 800 mm between centres. A 20° straight tooth spur gear having 600 mm pitch diameter, is located 200 mm to the right of the left hand bearing A, and a 700 mm diameter pulley is mounted 250 mm towards the left of bearing B. The gear is driven by a pinion with a downward tangential force while the pulley drives a horizontal belt having 180° angle of wrap. The pulley also serves as a flywheel and weighs 2000 N. The maximum belt tension is 3000 N and the tension ratio is 3 : 1. Determine the maximum bending moment and the necessary shaft diameter if the allowable shear stress of the material is 40 MPa. **(16)**

Unit - V

5. a) Discuss screw the initial stresses developed in fastening due to screwing up forces. http://www.rtuonline.com **(4)**

b) The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm². It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa. **(12)**

OR

5. A steam engine of effective diameter 300 mm is subjected to a steam pressure of 1.5 N/mm². The cylinder head is connected by 8 bolts having yield point 330 MPa and endurance limit at 240 MPa. The bolts are tightened with an initial preload of 1.5 times the steam load. A soft copper gasket is used to make the joint leak-proof. Assuming a factor of safety 2, find the size of bolt required. The stiffness factor for copper gasket may be taken as 0.5. **(16)**

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4E4143

Roll No. 15EGJME146

Total No. of Pages : 4

4E4143

B. Tech. IV-Sem. (Main) Exam; April-May 2017

Production & Industrial Engg.

4PI4A Design of Machines Elements - I

Time : 3 Hours

Maximum Marks : 80
Min. Passing Marks : 24

Instructions to Candidates :-

Attempt any five questions, selecting one question from each unit. All Questions carry equal marks. Schematic diagrams must be shown wherever necessary. Any data you feel missing suitably be assumed and stated clearly. Units of quantities used / calculated must be stated clearly. Use of following supporting materials is permitted during examination. (Mentioned in form No. 205)

1. NIL 2. NIL

UNIT - I

1 (a) What is a machine element ? Give two examples. 4

(b) What do you understand by mechanical properties of materials ? How these are helpful in machine design ? 12

OR

1 (a) What is 'machine design' ? Explain the basic procedure of machine design. 8

(b) Explain standardization, limits, fits and surface roughness for manufacturing consideration in design with neat sketch and suitable examples. 8

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UNIT - II

2 (a) What is 'Stress concentration' ? How it can be reduced in a component ? 8

(b) Determine the diameter of a circular rod made of ductile material with a fatigue strength (complete stress reversal) $\sigma_c = 265$ MPa and a tensile yield strength of 350 MPa. The member is subjected to a variable axial load from $W_{min} = -300 \times 10^3$ N to $W_{max} = 700 \times 10^3$ N and has a stress concentration factor = 1.8. Use factor of safety as 2.0. 8

OR

2 It is required to design a cotter joint to connect two steel rods of equal diameters. Each rod is subjected to an axial tensile force of 50 kN. Design the joint and specify its main dimensions. 16

UNIT - III

3 (a) What is a 'beam' ? Which type of stresses can be induced in it ? Discuss the role of section modulus in beams design with two examples of different shapes. 8

(b) A truck spring has 12 number of leaves, two of which are full length leaves. The spring supports are 1.05 m and the central band is 85 mm wide. The central load is to be 5.4 kN with a permissible stress of 280 MPa. Determine the thickness and width of the steel spring leaves. The ratio of the total depth to the width of the spring is 3. Also determine the deflection of the spring. 8

OR

3 (a) What is a 'lever' ? Explain the principle of it and leverage. Classify the levers. 8

$t^2 \times 4t = 2326.4$

4E4143] [P.T.O.

(b) A right angled bell-crank lever is to designed to raise a load of 5 kN at the short arm end. The lengths of short and long arms are 100 and 450 mm respectively. The lever and the pins are made of steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has rectangular cross-section and the ratio of width to thickness is 3 : 1. The length to diameter ratio of fulcrum pin is 1.25 : 1

Calculate :

page 123
4116

(i) The diameter and the length of fulcrum pin
(ii) The shear stress in the pin
(iii) The dimensions of the boss of the lever of the fulcrum and
(iv) The dimensions of the cross-section of the lever.

Assume that the arm of bending moment on the lever expands upto the axis of the fulcrum. **8**

UNIT - IV

4 (a) A line shaft transmits 25 kW power at 200 rpm by means of a vertical belt drive. The diameter of the belt pulley is 1 m and the pulley overhangs 150 mm beyond the centre line of the end bearing. The belt tension acts vertically downward. The tension on the tight side of the belt is 2.5 times that on slack side. The shaft is made of plain carbon steel 40C8 (S_{yt} = 380 N/mm^2) and the factor of safety is 2.5. The mass of the pulley is 25 kg. Determine the diameter of the shaft. **12**

(b) What is a 'key' ? Explain the failure of key. **4**

OR

4 (a) What is coupling ? Classify it. **4**

(b) Design a muff coupling which is used to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 (S_{yt} = S_{yc} = 400 N/mm^2). The sleeve is made of grey cast iron FG 200 (S_{ut} = 200 N/mm^2). The factor of safety for the shaft and key is 4. For the sleeve, the factor of safety is 6 based on ultimate strength. **12**

DDHB for Key

[P.T.O.

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UNIT - V

5 (a) Explain the concept of thread for single start and double start, relative to lead of them. Explain the terminologies used to define the threads with neat sketches. **8**

(b) What are the 'locking devices' ? Classify them and explain their working concept with neat sketches. **8**

OR

5 (a) Why uniform strength is required in bolts ? How it can be achieved ? Determine the diameter of the hole that must be drilled in a M48 bolt such that the bolt becomes of uniform strength. **8**

(b) A bracket, as shown in Fig. supports a load of 30 kN. Determine the size of bolts, if the maximum allowable tensile stress is in the bolt material is 60 MPa. the distances are L_1 = 80 mm, L_2 = 250 mm and L = 500 mm. **8**

no of bolts missing

DDHP

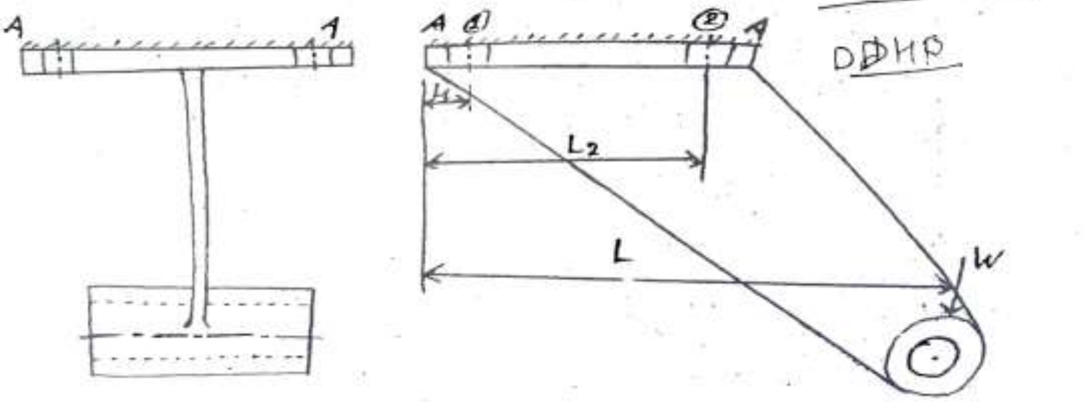

Fig. : Bracket with eccentric loading.

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4E4143

Roll No. ____________ Total No of Pages: 4

4E4143
B.Tech. IV-Sem (Main & Back) Exam; June-July 2016
Automobile Engineering
4AE4A Design of Machine Elements-I
AE, ME, PI

Time: 3 Hours **Maximum Marks: 80**
Min. Passing Marks (Main & Back): 26
Min. Passing Marks (Old Back): 24

Instructions to Candidates:-

*Attempt any **five questions**, selecting **one question** from **each unit**. All Questions carry **equal** marks. Schematic diagrams must be shown wherever necessary. Any data you feel missing suitably be assumed and stated clearly.*
Units of quantities used/ calculated must be stated clearly.
Use of following supporting material is permitted during examination.
(Mentioned in form No.205)

1. Design Data Book ________ 2. ____________________

UNIT-I

Q.1 (a) What is standardization? What are the advantages of standardization? [3+3=6]

(b) Explain in detail the design considerations of casting with neat sketches. [10]

OR

Q.1 (a) Write short note on mechanical properties of materials. [6]

(b) Explain in detail design considerations of machine parts. [10]

UNIT-II

Q.2 (a) Write the design procedure of cotter joint. [8]

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(b) It is required to design a square key for fixing a pulley on the shaft, that is 25mm diameter. The shaft is transmitting 15 kW power at 720 rpm. The key is made of steel (S_{yt} = 460 N/mm^2) and the factor of safety is 3. For key material, the yield strength in compression can be assumed to be equal to the yield strength in tension. Determine the dimension of the key. [8]

OR

Q.2 (a) What is knuckle joint? Where do you use knuckle joint? Give practical examples. [4]

(b) The stresses induced at a critical point in a machine component made of steel (Syt = 380 N (mm^2) are as follows [12]

σ x = 100N/mm^2

σ y = 40 N/mm^2

τ xy = 80 N/mm^2

Calculate the factor of safety by

(i) The maximum normal stress theory

(ii) The maximum shear stress theory

(iii) The distortion energy theory

UNIT-III

Q.3 (a) A semi – elliptic leaf spring used for automobile suspension consists of three extra full – length leaves and 15 graduated – length leaves, including the master leaf. The centre – to – centre distance between two eyes of the spring is 1m. The maximum force that can act on the spring is 75 kN. For each leaf, the ratio of width to thickness is 9:1. The modulus of elasticity of the leaf material is 207000 N/mm^2. The leaves are pre – stressed in such a way that when the force is maximum, the stresses in all leaves are same and equal to 450 N/mm^2. [12]

Determine-

(i) The width and thickness of the leaves.

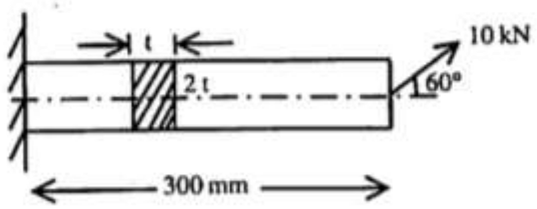
(ii) The initial nip

(iii) The initial pre – load required to close the gap 'C' between extra full – length leaves and graduated length leaves.

(b) What is the objective of shot peening of spring? [4]

OR

Q.3 (a) For a beam made of C.I. given below, determine the dimensions of the cross section.



Use followings:

$S_{ut} = 200$ N/mm^2

F.S. = 2.5

The depth of cross section is twice of the width.

Use maximum normal stress theory. [12]

(b) What are the second type of lever and third type of lever? Give their examples. [4]

UNIT-IV

Q.4 It is required to design a rigid type of flange coupling to connect two shafts. The input shaft transmits 37.5 kW power at 180 rpm to the output shaft through the coupling. [16]

Use following:

Service factor = 1.5

Allowable shear stress for shaft = 76 $\frac{N}{mm^2}$

Allowable shear stress for keys & bolt = 80 $\frac{N}{mm^2}$

Allowable cruishing stress for keys & bolt = 240 $\frac{N}{mm^2}$

Allowable shear stress for flanges = 16.67 $\frac{N}{mm^2}$

Take Number of bolts = 4

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OR

Q.4 (a) A solid shaft of diameter d is used in power transmission. Due to modification of existing system, it is required to replace the solid shaft by a hollow shaft of the same material and equally strong in torsion. Further the weight of hollow shaft per metre length should be half of the solid shaft. Determine the outer diameter of hallow shaft in terms of 'd'. [10]

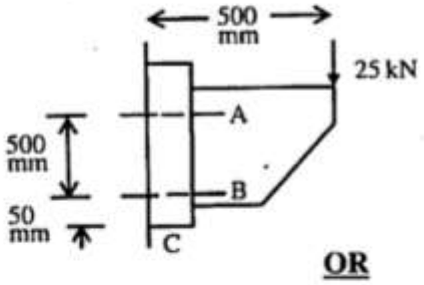
(b) Explain equivalent twisting moment for shaft. [4]

(c) What do you understand by torsional rigidity? [2]

UNIT-V

Q.5 (a) What do you mean by 'Bolt of uniform strength'? [4]

(b) A wall bracket is attached to a wall by means of four identical bolts, two at A and two at B. Assuming that the bracket is held against the wall and prevented from tipping about C by all four bolts and using an allowable tensile stress in the bolts as 35 N/mm^2. Determine the size of the bolts on the basis of maximum principal stress theory. [12]



OR

Q.5 (a) Give in detail, the design procedure of screw Jack. [10]

(b) Explain self – locking of power screw. [3]

(c) What do you mean by overhauling of power screw? [3]

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4E4143

Roll No. ______________________ [Total No. of Pages : 4

4E4143

B.Tech. IVsem(Main/Back) Examination ,June/July- 2015

Mechanical Engg

4ME4 Design of Machine Elements-I

Common with Automobile

Time : 3 Hours

Maximum Marks : 80
Min. Passing Marks : 26

Instructions to Candidates:

*Attempt any **five** questions, selecting one question from **each unit**. All questions carry equal marks. (Schematic diagrams must be shown wherever necessary. Any data you feel missing suitably be assumed and stated clearly. Units of quantities used/calculated must be stated clearly.*

Unit - I

1. a) What are the various types of fits? Explain them with the help of neat diagrams. Give an example of each. **(6)**

b) Represent the following on a suitable diagram: Upper deviation, lower deviation, Fundamental deviation, Tolerance zone, Basic size **(4)**

c) What precautions should be observed while designing a Forging. **(4)**

d) The 'cost' factor influence the material selection. Explain. **(2)**

OR

1. a) Write a note on 'Design for Assembly'. **(4)**

b) Manufacturing consideration is an important material selection criterion. Explain. **(4)**

c) The surface roughness is limited by the manufacturing method used. Explain. **(4)**

d) Explain the meaning of designation of the following steel:

i) 55 C8 ii) 16 Ni_3 Cr_2 **(4)**

Unit - II

2. a) Compare ductile and brittle failure with the help of theories of failure. **(4)**

b) Discuss the methods of stress concentration mitigation. Give suitable diagrams. **(6)**

c) What types of stresses are induced in a cotter of the cotter joint when subjected to tensile load and also, give the expression for the respective resisting area along with suitable diagrams. **(6)**

OR

2. Design a knuckle joint to connect two round rods subjected to a tensile load of 100 kN. The permissible stresses may be taken as 75 MPa in tension, 50 MPa in shear and 135 MPa in crushing. **(16)**

Unit - III

3. a) Give steps to design a cantilever beam for stiffness. **(6)**

b) What is the objective of nipping of leaf spring. **(4)**

c) How the pin-joint at eye-end in the leaf spring is designed? **(6)**

OR

3. Design a cranked lever for the following data:

Length of handle = 200 mm

Length of the lever arm = 300 mm

Overhung of the shaft from the fournal = 50 mm

Effort applied by an average person = 400 N

The shaft is also to be designed. The permissible stresses are:

Lever: $\sigma_t = 70 MPa, \tau = 50 MPa$

Shaft: $\tau = 40 MPa, \sigma_{cr} = 102 MPa$ **(16)**

Unit - IV

4. Power is transmitted by a shaft 900 mm long and is supported at the ends. A pulley of diameter 420 m is placed at 150 mm to the left of right hand bearing and another pulley of diameter 270 mm is mounted midway between the bearings. Determine the diameter of the shaft transmitting 24 kW at 300 rpm using both maximum shear stress theory and maximum normal stress theory.

The permissible tensile and shear stresses for shaft material are 120 MPa and 80 MPa respectively. The belt drives are at right angles to each other with tensions ratio as 3:1 **(16)**

OR

4. a) Design a muff coupling to transmit 6.5 kW at 1000 rpm. The permissible shear stress for shaft, key & muff is 50 MPa and permissible crushing stress for key is 120 MPa. **(10)**

b) For a rigid flange coupling (shaft diameter 'd') transmitting torque T, give design equation/procedure to calculate:

i) Flange thickness

ii) Bolt diameter **(6)**

Unit - V

5. a) Find the diameter of bolts used to connect the bracket as shown is Figure.1

Given: l=650 mm, a=100mm, b=150 mm, p=5kN The permissible shear stress is 40 MPa. **(10)**

b) What are the different types of stresses induced in the bolts due to initial tighting and give their expressions. **(6)**

OR

5. a) What is self-locking screw? How is it achieved? **(4)**

b) A U-frame, made of cast steel, has a maximum force of 70kN as shown in Figure 2. The cross-section of the frame is 125×b (rectangular). Determine the dimension 'b'. Using straight beam formulae and curved beam formula. Take stress concentration/correction factor a 1.4. The permissible tensile stress is 100 MPa. **(12)**

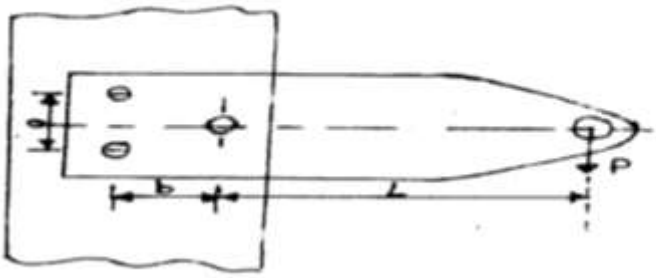

Figure-1 (Q.5 a)

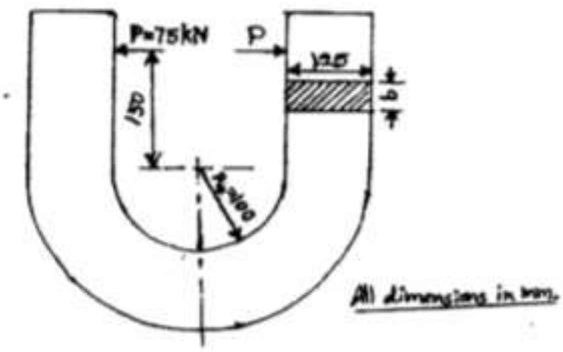

Figure -2

Mid Term Papers (Mapping with Bloom's Taxonomy & COs)

Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur

I Mid Term Examination, Nov.-2022

Semester:	**V**	**Branch:**	**ME**
Subject:	**DME-I**	**Subject Code:**	**5ME4-04**
Time:	**1.5 Hours**	**Maximum Marks:**	**20**
Session (I/II/III): II			

PART A (short-answer type questions)

(All questions are compulsory) **(3*2=6)**

Q.1 Write IS designation of: (a) Grey Cast Iron having minimum tensile strength of 220MPa (b) Plain carbon steel having average content of 0.5% carbon and 0.8% manganese.

Q.2 Define interchangeability and standardization.

Q.3 What is factor of safety?

PART B (Analytical/Problem solving questions)

(Attempt any 2 Questions) **(2*4=8)**

Q.4 Explain the design considerations of casting products with the neat sketches.

Q.5 Explain any six mechanical properties of engineering materials.

Q.6 Explain the causes of stress concentration and its methods of reduction in a component with suitable sketch.

PART C (Descriptive/Analytical/Problem solving/Design questions)

(Attempt any 1 Question) **(1*6=6)**

Q.7 Design a spigot and socket type cotter joint for axial load of 50 kN which alternately changes from tensile to compressive. Allowable stresses for the material used are 70 MPa in tensions, 50 MPa in shear, and 140 MPa in crushing.

Q.8 Design a knuckle joint to connect two round rods subjected to a tensile load of 60 kN. The permissible stresses may be taken as 80 MPa in tension, 55 MPa in shear and 100 MPa in crushing.

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I Mid Term Examination, Nov.-2022

Semester:	**V**	**Branch:**	**ME**
Subject:	**DME-I**	**Subject Code:**	**5ME4-04**

A. Distribution of Course Outcome and Bloom's Taxonomy in Question Paper

Q.No	Questions	Marks	CO	BL
1.	Write IS designation of: (a) Grey Cast Iron having minimum tensile strength of 220MPa (b) Plain carbon steel having average content of 0.5% carbon and 0.8% manganese.	2	1	1
2.	Define interchangeability and standardization	2	1	2
3.	What is factor of safety?	2	2	1
4.	Explain the design considerations of casting products with the neat sketches.	4	1	2
5.	Explain any six mechanical properties of engineering materials	4	1	2
6.	Explain the causes of stress concentration and its methods of reduction in a component with suitable sketch.	4	2	2
7.	Design a spigot and socket type cotter joint for axial load of 50 kN which alternately changes from tensile to compressive. Allowable stresses for the material used are 70 MPa in tensions, 50 MPa in shear, and 140 MPa in crushing.	6	3	3
8.	Design a knuckle joint to connect two round rods subjected to a tensile load of 60 kN. The permissible stresses may be taken as 80 MPa in tension, 55 MPa in shear and 100 MPa in crushing.	6	6	3

BL – Bloom's Taxonomy Levels

(1- Remembering, 2- Understanding, 3 – Applying, 4 – Analyzing, 5 – Evaluating, 6 - Creating)

CO – Course Outcomes

B. Questions and Course Outcomes (COs) Mapping in terms of correlation

COs	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8
CO1	2	2	-	3	3	-	-	-
CO2	-	-	2	-	-	3	3	3
CO3	-	-	-	-	-	-	-	-
CO4	-	-	-	-	-	-	-	-
CO5	-	-	-	-	-	-	-	-

1-Low Correlation; 2- Moderate Correlation; 3- Substantial Correlation

C. Mapping of Bloom's Level and Course Outcomes with Question Paper

Bloom's Level Mapping		**CO Mapping**	
Bloom's Level	Percentage	CO	Percentage
BL1	13.33	CO1	40
BL2	46.67	CO2	60
BL3	40.00	CO3	-
BL4	00	CO4	-
BL5	00	CO5	-
BL6	00		

Solution of I Mid Term Examination, Nov.-2022

PART-A

Answer 1.

(a) FG220 **(1 mark)**
(b) 50C8 **(1 mark)**

Answer 2.

Standardization is defined as obligatory norms, to which various characteristics of a product should conform. (1 mark)

Interchangeability: T*he ability that an object can be replaced by another object without affecting code using the object.* *(1 mark)*

Answer 3

Factor of safety: The factor of safety is defined as

$$fs=\frac{\text{failure stress}}{\text{allowable stress}}$$ **(2 marks)**

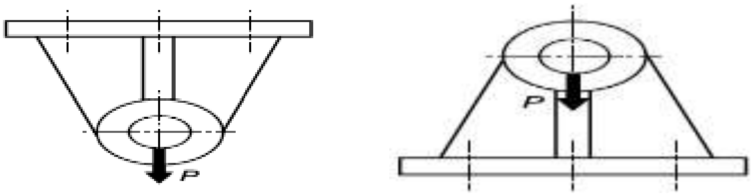
Part B

Answer 4

Design Considerations of Castings

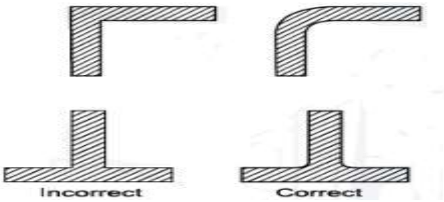
In designing a casting, the following points should always be considered.

- Always keep the stressed areas of the part in compression
- Cast iron has more compressive strength than its tensile strength.
- The castings should be placed in such a way that they are subjected to compressive rather than tensile stresses.



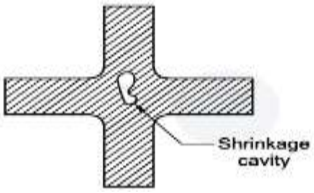
2. Round all external corners

- It has two advantages—it increases the endurance limit of the component and reduces the formation of brittle chilled edges.
- When the metal in the corner cools faster than the metal adjacent to the corner, brittle chilled edges are formed due to iron carbide.

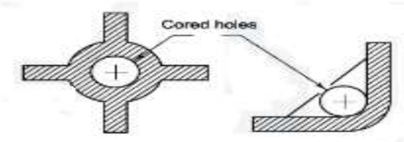


- Abrupt changes in the cross-section result in high stress concentration, so the section thickness throughout should be held as uniform.
- If the thickness is to be varied at all, the change should be gradual

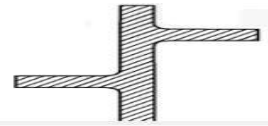
3. Avoid concentration of metal at the junction
    - Even after the metal on the surface solidifies, the central portion still remains in the molten stage, with the result that a shrinkage cavity or blowhole may appear at the center.



There are two ways to avoid the concentration of metal.
One is to provide a cored opening in webs and ribs,



One can stagger the ribs and webs.



4. Avoid very thin sections
5. Shot blast the parts wherever possible
    - The shot blasting process improves the endurance limit of the component, particularly in case of thin sections.
6. In designing a casting, the various allowances must be provided in making a pattern. **(4 marks)**

Answer 5

Mechanical Properties of Metals

The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load. Mechanical properties are given as:

i. Strength. It is the ability of a material to resist the externally applied forces without breaking or yielding. The internal resistance offered by a part to an externally applied force is called stress.

ii. Stiffness or rigidity: It is the ability of the material to resist deformation under the action of an external load. The modulus of elasticity is the measure of stiffness.

iii. Elasticity. It is the property of a material to regain its original shape after deformation when the external forces are removed. This property is desirable for materials used in tools and machines. It may be noted that steel is more elastic than rubber.

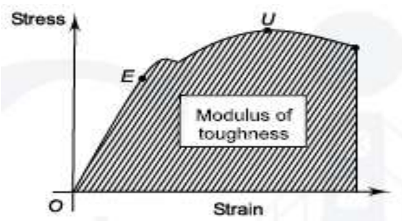
iv. Plasticity. It is property of a material which retains the deformation produced under load permanently. This property of the material is necessary for forgings, in stamping images on coins and in ornamental work.

v. Ductility. It is the property of a material enabling it to be drawn into wire with the application of a tensile force. A ductile material must be both strong and plastic. The ductility is usually measured by the terms, percentage elongation and percentage reduction in area. The ductile material commonly used in engineering practice (in order of diminishing ductility) are mild steel, copper, aluminum, nickel, zinc, tin and lead.

vi. Brittleness. It is the property of a material which shows negligible plastic deformation before fracture takes place. Brittleness is the opposite to ductility. In ductile materials, failure takes place by yielding. Brittle components fail by sudden fracture. A tensile strain of 5% at fracture in a tension test is considered as the dividing line between ductile and brittle materials. Cast iron is a brittle material.

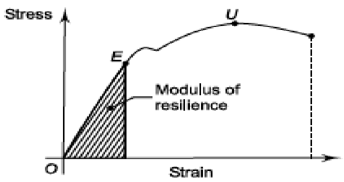
vii. Malleability. It is a special case of ductility which permits materials to be rolled or hammered into thin sheets. A malleable material should be plastic but it is not essential to be so strong. The malleable materials commonly used in engineering practice (in order of diminishing malleability) are lead, soft steel, wrought iron, copper and aluminum.

viii. Toughness. It is the ability of the material to absorb energy before fracture takes place. This property is essential for machine components which are required to withstand impact loads. Tough materials have the ability to bend, twist or stretch before failure takes place. All structural steels are tough materials. Toughness is measured by a quantity called modulus of toughness. Modulus of toughness is the total area under stress–strain curve in a tension test, which also represents the work done to fracture the specimen. In practice, toughness is measured by the Izod and Charpy impact testing machines.



ix. Machinability. It is the property of a material which refers to a relative case with which a material can be cut. The machinability of a material can be measured in a number of ways such as comparing the tool life for cutting different materials or thrust required to remove the material at some given rate or the energy required to remove a unit volume of the material. It may be noted that brass can be easily machined than steel.

x. Resilience. It is the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded. Resilience is measured by a quantity, called modulus of resilience, which is the strain energy per unit volume. It is represented by the area under the stress–strain curve from the origin to the elastic limit point. This property is essential for spring materials.



xi. Creep. When a part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called creep. This property is considered in designing internal combustion engines, boilers and turbines.

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xii. Fatigue. When a material is subjected to fluctuating stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as fatigue. The failure is caused by means of a progressive crack formation which are usually fine and of microscopic size. This property is considered in designing shafts, connecting rods, springs, gears, etc.

xiii. Hardness. It is defined as the resistance of the material to penetration or permanent deformation. It usually indicates resistance to abrasion, scratching, cutting or shaping.

(4 marks)

Answer 6

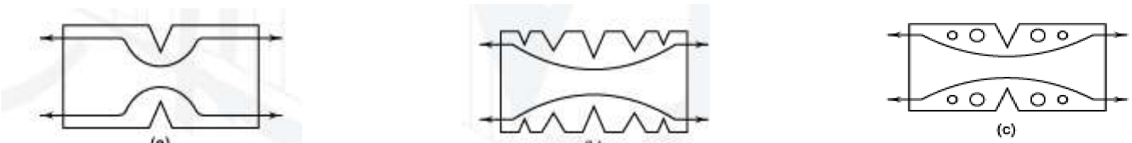
The causes of stress concentration are as follows**:** **(2 marks)**

- Variation in Properties of Materials
- Load Application
- Abrupt Changes in Section
- Discontinuities in the Component
- Machining Scratches

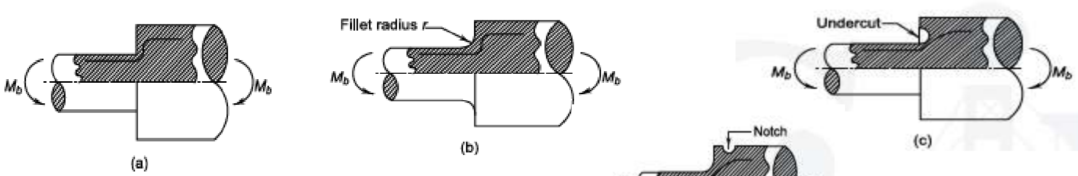
Reduction of Stress Concentration: **(2 marks)**

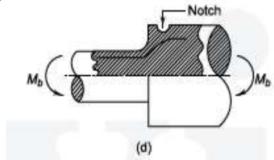
Reduction of stress concentration is achieved by the following methods:

- Additional Notches and Holes in Tension Member



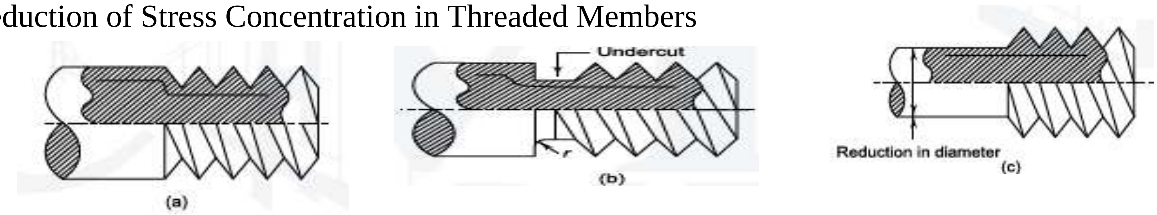
- Fillet Radius, Undercutting and Notch for Member in Bending





- Drilling Additional Holes for Shaft

- Reduction of Stress Concentration in Threaded Members



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Part C

Answer 7

Given:

$P = 50\ kN;\ \sigma_t = 70\ MPa;\ \sigma_c = 140\ MPa;\ \tau = 50\ MPa$

(i) Diameter of each rod:

$$d = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4\times50\times1000}{\pi\times70}} = 30.16\ mm \simeq 31\ mm$$

(01 mark)

(ii) Thickness of cotter:

$t = 0.31\times31 = 9.61\ mm \simeq 10\ mm$ ($\frac{1}{4}$ mark)

(iii) diameter of spigot, (d_2)

$$\frac{50\times10^3}{70} = \frac{\pi}{4}d_2^2 - d_2\times10 \Rightarrow d_2^2 - 12.732d_2 - 909.457 = 0$$

$d_2 = 37.188\ mm \simeq 38\ mm$

(01 mark)

(iv) outer diameter of socket: (d_1)

$$\frac{50\times10^3}{70} = \frac{\pi}{4}(d_1^2 - 38^2) - (d_1 - 38)\times10$$

$d_1^2 - 12.732d_1 - 1869.626 = 0 \Rightarrow d_1 = 51.07\ mm \simeq 51\ mm$

01 mark

(v) diameter of spigot collar and socket collar:

$d_3 = 1.5d = 1.5\times31 = 46.5\ mm \simeq 47\ mm$

$d_4 = 2.4\times d = 2.4\times31 = 74.4\ mm \simeq 75\ mm$ ($\frac{1}{2}$ mark)

(vi) Dimensions a and c

$a = c = 0.75d = 0.75\times31 = 23.25\ mm$

$\simeq 24\ mm$ ($\frac{1}{4}$ mark)

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(VII) Width of Cotter:

$$b = \frac{P}{2\tau t} = \frac{50\times 1000}{2\times 50\times 10} = 50\ mm$$

$$b = \sqrt{\frac{3P}{t\,\sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$$

$$b = \sqrt{\frac{3\times 50\times 10^3}{10\times 70}\left[\frac{38}{4} + \frac{75-38}{6}\right]} = 57.94\ mm$$

$$\simeq 58\ mm$$

(01 Mark)

Hence $b = 58\ mm$

(VIII) Check for stresses: Spigot end

$$\sigma_c = \frac{50\times 10^3}{10\times 38} = 131.58\ \frac{N}{mm^2} < 140\ \frac{N}{mm^2}$$

$$\tau = \frac{50\times 10^3}{2\times 24\times 38} = 27.41\ \frac{N}{mm^2} < 50\ \frac{N}{mm^2}$$

Socket end:

$$\sigma_c = \frac{50\times 10^3}{(75-38)\times 10} = 135.14\ \frac{N}{mm^2} < 140\ \frac{N}{mm^2}$$

$$\tau = \frac{50\times 10^3}{2\times(75-38)\times 24} = 28.15\ \frac{N}{mm^2} < 50\ \frac{N}{mm^2}$$

(01 Mark)

stresses are within limits.

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Answer 8

Given: $P = 60\,kN$; $\sigma_t = 80\,MPa$; $\sigma_c = 100\,MPa$; $\tau = 55\,MPa$

(i) Diameter of each rod, D:

$$D = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4 \times 60 \times 1000}{\pi \times 80}} = 30.90\,mm \simeq 31\,mm$$

(01 Mark)

(ii) Enlarged diameter of rod, D_1:

$$D_1 = 1.1D = 1.1 \times 31 = 34.1\,mm \simeq 35\,mm$$

($\frac{1}{2}$ Mark)

(iii) Dimensions (a) and (b):

$$a = 0.75 \times 31 = 23.25\,mm \simeq 24\,mm$$

$$b = 1.25 \times 31 = 38.75\,mm \simeq 39\,mm$$

(01 Mark)

(iv) Diameter of pin, d

$$d = \sqrt{\frac{2P}{\pi\tau}} = \sqrt{\frac{2 \times 60 \times 1000}{\pi \times 55}} = 26.35\,mm \simeq 27\,mm$$

$$d = \left[\frac{32}{\pi\sigma_b} \times \frac{P}{2}\left(\frac{b}{4} + \frac{a}{3}\right)\right]^{1/3} = 40.77\,mm \simeq 41\,mm$$

Hence $d = 41\,mm$ (01 Mark)

(v) Dimension d_o and d_1:

$$d_o = 2 \times 41 = 82\,mm; \quad d_1 = 1.5 \times 41 = 61.5\,mm \simeq 62\,mm$$

($\frac{1}{2}$ mark)

(vi) Check for stresses:

Eye: $\sigma_t = \frac{60 \times 1000}{39 \times (82 - 41)} = 37.52\,\frac{N}{mm^2} < 80\,MPa$; $\sigma_c = \frac{60 \times 1000}{39 \times 41} < 100\,\frac{N}{mm^2}$

$\tau = \frac{60 \times 1000}{39 \times 41} = 37.52\,\frac{N}{mm^2} < 55\,\frac{N}{mm^2}$

Fork: $\sigma_t = 30.48 < 80\,\frac{N}{mm^2}$; $\sigma_c = 30.48\,\frac{N}{mm^2} < 100\,\frac{N}{mm^2}$; $\tau = 30.48\,N/mm^2 < 55\,MPa$

Stresses are within limits. (2 mark)

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Swami Keshvanand Institute of Technology, Management & Gramothan, Jaipur

II Mid Term Examination, Jan.-2023

Semester:	**V**	**Branch:**	**ME**
Subject:	**DME-I**	**Subject Code:**	**5ME4-04**
Time:	**1.5 Hours**	**Maximum Marks:**	**20**
Session (I/II/III): II			

PART A (short-answer type questions)

(All questions are compulsory) **(3*2=6)**

Q.1 What do mean by bolt of uniform strength?

Q.2 What is self-locking screw?

Q.3 How will you designate ISO metric coarse threads?

PART B (Analytical/Problem solving questions)

(Attempt any 2 Questions) **(2*4=8)**

Q.4 Find the diameter of a solid steel shaft to transmit 20 kW at 200 rpm. The ultimate shear stress for the steel may be taken as 360 MPa and a factor of safety as 8. If a hollow shaft is to be used in place of the solid shaft, find the inside and outside diameters when the ratio of inside to outside diameter is 0.5.

Q.5 It is required to design a square key for fixing a pulley on the shaft, which is 50 mm in diameter. The pulley transmits 10 kW power at 200 rpm to the shaft. The key is made of steel 45C8 ($S_{yt} = S_{yc} = 380$ N/mm^2) and the factor of safety is 3. Determine the dimensions of the key. Assume ($S_{sy} = 0.577 S_{yt}$).

Q.6 A muff coupling to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 ($S_{yt} = S_{yc} = 400$ N/mm^2). The sleeve is made of grey cast iron FG 200 ($S_{ut} = 200$ N/mm^2). The factor of safety for the shafts and key is 4. For the sleeve, the factor of safety is 6 based on ultimate strength. Find out the (i) diameter of each shaft (ii) Dimensions of sleeve.

PART C (Descriptive/Analytical/Problem solving/Design questions)

(Attempt any 1 Question) **(1*6=6)**

Q.7 A lever-loaded safety valve is mounted on the boiler to blow off at a pressure of 1.5 MPa gauge. The effective diameter of the opening of the valve is 50 mm. The distance between the fulcrum and the dead weights on the lever is 1000 mm. The distance between the fulcrum and the pin connecting the valve spindle to the lever is 100 mm. The lever and the pin are made of plain carbon steel 30C8 (S_{yt} = 400

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N/mm^2) and the factor of safety is 5. The permissible bearing pressure at the pins in the lever is 25 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. The length to diameter ratio of the fulcrum pin is 1. Design a suitable lever for the safety valve.

Q.8 A semi elliptic laminated leaf spring has an eye-to-eye span of 1.2 m and supports a central load of 20 kN, for the purpose three full length and six graduated leaves including master leaf are used. Width of the central brand is 0.2 m. The width to thickness ratio for each leaf is 6. The allowable stress is 200 N/mm^2. Take $E = 2 \times 10^5$ N/mm^2. Calculate the section of leaves and deflection at full load if: (i) Leaves are not stressed initially (ii) Leaves are stressed initially for equalized stresses at maximum load.

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II Mid Term Examination, Jan.-2023

Semester:	**V**	**Branch:**	**ME**
Subject:	**DME-I**	**Subject Code:**	**5ME4-04**

A. Distribution of Course Outcome and Bloom's Taxonomy in Question Paper

Q. No	Questions	Marks	CO	BL
1	What do mean by bolt of uniform strength?	2	5	1
2	What is self-locking screw?	2	5	1
3	How will you designate ISO metric coarse threads?	2	5	2
4	Find the diameter of a solid steel shaft to transmit 20 kW at 200 rpm. The ultimate shear stress for the steel may be taken as 360 MPa and a factor of safety as 8. If a hollow shaft is to be used in place of the solid shaft, find the inside and outside diameters, when the ratio of inside to outside diameter is 0.5.	4	4	3
5	It is required to design a square key for fixing a pulley on the shaft, which is 50 mm in diameter. The pulley transmits 10 kW power at 200 rpm to the shaft. The key is made of steel 45C8 ($S_{yt} = S_{yc} =$ 380 N/mm^2) and the factor of safety is 3. Determine the dimensions of the key. Assume ($S_{sy} = 0.577 S_{yt}$).	4	4	3
6	A muff coupling to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 ($S_{yt} = S_{yc}$ = 400 N/mm^2). The sleeve is made of grey cast iron FG 200 (S_{ut} = 200 N/mm^2). The factor of safety for the shafts and key is 4. For the sleeve, the factor of safety is 6 based on ultimate strength. Find out the (i) diameter of each shaft (ii) Dimensions of sleeve.	4	4	3
7	A lever-loaded safety valve is mounted on the boiler to blow off at a pressure of 1.5 MPa gauge. The effective diameter of the opening of the valve is 50 mm. The distance between the fulcrum and the dead weights on the lever is 1000 mm. The distance between the fulcrum and the pin connecting the valve spindle to the lever is 100 mm. The lever and the pin are made of plain carbon steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure at the pins in the lever is 25 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. Design a suitable lever for the safety valve.	6	3	3
8	A semi elliptic laminated leaf spring has an eye-to-eye span of 1.2 m and supports a central load of 20 kN, for the purpose three full length and six graduated leaves including master leaf are used. Width of the central brand is 0.2 m. The width to thickness ratio for each leaf is 6. The allowable stress is 200 N/mm^2. Take $E = 2 \times 10^5$ N/mm^2. Calculate the section of leaves and deflection at full load if: (i) Leaves are not stressed initially (ii) Leaves are stressed initially for equalized stresses at maximum load.	6	3	3

BL – Bloom's Taxonomy Level

(1- Remembering, 2- Understanding, 3 – Applying, 4 – Analyzing, 5 – Evaluating, 6 - Creating)

CO – Course Outcome

B. Questions and Course Outcomes (COs) Mapping in terms of correlation

COs	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8
CO1	-	-	-	-	-	-	-	-
CO2	-	-	-	-	-	-	-	-
CO3	-	-	-	-	-	-	3	3
CO4	-	-	-	3	3	3	-	-
CO5	2	2	2	-	-	-	-	-

1-Low Correlation; 2- Moderate Correlation; 3- Substantial Correlation

C. Mapping of Bloom's Level and Course Outcomes with Question Paper

Bloom's Level Mapping		**CO Mapping**	
Bloom's Level	Percentage	CO	Percentage
BL1	13.33	CO1	-
BL2	6.67	CO2	-
BL3	80.00	CO3	40
BL4	00	CO4	40
BL5	00	CO5	20
BL6	00		

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Solution of II Mid Term Examination, Jan.-2023

PART-A

Answer 1.

The energy absorbed during elastic deformation is proportional to the square of the stress induced in the material and the volume of which is subjected to same stress level at different cross-sections in the bolt. It is called the bolt of uniform strength. In a bolt of uniform strength, the entire bolt is stressed to the same limiting value, thus resulting in maximum energy absorption. **(2 marks)**

Answer 2.

When friction angle ($\emptyset$) $\geq$ *helix anglr* (α) a positive torque is required to lower the load. Under this condition, the load will not turn the screw and will not descend on its own unless an effort P is applied. In this case, the screw is said to be 'self-locking'. **(2 marks)**

Answer 3

A metric ISO screw thread is designated by the letter M followed by the value of the nominal diameter D (the maximum thread diameter) as M12. **(2 marks)**

Part B

Answer 4:

Given: $P = 20\,kW$; $N = 200\,rpm$; $S_{us} = 360\,MPa$; $FS = 8$

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

Permissible stress:

$$\tau_{per.} = \frac{360}{8} = 45\,MPa$$

Diameter of solid shaft:

$$\tau = \frac{16T}{\pi d^3} \leq \tau_{per.}$$

$$T = \frac{60P}{2\pi N} = \frac{60 \times 20 \times 1000}{2\pi \times 200} = 954.93\ N\text{-}m$$

$$\frac{16 \times 954.93}{\pi d^3} = 45 \times 10^6 \Rightarrow d = 0.04763\,m = 47.63\,mm$$

(02 marks)

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Inside and outside diameters:

$$\tau = \frac{T}{J}\cdot r = \frac{32T}{\pi(d_o^4 - d_i^4)}\cdot\frac{d_o}{2} = \frac{16T}{\pi d_o^3\left(1-\frac{d_i^4}{d_o^4}\right)} \le \tau_{per.}$$

$$\frac{16\times 954.93\times 1000}{\pi d_o^3(1-0.5^4)} = 45 \Rightarrow d_o = 48.67\ mm$$

$$d_i = 24.33\ mm$$ (02 marks)

Answer 5:

Given: $P = 10\ kW$; $N = 200\ rpm$; $S_{yt} = S_{yc} = 380\ \frac{N}{mm^2}$

$FS = 3$; $S_{sy} = 0.577 S_{yt}$; $d = 50\ mm$

Permissible stresses:

$$\sigma_c = \frac{S_{yc}}{FS} = \frac{380}{3} = 126.67\ \frac{N}{mm^2}$$

$$\tau = \frac{0.577 S_{yt}}{FS} = 73.09\ \frac{N}{mm^2}$$ (01 mark)

Dimensions of key:

$$b = h = \frac{d}{4} = \frac{50}{4} = 12.5\ mm$$ ($\frac{1}{2}$ mark)

$$T = \frac{60P}{2\pi N} = \frac{60\times 10\times 1000}{2\pi\times 200} = 477.465\ N\text{-}m$$ ($\frac{1}{2}$ mark)

Length based on shear stress:

$$l = \frac{2T}{\tau d b} = \frac{2\times 477.465\times 1000}{73.09\times 50\times 12.5} = 20.90\ mm$$

Length based on crushing stress:

$$l = \frac{4T}{\sigma_c d h} = \frac{4\times 477.465\times 1000}{126.67\times 50\times 12.5} = 24.12\ mm$$

Thus; $l = \max(20.90, 24.12) = 24.12\ mm$

Size of key: $12.5\times 12.5\times 24.12\ mm$ (02 marks)

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Answer 6:

Given: $P = 25\ kW$; $N = 360\ rpm$

for shaft and key, $S_{yt} = S_{yc} = 400\ \frac{N}{mm^2}$; $fs = 4$

for sleeve, $S_{ut} = 200\ \frac{N}{mm^2}$; $fs = 6$

Permissible stress.

for shaft and key, $\sigma_t = \frac{400}{4} = 100\ \frac{N}{mm^2}$

$\sigma_c = \frac{400}{4} = 100\ \frac{N}{mm^2}$; $\tau_{per.} = \frac{0.5\times400}{4} = 50\ \frac{N}{mm^2}$

for sleeve, $\tau_{per.} = \frac{0.5\times200}{6} = 16.67\ \frac{N}{mm^2}$ (01 mark)

Diameter of each shaft:

$$T = \frac{60P}{2\pi N} = \frac{60\times25\times1000\times10^3}{2\pi\times360} = 663145.60\ N\text{-}mm$$

$$\tau = \frac{16T}{\pi d^3} \le \tau_{per.} \Rightarrow \frac{16\times663145.60}{\pi d^3} = 50 \Rightarrow d = 40.73\ mm \simeq 41\ mm$$

(02 marks)

Dimensions of sleeve: $D = (2d+13) = (2\times41+13) = 95\ mm$

$L = 3.5\times41 = 143.5\ mm \simeq 144\ mm$

$$\tau = \frac{16\times663145.60\times95}{\pi(95^4-41^4)} = 4.08\ \frac{N}{mm^2} < 16.67\ \frac{N}{mm^2}$$ (01 mark)

Dimensions of key: $l = \frac{L}{2} = \frac{144}{2} = 72\ mm\ \times$

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Part C

Answer 7:

Given: for Valve; $d = 50mm$; $p = 1.5 MPa$

$S_{yt} = 400 \frac{N}{mm^2}$; $FS = 5$

for lever; $l_1 = 1000\,mm$; $l_2 = 100\,mm$; $\frac{d}{b} = 3$

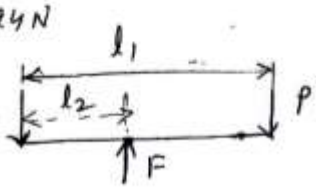
for pin; $P_b = 25 \frac{N}{mm^2}$; $\frac{l_c}{d_1} = 1$

Permissible stresses

$\sigma_t = \frac{400}{5} = 80 \frac{N}{mm^2}$; $\tau = \frac{0.5 \times 400}{5} = 40 \frac{N}{mm^2}$ (01 mark)

Forces on lever:

$F = \frac{\pi}{4} d^2 \times p = \frac{\pi}{4} \times 50^2 \times 1.5 = 2945.24\,N$



$F = R + P$ ---- (1)

$F l_2 = P l_1 \Rightarrow P = 2945.24 \times \frac{100}{1000}$

$P = 294.52\,N$

From equation (1):

$R = 2650.72\,N$ (02 marks)

Dimensions of pin:

$l_c = d_1$

$P_b = \frac{2945.24}{d_1^2} \Rightarrow d_1 = \sqrt{\frac{2945.24}{25}} = 10.85\,mm \simeq 11\,mm$

$l_c = 11\,mm$ (01 mark)

Dimension of lever:

$M_b = 294.52 \times (1000 - 100) = 265068\,N\text{-}mm$

$\sigma_b = \frac{M_b y}{I} \leq \sigma_t \Rightarrow \frac{265068 \times 1.5b}{\frac{1}{12} \times [b\,(3b)^3]} = 80$

$b = 13.02$ mm

$d = 3 \times 13.02 = 39.06$ mm ≈ 40 mm (02 marks)

Answer 8:

Given: $2L_1 = 1.2\,m = 1200$ mm

$2P = 20$ kN

$n_f = 3; \quad n_g = 6$

$l = 0.2\,m = 200$ mm ; $\frac{b}{t} = 6$

$\sigma_{all} = 200 \frac{N}{mm^2}$; $E = 2 \times 10^5 \frac{N}{mm^2}$

Effective length, $2L = 2L_1 - l = 1200 - 200$

$L = 500$ mm

(i) Leaves are not stresses initially.

stress in full length leaves is more than that of graduated length leaves.

$$\sigma_f = \frac{18PL}{(3n_f + 2n_g)\,bt^2} \leq \sigma_{all}$$

$$\frac{18 \times (\frac{20}{2}) \times 1000 \times 500}{(3 \times 3 + 2 \times 6) \times 6t^3} = 200$$

$t = 15.29$ mm $= 16$ mm (02 marks)

$b = 6 \times 16 = 96$ mm

Deflection: $$\delta = \frac{12PL^3}{Ebt^3(3n_f + 2n_g)}$$

$$\delta = \frac{12 \times 10 \times 1000 \times 500^3}{2 \times 10^5 \times 96 \times 16^3 (3 \times 3 + 2 \times 6)} = 9.08 \text{ mm}$$ (01 mark)

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(ii) Leaves are stresses:

$$\sigma_b = \frac{6PL}{nbt^2} \leq S_{all}$$

$$\frac{6 \times 10 \times 1000 \times 500}{9 \times 6t^3} = 200$$

$t = 14.06\ mm \simeq 15\ mm$

$b = 15 \times 6 = 90\ mm$ (02 marks)

$$\delta = \frac{12 \times 10 \times 1000 \times 500^3}{2 \times 10^5 \times 90 \times 15^3 (3 \times 3 + 2 \times 6)}$$

$\delta = 11.76\ mm$ (01 mark)

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Extra/III Mid Term Examination, Feb.-2023

Semester:	**V**	**Branch:**	**ME**
Subject:	**DME-I**	**Subject Code:**	**5ME4-04**
Time:	**1.5 Hours**	**Maximum Marks:**	**30**
Session (I/II/III): III			

PART A (short-answer type questions)
(All questions are compulsory) **(4*3=12)**

Q.1What is the ductility and Brittleness.

Q.2 Explain factor of safety.

Q.3 How will you designate ISO metric fine and coarse threads?

Q.4 What is ‘overhauling’ of power screw?

PART B (Analytical/Problem solving questions)
(Attempt any 3 Questions) **(3*4=12)**

Q.5 What are the three basic modes of failure of mechanical components?

Q.6 Explain BIS or IS system of plain carbon steel.

Q.7 Explain the hole-basis and shaft-basis system of fits with neat sketches.

Q.8 Explain the stress concentration causes and mitigation with suitable figures?

PART C (Descriptive/Analytical/Problem solving/Design questions)
(Attempt any 1 Question) **(1*6=6)**

Q.9 A right-angled bell-crank lever is to be designed to raise a load of 5 kN at the short arm end. The lengths of short and long arms are 100 and 450 mm, respectively. The lever and the pins are made of steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. The length to diameter ratio of the fulcrum pin is 1.25:1. Calculate (i) The diameter and the length of the fulcrum pin (ii) The shear stress in the pin (iii) The dimensions of the cross-section of the lever. Assume that the arm of the bending moment on the lever extends up to the axis of the fulcrumthe lever at the fulcrum.

Q.10 A steel spindle transmits 4 kW at 800 RPM. The angular deflection should not exceed 0.25° per meter of the spindle. If the modulus of rigidity for the material of the spindle is 84 GPa, find the diameter of the spindle and the shear stress induced in the spindle.

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Extra/III Mid Term Examination, Feb.-2023

Semester:	**V**	**Branch:**	**ME**
Subject:	**DME-I**	**Subject Code:**	**5ME4-04**
Time:	**1.5 Hours**	**Maximum Marks:**	**30**

A. Distribution of Course Outcome and Bloom's Taxonomy in Question Paper

Q. No	Questions	Marks	CO	BL
1	What is the ductility and Brittleness.	3	1	1
2	Explain the factor of safety.	3	2	2
3	How will you designate ISO metric fine and coarse threads?	3	5	2
4	What is 'overhauling' of power screw?	3	5	1
5	What are the three basic modes of failure of mechanical components?	4	2	1
6	Explain BIS or IS system of plain carbon steel.	4	1	2
7	Explain the hole-basis and shaft-basis system of fits with neat sketches.	4	1	2
8	Explain the stress concentration causes and mitigation with suitable figures	4	2	2
9	A right-angled bell-crank lever is to be designed to raise a load of 5 kN at the short arm end. The lengths of short and long arms are 100 and 450 mm, respectively. The lever and the pins are made of steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. The length to diameter ratio of the fulcrum pin is 1.25:1. Calculate (i) The diameter and the length of the fulcrum pin (ii) The shear stress in the pin (iii) The dimensions of the cross-section of the lever. Assume that the arm of the bending moment on the lever extends up to the axis of the fulcrum the lever at the fulcrum.	6	3	3
10	A steel spindle transmits 4 kW at 800 RPM. The angular deflection should not exceed 0.25° per meter of the spindle. If the modulus of rigidity for the material of the spindle is 84 GPa, find the diameter of the spindle and the shear stress induced in the spindle.	6	4	3

BL – Bloom's Taxonomy Level
(1- Remembering, 2- Understanding, 3 – Applying, 4 – Analyzing, 5 – Evaluating, 6 - Creating)
CO – Course Outcome

B. Questions and Course Outcomes (COs) Mapping in terms of correlation

COs	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8	Q9	Q10
CO1	2	-	-	-	-	3	3	-	-	-
CO2	-	2	-	-	3	-	-	3	-	-
CO3	-	-	-	-	-	-	-	-	3	-
CO4	-	-	-	-	-	-	-	-	-	3
CO5	-	-	2	2	-	-	-	-	-	-

1-Low Correlation; 2- Moderate Correlation; 3- Substantial Correlation

D. Mapping of Bloom's Level and Course Outcomes with Question Paper

Bloom's Level Mapping		**CO Mapping**	
Bloom's Level	Percentage	CO	Percentage
BL1	25	CO1	27.5
BL2	45	CO2	27.5
BL3	30	CO3	15
BL4	00	CO4	15
BL5	00	CO5	15
BL6	00		

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Solution of Extra/III Mid Term Examination, Feb.-2023

PART-A

Answer 1.

Ductility is the ability of the solid material to deform plastically under loading.
Brittleness is when the material has the tendency to not deform plastically under tensile loading, but instead to fracture / break. **(3)**

.

Answer 2.

The factor of safety is defined as *the ratio of ultimate stress to the working stress.* the factor of safety is a load carrying capacity of a system up to which load is sustained. Factor of safety shows how much system is stronger than the required. ***(3)***

Answer 3

A metric ISO screw thread is designated by the letter M followed by the value of the nominal diameter D (the maximum thread diameter) and the pitch P, both expressed in millimeters and separated by the multiplication sign, × (e.g., M8×1.25).

A metric ISO screw thread is designated by the letter M followed by the value of the nominal diameter D (the maximum thread diameter) as M12 **(3)**

Answer 4

It indicates a condition that no force is required to lower the load. The load itself will begin to turn the screw and descend unless a restraining torque is applied. The condition is called overhauling of the screw. **(3)**

Part B

Answer 5

A mechanical component may fail, that is, may be unable to perform its function satisfactorily, as a result of any one of the following three modes of failure1:

(i) failure by elastic deflection

(ii) failure by general yielding

(iii) failure by fracture **(4)**

Answer 6

BIS or IS system of plain carbon and alloy steel:

Steels Designated based on Mechanical Properties: According to Indian standard IS: 1570 (Part–I)-1978 (Reaffirmed 1993), these steels are designated by a symbol 'Fe' or 'Fe E' depending on whether the steel has been specified based on minimum tensile strength or yield strength, followed by the figure indicating the minimum tensile strength or yield stress in N/mm^2. For example, 'Fe 290' means a steel having minimum tensile strength of 290 N/mm^2 and 'Fe E 220' means a steel having yield strength of 220 N/mm^2. **(2)**

Steels Designated based on Chemical Composition: According to Indian standard, IS: 1570 (Part II/Sec I)-1979 (Reaffirmed 1991), The designation of plain carbon steel consists of the following three quantities:

(i) a figure indicating 100 times the average percentage of carbon; (ii) a letter C; and (iii) a figure indicating 10 times the average percentage of manganese.

As an example, 55C4 indicates a plain carbon steel with 0.55% carbon and 0.4% manganese. Steel with 0.35–0.45% carbon and 0.7–0.9% manganese is designated as 40C8.

The designation of unalloyed free cutting steels: it consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon; (ii) a letter C; (iii) a figure indicating 10 times the average percentage of manganese; (iv) a symbol 'S', 'Se', 'Te' or 'Pb' depending

upon the element that is present, and which makes the steel free cutting; and (v) a figure indicating 100 times the average percentage of the above element that makes the steel free cutting.

As an example, 25C12S14 indicates a free cutting steel with 0.25% carbon, 1.2% manganese.

and 0.14% Sulphur. Similarly, a free cutting steel with an average of 0.20% carbon, 1.2% manganese

and 0.15% lead is designated as 20C12Pb15. **(2)**

Answer 7: Hole-basis and shaft-basis system of fits:

Hole Basis and Shaft Basis Systems

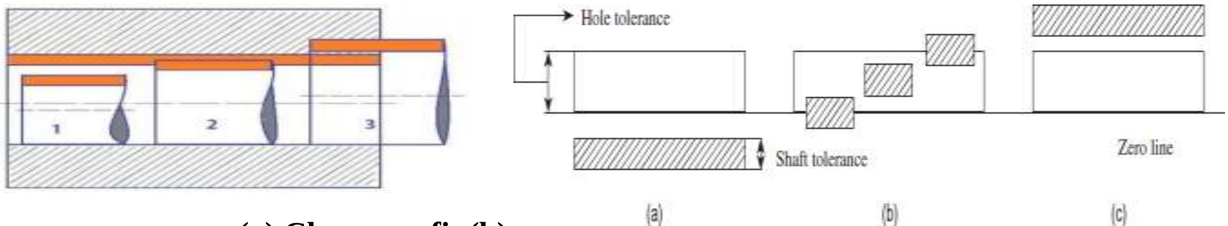
- To obtain the desired class of fits, either the size of the hole or the size of the shaft must vary.

Two types of systems are used to represent three basic types of fits, clearance, interference, and transition fits.

- Hole basis system
- Shaft basis system.

Hole Basis systems

- The size of the hole is kept constant and the shaft size is varied to give various types of fits.
- Lower deviation of the hole is zero, i.e. the lower limit of the hole is same as the basic size.
- Two limits of the shaft and the higher dimension of the hole are varied to obtain the desired type of fit.

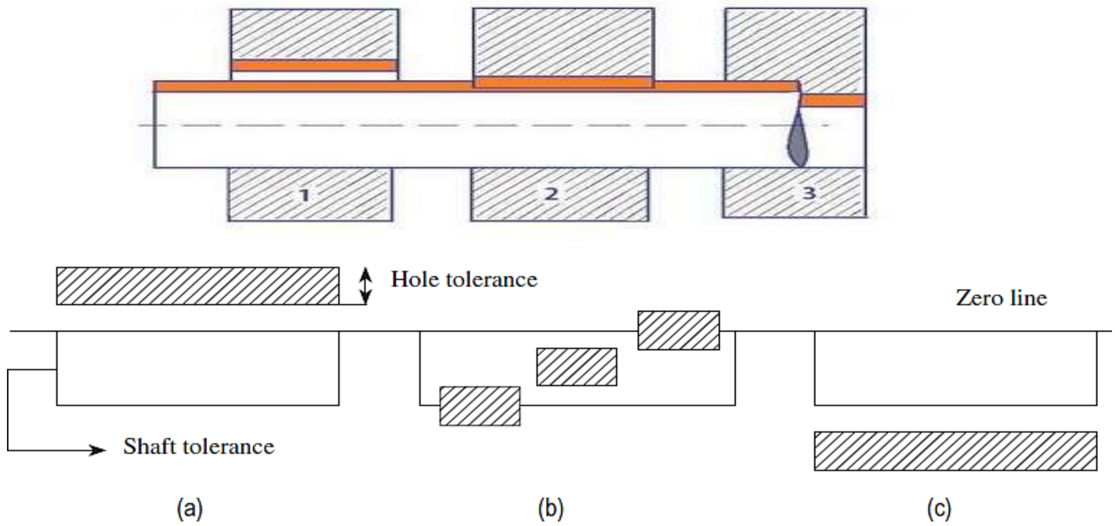


(a) Clearance fit (b) Transition fit (c) Interference fit

This system is widely adopted in industries, easier to manufacture shafts of varying sizes to the required tolerances. Standard-size plug gauges are used to check hole sizes accurately **(2)**

Shaft Basis systems

- The size of the shaft is kept constant, and the hole size is varied to obtain various types of fits.
- Fundamental deviation or the upper deviation of the shaft is zero.
- System is not preferred in industries, as it requires more number of standard- size tools, like reamers, broaches, and gauges, increases manufacturing and inspection costs.



(a) Clearance fit (b) Transition fit (c) Interference fit **(2)**

Answer 8

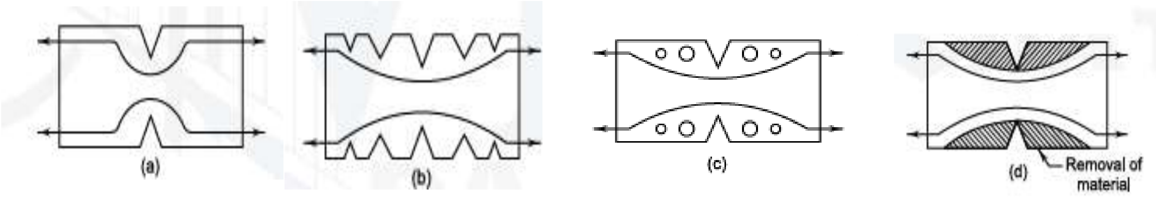
The causes of stress concentration are as follows: **(2)**

- Variation in Properties of Materials
- Load Application
- Abrupt Changes in Section
- Discontinuities in the Component
- Machining Scratches

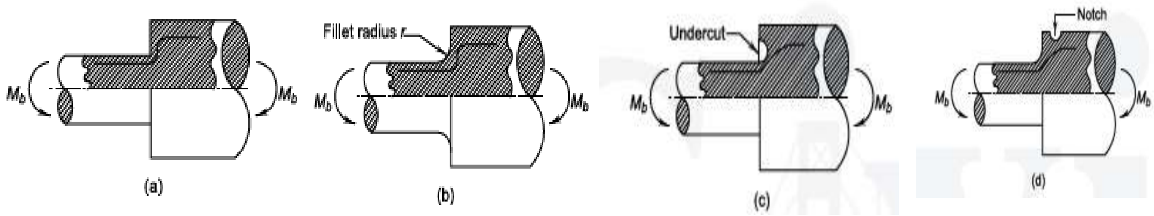
Reduction of Stress Concentration: **(2)**

Reduction of stress concentration is achieved by the following methods:

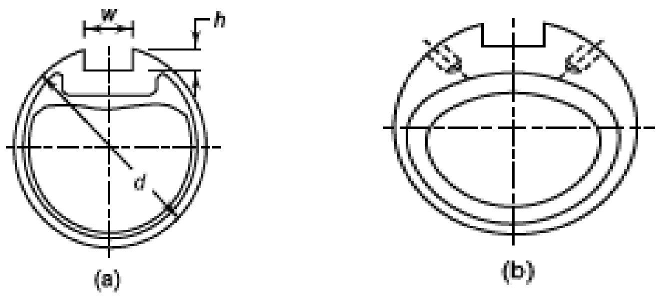
- Additional Notches and Holes in Tension Member



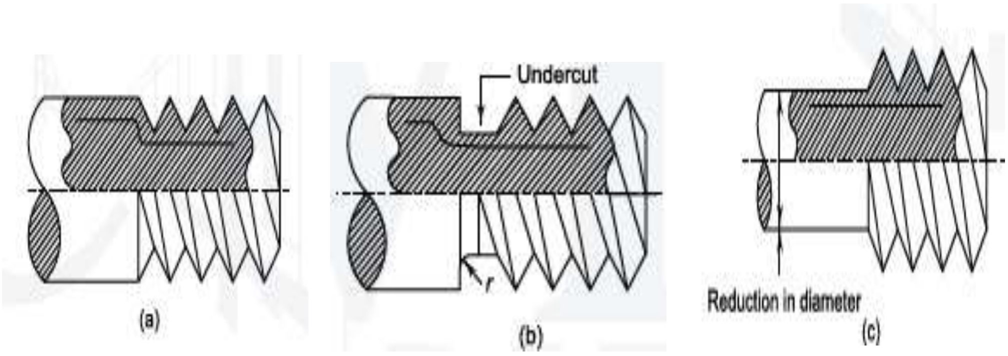
- Fillet Radius, Undercutting and Notch for Member in Bending



- Drilling Additional Holes for Shaft



- Reduction of Stress Concentration in Threaded Members



Part C

Answer 9

Given $S_{yt} = 400N/mm^2 \quad (fs) = 5 \quad F = 5kN.$

For lever, long arm = 450 mm short arm = 100 mm d/b = 3.

For pin $p = 10N/mm^2 \quad l_1/d_1 = 1.25.$

Step I Calculation of permissible stresses for the pin and lever

$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80N/mm^2.$

(1)

$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{5} = 40N/mm^2.$

Step II Calculation of forces acting on the lever

The forces acting on the lever are shown in Fig. 4.53. Taking moment of forces about the axis of the fulcrum,

$(5 \times 10^3)(100) = P \times 450 \quad \therefore P = 1111.11N.$

$R = \sqrt{(5000)^2 + (1111.11)^2} = 5121.97N.$

(1)

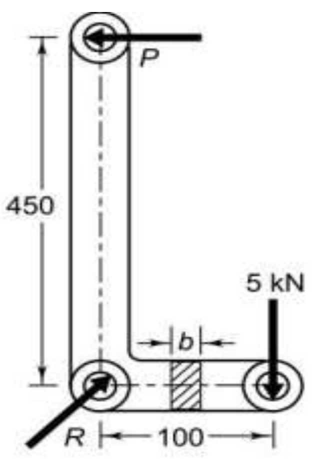


Fig. 4.53

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Step III Diameter and length of fulcrum pin

Considering bearing pressure on the fulcrum pin,

$R = p(\text{ projected area of the pin }) = p\,(d_1 \times l_1)$

or $\quad R = p\,(d_1 \times l_1) \quad$ (a).

where,

$d_1 =$ diameter of the fulcrum pin (mm).

$l_1 =$ length of the fulcrum pin (mm) .

p = permissible bearing pressure (N/mm²)

Substituting values in Eq. (a), **(2)**

$5121.97 = 10\,(d_1 \times 1.25 d_1).$

$\therefore \quad d_1 = 20.24 mm.$

$l_1 = 1.25 d_1 = 1.25(20.24) = 25.30 mm$ (i).

Step IV Shear stress in pin

The pin is subjected to double shear. The shear stress in the pin is given by,

$\tau = \frac{R}{2\left[\frac{\pi}{4} d_1^2\right]} = \frac{5121.97}{2\left[\frac{\pi}{4}(20.24)^2\right]} = 7.96 N/mm^2$ (ii) .

Step V Dimensions of the boss

The dimensions of the boss of the lever at the fulcrum are as follows:

inner diameter = 21 mm

outer diameter = 42 mm

length = 26 mm (iii)

Step VI Dimensions of cross-section of lever

For the lever

$d = 3b \quad M_b = (5000 \times 100) N\text{–}mm.$

Therefore,

$\sigma_b = \frac{M_b y}{I} \quad \text{or} \quad 80 = \frac{(5000 \times 100)(1.5b)}{\left[\frac{1}{12}(b)(3b)^3\right]}.$ **(2)**

$\therefore$ b = 16.09 mm

d = 3b = 3 (16.09) = 48.27 mm (iv).

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Answer 10

Solution. Given : $P = 4$ kW $= 4000$ W ; $N = 800$ r.p.m. ; $\theta = 0.25° = 0.25 \times \frac{\pi}{180} = 0.0044$ rad ; $L = 1$ m $= 1000$ mm ; $G = 84$ GPa $= 84 \times 10^9$ N/m^2 $= 84 \times 10^3$ N/mm^2

Diameter of the spindle

Let d = Diameter of the spindle in mm.

We know that the torque transmitted by the spindle,

$$T = \frac{P \times 60}{2\pi N} = \frac{4000 \times 60}{2\pi \times 800} = 47.74 \text{ N-m} = 47\,740 \text{ N-mm}$$

We also know that $\frac{T}{J} = \frac{G \times \theta}{L}$ or $J = \frac{T \times l}{G \times \theta}$

or $$\frac{\pi}{32} \times d^4 = \frac{47\,740 \times 1000}{84 \times 10^3 \times 0.0044} = 129\,167$$ **(3)**

$\therefore$ $d^4 = 129\,167 \times 32/\pi = 1.3 \times 10^6$ or $d = 33.87$ say 35 mm **Ans.**

Shear stress induced in the spindle

Let τ = Shear stress induced in the spindle.

We know that the torque transmitted by the spindle (T),

$$47\,740 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times \tau\,(35)^3 = 8420\,\tau$$

$\therefore$ $\tau = 47\,740 / 8420 = 5.67$ N/mm^2 $= 5.67$ MPa **Ans.**

(3)

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I Mid Term Examination-2021: SET- B

B.Tech./ Semester – III Yr./V Sem. **Branch: ME**
Subject: Design of Machine Elements-I **Subject Code: 5ME4-04**
Time: 1½ Hours **Maximum Marks: 50**

B. Distribution of Course Outcome and Bloom's Taxonomy in Question Paper

Q.No	Questions	Marks	CO	BL
9.	Explain the BIS system of designation for plain carbon and alloy steel with suitable examples	10	1	2
10.	Explain the design considerations for forging products with neat sketches.	10	1	2
11.	Explain the hole-basis and shaft-basis system of fits with neat sketches.	10	1	2
12.	The stresses induced at a critical point in a machine component made of steel (Syt= 380 N/mm^2) are as follows $\sigma x = 100\ MPa$, $\sigma y = 40\ MPa$, $\tau xy = 80 MPa$, calculate the factor of safety by (i) The maximum normal stress theory, (ii) The maximum shear stress theory (iii) The distortion energy theory	10	2	3
13.	It is required to design a knuckle joint to connect two circular rods subjected to an axial tensile force of 60 kN. The permissible stresses may be taken as 80 MPa in tension, 55 MPa in shear and 100 MPa in crushing.	10	2	3

BL – Bloom's Taxonomy Levels

(1- Remembering, 2- Understanding, 3 – Applying, 4 – Analyzing, 5 – Evaluating, 6 - Creating)

CO – Course Outcomes

C. Questions and Course Outcomes (COs) Mapping in terms of correlation

COs	Q1	Q2	Q3	Q4	Q5
CO1	3	3	3	0	0
CO2	0	0	0	3	3
CO3	0	0	0	0	0
CO4	0	0	0	0	0
CO5	0	0	0	0	0

1-Low Correlation; 2- Moderate Correlation; 3- Substantial Correlation

D. Mapping of Bloom's Level and Course Outcomes with Question Paper

Bloom's Level Mapping		CO Mapping	
Bloom's Level	Percentage	CO	Percentage
BL1	0	CO1	60
BL2	60	CO2	40
BL3	40	CO3	00
BL4	00	CO4	00
BL5	00	CO5	00
BL6	00	CO6	00

Solution of First Mid Term 2021 (Set-B)

Answer 1: BIS or IS system of plain carbon and alloy steel:

Steels Designated on the Basis of Mechanical Properties: According to Indian standard **IS: 1570 (Part–I)-1978 (Reaffirmed 1993), these steels are designated by a symbol 'Fe' or 'Fe E' depending on whether the steel has been specified on the basis of minimum tensile strength or yield strength, followed by the figure indicating the minimum tensile strength or yield stress in N/mm^2. For example, 'Fe 290' means a steel having minimum tensile strength of 290 N/mm^2 and 'Fe E 220' means a steel having yield strength of 220 N/mm^2. **(2)**

Steels Designated on the Basis of Chemical Composition: According to Indian standard, IS: 1570 (Part II/Sec I)-1979 (Reaffirmed 1991), The designation of plain carbon steel consists of the following three quantities:

(i) a figure indicating 100 times the average percentage of carbon; (ii) a letter C; and (iii) a figure indicating 10 times the average percentage of manganese.

As an example, 55C4 indicates a plain carbon steel with 0.55% carbon and 0.4% manganese. A steel with 0.35–0.45% carbon and 0.7–0.9% manganese is designated as 40C8.

The designation of unalloyed free cutting steels: it consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon; (ii) a letter C; (iii) a figure indicating 10 times the average percentage of manganese; (iv) a symbol 'S', 'Se', 'Te' or 'Pb' depending
upon the element that is present and which makes the steel free cutting; and (v) a figure indicating 100 times the average percentage of the above element that makes the steel free cutting.

As an example, 25C12S14 indicates a free cutting steel with 0.25% carbon, 1.2% manganese and 0.14% Sulphur. Similarly, a free cutting steel with an average of 0.20% carbon, 1.2% manganese and 0.15% lead is designated as 20C12Pb15. **(3)**

Indian Standard Designation of Low and Medium Alloy Steels:

The term 'alloy' steel is used for low and medium alloy steels containing total alloying elements not exceeding 10%. The designation of alloy steels consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon; and (ii) chemical symbols for alloying elements each followed by the figure for its average percentage content multiplied by a factor.

The multiplying/Division factor depends upon the upon the alloying element. The values of this factor are as follows:

Element	*Multiplying factor*
Cr, Co, Ni, Mn, Si and W	4
Al, Be, V, Pb, Cu, Nb, Ti, Ta, Zr and Mo	10
P, S and N	100

In alloy steels, 'Mn and silicon (Si)' is included only if the content of manganese is equal to or greater than 1%. The chemical symbols and their figures are arranged in descending order of their percentage content. As an example, 25Cr4Mo2 is an alloy steel having average 0.25% of carbon, 1% chromium

and 0.2% molybdenum. Similarly, 40Ni8Cr8V2 is an alloy steel containing average 0.4% of carbon, 2% nickel, 2% chromium and 0.2% vanadium.
Consider an alloy steel with the following composition:
carbon = 0.12–0.18%
silicon = 0.15–0.35%
manganese = 0.40–0.60%
chromium = 0.50–0.80%
The average percentage of carbon is 0.15%, which is denoted by the number (0.15×100) or 15. The percentage content of silicon and manganese is negligible and, as such, they are deleted from the designation. The significant element is chromium and its average percentage is 0.65. The multiplying factor for chromium is 4 and (0.65×4) is 2.6, which is rounded to 3. Therefore, the complete designation of steel is 15Cr3. As a second example, consider a steel with the following chemical composition:
carbon = 0.12–0.20%
silicon = 0.15–0.35%
manganese = 0.60–1.00%
nickel = 0.60–1.00%
chromium = 0.40–0.80% The average percentage of carbon is 0.16% and multiplying this value by 100, the first figure in the designation of steel is 16. The average percentage of silicon and manganese is very small and, as such, the symbols Si and Mn are deleted. Average percentages of nickel and chromium are 0.8 and 0.6, respectively, and the multiplying factor for both elements is 4. Therefore, nickel: 0.8 × 4 = 3.2 rounded to 3 or Ni3 chromium: 0.6×4 = 2.4 rounded to 2 or Cr2. The complete designation of steel is 16Ni3Cr2. **(3)**

Indian Standard Designation of high Alloy Steels:

The term 'high alloy steels' is used for alloy steels containing more than 10% of alloying elements. The designation of high alloy steels consists of the following quantities:
(i) a letter 'X';
(ii) a figure indicating 100 times the average percentage of carbon;
(iii) chemical symbol for alloying elements each followed by the figure for its average percentage content rounded off to the nearest integer; and
(iv) chemical symbol to indicate a specially added element to attain desired properties, if any. As an example, X15Cr25Ni12 is a high alloy steel with 0.15% carbon, 25% chromium and 12% nickel. As a second example, consider a steel with the following chemical composition:
carbon = 0.15–0.25%
silicon = 0.10–0.50%
manganese = 0.30–0.50%
nickel = 1.5–2.5%
chromium = 16–20% **(2)**
The average content of carbon is 0.20%, which is denoted by a number (0.20×100) or 20. The major alloying elements are chromium (average 18%) and nickel (average 2%). Hence, the designation of steel is X20Cr18Ni2.

Answer (2): Design considerations of forgings

The general principles for the design of forged components are as follows:

i. While designing a forging, the profile is selected in such a way that fibre lines are parallel to tensile forces and perpendicular to shear forces.

ii. Machining that cuts deep into the forging should be avoided, otherwise the fibre lines are broken and the part becomes weak.
iii. The forged component should be provided with an adequate draft.
iv. The parting line should be in one plane as far as possible and it should divide the forging into two equal parts.
v. The forging should be provided with adequate fillet and corner radii.
vi. Thin sections and ribs should be avoided in forged components.

Answer (3): Hole-basis and shaft-basis system of fits

Hole Basis and Shaft Basis Systems

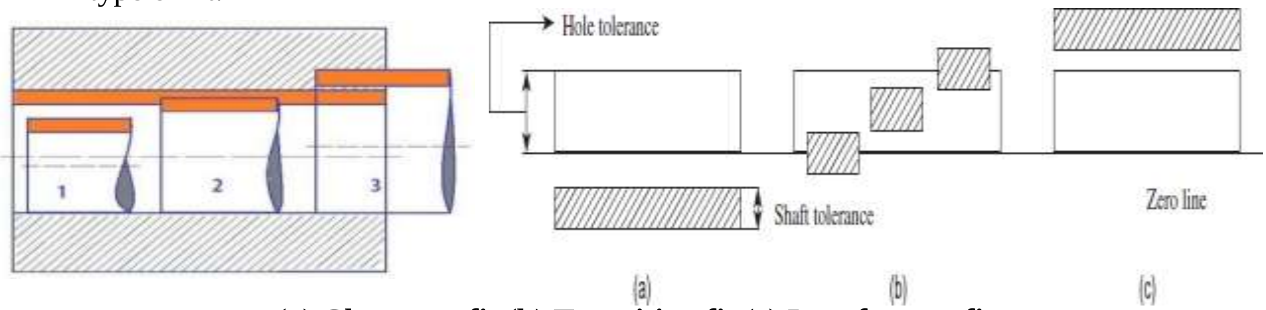
- To obtain the desired class of fits, either the size of the hole or the size of the shaft must vary.

Two types of systems are used to represent three basic types of fits, clearance, interference, and transition fits.

- Hole basis system
- Shaft basis system.

Hole Basis systems

- The size of the hole is kept constant and the shaft size is varied to give various types of fits.
- Lower deviation of the hole is zero, i.e. the lower limit of the hole is same as the basic size.
- Two limits of the shaft and the higher dimension of the hole are varied to obtain the desired type of fit.



(a) Clearance fit (b) Transition fit (c) Interference fit

This system is widely adopted in industries, easier to manufacture shafts of varying sizes to the required tolerances. Standard-size plug gauges are used to check hole sizes accurately. **(5)**

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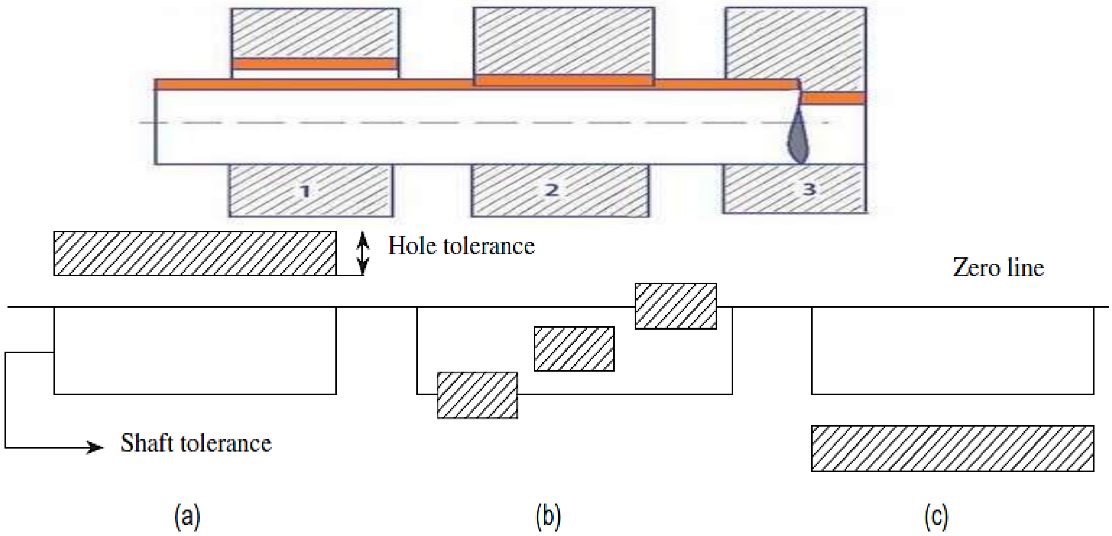
Shaft Basis systems

- The size of the shaft is kept constant and the hole size is varied to obtain various types of fits.
- Fundamental deviation or the upper deviation of the shaft is zero.
- System is not preferred in industries, as it requires more number of standard- size tools, like reamers, broaches, and gauges, increases manufacturing and inspection costs.



(a) Clearance fit (b) Transition fit (c) Interference fit **(5)**

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Answer: 4

$S_{yt} = 380 \frac{N}{mm^2}$

$\sigma_x = 100$ MPa ; $\sigma_y = 40$ MPa ; $\tau_{xy} = 80$ MPa

Principal stresses:

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{100+40}{2} + \sqrt{\left(\frac{100-40}{2}\right)^2 + 80^2} = 155.44 \text{ MPa}$$ 2 marks

$$\sigma_2 = \frac{100+40}{2} - \sqrt{\left(\frac{100-40}{2}\right)^2 + 80^2} = -15.44 \text{ MPa}$$ 2 marks

Factor of Safety (n_f) by

(I) The maximum normal stress theory

$$n_f = \frac{380}{155.44} = 2.44$$ 2 marks

(II) Maximum shear stress theory:

$$n_f = \frac{380}{155.44 + 15.44} = 2.22$$ 2 marks

(III) The distortion energy theory:

$$n_f = \frac{380}{\sqrt{155.44^2 + 155.44 \times 15.44 + 15.44^2}} = 2.32$$ 2 marks

Answer 5:

$P = 60\ kN$

$\sigma_t = 80\ MPa$

$\tau = 55\ mPa$

$\sigma_c = 100\ MPa$

(I) Diameter of rods: (D)

$$D = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4\times 60\times 1000}{\pi\times 80}} = 30.90 \simeq 31\ mm$$ 1 Mark

(II) Enlarged diameter of rod (D_1)

$$D_1 = 1.1\times D = 1.1\times 31 = 34.1 \simeq 35\ mm$$ 1 mark

(III) Dimensions a and b:

$$a = 0.75D = 0.75\times 31 = 23.25 \simeq 24\ mm$$

$$b = 1.25D = 1.25\times 31 = 38.75 \simeq 39\ mm$$ 1 Mark

(IV) Diameter of pin:

$$d = \sqrt{\frac{2P}{\pi\tau}} = \sqrt{\frac{2\times 60\times 1000}{\pi\times 55}} = 26.35\ mm \simeq 27\ mm$$

$$d = \left(\frac{32}{\pi\sigma_b}\times\frac{P}{2}\left[\frac{b}{4}+\frac{a}{3}\right]\right)^{1/3} = \left[\frac{32}{\pi\times 80}\times\frac{60\times 1000}{2}\left[\frac{39}{4}+\frac{24}{3}\right]\right]^{1/3}$$

$$d = 40.77\ mm = 41\ mm$$

Hence $d = 41\ mm$ 2 marks

(V) Dimensions d_0 and d_1: $d_0 = 2d = 2\times 41 = 82\ mm$

$d_1 = 1.5d = 1.5\times 41 = 61.5 \simeq 62\ mm$ 1 mark

(VI) Check for stresses in eye:

$$\sigma_t = \frac{60\times 1000}{39\times(82-41)} = 37.52 < 80\ MPa;\quad \sigma_c = \frac{60\times 1000}{39\times 41} = 37.5 \le 55\ MPa$$

$$\tau = \frac{60\times 1000}{39\times 41} = 37.52\ MPa < 100\ MPa$$ 2 marks

(VII) Check for stress in fork:

$$\sigma_t = \frac{60\times 1000}{2\times 24\times(82-41)} = 30.48\ MPa < 80\ MPa;\quad \sigma_c = \frac{60\times 1000}{2\times 24\times 41} = 30.48\ MPa < 55\ MPa$$

$\tau = 30.48 < 100\ MPa$ Stresses are within Limits.
2 marks

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II Mid Term Examination-2021

B.Tech./ Semester – III Yr./V Sem. **Branch: ME**
Subject: Design of Machine Elements-I **Subject Code: 5ME4-04**
Time: 1½ Hours **Maximum Marks: 24**

A. Distribution of Course Outcome and Bloom's Taxonomy in Question Paper

Q.No	Questions	Marks	CO	BL
1.	Draw the neat sketch of cotter joint.	2	2	1
2.	What is bolt of uniform strength?	2	5	1
3.	What is 'self-locking' of power screw? What is the condition for self-locking?	2	5	1
4.	Write the expressions of bending stress at outer fibre and inner fibre in curved beam.	2	5	1
5.	A Semi-elliptical laminated spring is made of 50 mm wide and 3 mm thick plates. The length between the supports is 650 mm and the width of the band is 60 mm. The spring has two full length leaves and five graduated leaves. If the spring carries a central load of 1600 N, Find: i. Maximum stress in full length and graduated leaves for an initial condition of no stress in the leaves. The maximum stress if the initial stress is provided to cause equal stress when loaded. ii. The deflection.	3+1	3	3
6.	A flat key is used to connect a pulley to a 45 mm diameter shaft. The standard cross section of the key is 14x9 mm. The key is made of commercial steel ($S_{yt} = S_{yc} = 230$ N/mm^2) and the factor of safety is 3. Determine the length of the key on the basis of shear and compression considerations, if 15 kW power at 360 rpm is transmitted through the keyed joint. Assume ($S_{sy} = 0.5S_{yt}$)	2+2	4	3
7.	Design a muff coupling to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 ($S_{yt} = S_{yc} = 400$ N/mm^2). The sleeve is made of grey cast iron FG 200 ($S_{ut} = 200$ N/mm^2). The factor of safety for the shafts and key is 4.	4	4	3

	For the sleeve, the factor of safety is 6 based on ultimate strength			
8.	A right-angled bell-crank lever is to be designed to raise a load of 5 kN at the short arm end. The lengths of short and long arms are 100 and 450 mm respectively. The lever and the pins are made of steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. The length to diameter ratio of the fulcrum pin is 1.25:1. Calculate (i) The diameter and the length of the fulcrum pin (ii) The shear stress in the pin (iii) The dimensions of the boss of the lever at the fulcrum (iv) The dimensions of the cross-section of the lever. Assume that the arm of the bending moment on the lever extends up to the axis of the fulcrum.	2+2+2+2	3	3
9.	A transmission shaft is supported between two bearings, which are 750 mm apart. Power is supplied to the shaft through a coupling, which is located to the left of the left-hand bearing. Power is transmitted from the shaft by means of a belt pulley, 450 mm in diameter, which is located at a distance of 200 mm to the right of the left-hand bearing. The weight of the pulley is 300 N and the ratio of the belt tension of tight and slack sides is 2:1. The belt tensions act in vertically downward direction. The shaft is made of steel FeE300 (S_{yt} = 300 N/mm^2) and the factor of safety is 3. Determine the shaft diameter, if it transmits 12.5 kW power at 300 rpm from the coupling to the pulley. Assume (S_{sy} = 0.5S_{yt}).	8	4	3

BL – Bloom's Taxonomy Levels

(1- Remembering, 2- Understanding, 3 – Applying, 4 – Analyzing, 5 – Evaluating, 6 - Creating)

CO – Course Outcomes

B. Questions and Course Outcomes (COs) Mapping in terms of correlation

COs	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8	Q9
CO1	0	0	0	0	0	0	0	0	0
CO2	1	0	0	0	0	0	0	0	0
CO3	0	0	0	0	2	0	0	3	0
CO4	0	0	0	0	0	2	2	0	3
CO5	0	1	1	1	0	0	0	0	0

1-Low Correlation; 2- Moderate Correlation; 3- Substantial Correlation

C. Mapping of Bloom's Level and Course Outcomes with Question Paper

Bloom's Level Mapping		**CO Mapping**	
Bloom's Level	Percentage	CO	Percentage
BL1	22.22	CO1	0
BL2	0	CO2	5.56
BL3	77.78	CO3	33.33
BL4	00	CO4	44.44
BL5	00	CO5	16.67
BL6	00	CO6	00

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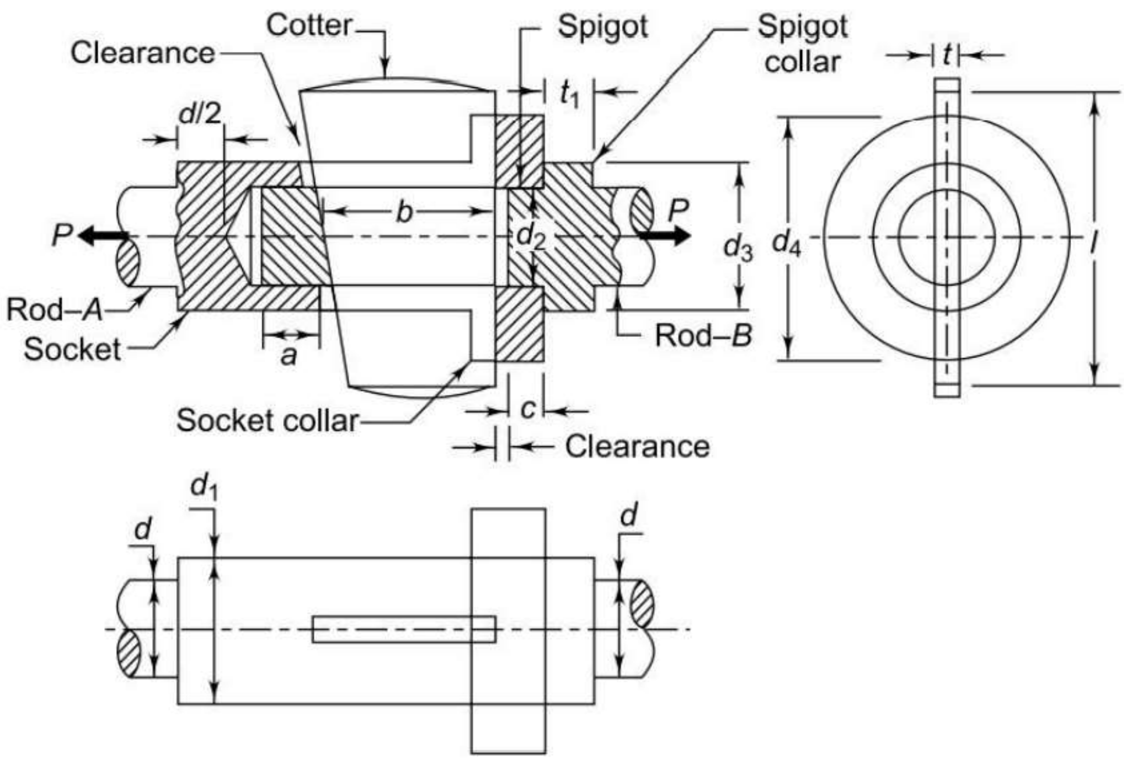
Solution of Second Mid Term Exam-2021

Part-A

Ans:1. Sketch of cotter joint **(2)**



Ans:2. The energy absorbed during elastic deformation is proportional to the square of the stress induced in the material and the volume of which is subjected to same stress level at different cross-sections in the bolt. It is called the bolt of uniform strength. In a bolt of uniform strength, the entire bolt is stressed to the same limiting value, thus resulting in maximum energy absorption. **(2)**

Ans:3. Self-locking of power screw

When friction angle $(\emptyset) \geq helix\ anglr\ (\alpha)$ a positive torque is required to lower the load. Under this condition, the load will not turn the screw and will not descend on its own unless an effort P is applied. In this case, the screw is said to be 'self-locking'. **(2)**

Ans:4. Bending stress at outer fibre and inner fibre in curved beam. **(2)**

The bending stress at the inner fibre is given by,

The bending stress at the outer fi $$\sigma_{bi} = \frac{M_b h_i}{AeR_i}$$

$$\sigma_{bo} = \frac{M_b h_o}{AeR_o}$$

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Part-B

Ans-5 Given Data:

$b = 50mm$, $t = 3mm$, $2L_1 = 650mm$, $l_1 = 60mm$

$n_f = 2$, $n_g = 5$, $2P = 1600 N$

To be find out: Ans (i) Case-1 Max. stress in full length leaves and graduated length leaves when no pre stress in leaves.

Case-2 Max. stress if the initial stress is provided

Ans-2 → deflection in the leaves

Ans: Case-1 Effective length

$$2L = 2L_1 - l_1$$

$$= 650 - 60 = 590mm$$

$$2L = 590, \quad L = 295$$

$$(\sigma_b)_f = \frac{18PL}{(3n_f + 2n_g)bt^2} = \frac{18 \times 800 \times 295}{(3 \times 2 + 2 \times 5) \times 50 \times (3)^2}$$

$$\boxed{(\sigma_b)_f = 590 N/mm^2}$$ [1 - Mark]

$$(\sigma_b)_g = \frac{12PL}{(3n_f + 2n_g) \cdot bt^2} = \frac{12 \times 800 \times 295}{(3 \times 2 + 2 \times 5) \times 50 \times (3)^2}$$

$$\boxed{(\sigma_b)_g = 393.33 N/mm^2}$$ [1 - Mark]

Case-2

$$\sigma_b = \frac{6PL}{n \cdot bt^2} = \frac{6 \times 800 \times 295}{7 \times 50 \times (3)^2}$$

$$\boxed{\sigma_b = 449.52 N/mm^2}$$ [1 - Mark]

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(ii) Deflection:

$$\delta = \frac{12PL^3}{Ebt^3(3n_f + 2n_g)}$$

$$\delta = \frac{12 \times 800 \times (295)^3}{210 \times 10^3 \times 50 \times (3)^3 (3 \times 2 + 2 \times 5)}$$

$$\boxed{\delta = 0.184\ mm} \quad - \boxed{1\text{-Marks}}$$

Ans-6 $d = 45\,mm$, cross-section of key $= 14 \times 9\ mm$

Material of key = commercial steel

$(S_{yt} = S_{yc} = 230\ N/mm^2)$

$F.O.S = 3$, $kW = 15$, $N = 360\ r.p.m$

$S_{sy} = 0.5 \times S_{yt}$

To be find out:- (i) length of key on basis of shear considerations

(ii) length of key on compression considerations.

Solution: $\sigma_t = \sigma_c = \frac{230}{3} = 76.66\ N/mm^2$

$\tau = 0.5 \times 76.66 = 38.33\ N/mm^2 \rightarrow \boxed{1\text{-Mark}}$

~~Cased~~ — $kW = \frac{2\pi N \cdot M_t}{60 \times 10^6} = \frac{2 \times \pi \times 360 \times M_t}{60 \times 10^6}$

$M_t = 398089.1720 \rightarrow \boxed{1\text{-Mark}}$

Ans (i) $\tau = \frac{2M_t}{dhl} = \frac{2.0 \times 398089.1720}{45 \times 14 \times l} = 38.33$

$$\boxed{l = 32.97\,mm}$$

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$\boxed{l = 32.97\,mm}$ — $\boxed{1\text{-Mark}}$

Ans-(ii) – $\sigma_c = \dfrac{4M_t}{d\,h\,l} \Rightarrow 76.66 = \dfrac{4 \times 398089.1720}{45 \times 9 \times l}$

$\boxed{l = 51.26\,mm}$ $\boxed{1\text{-Mark}}$

Ans-7 $kW = 25$, $N = 360\,r.p.m$, key material $= 30C8$

$S_{yt} = S_{yc} = 400\,N/mm^2$. Sleeve material $= FG200$, F.O.S for key $= 4$ and for sleeve $= 6$

Solution:- For key and shaft – $\sigma_t = \sigma_c = \dfrac{S_{yt}}{F_s} = \dfrac{S_{yc}}{F.S} = \dfrac{400}{4} = 100\,N/mm^2$

$\tau = \dfrac{0.5 \times S_{yt}}{F_s} = \dfrac{0.5 \times 400}{4} = 50\,N/mm^2$

Permissible stress for sleeve – $\tau = \dfrac{S_{su}}{F_s} = \dfrac{0.5 \times S_{ut}}{F_s} = \dfrac{0.5 \times 200}{6} = 16.67\,N/mm^2$

Diameter of each shaft: → $M_t = \dfrac{60 \times 10^6 (kW)}{2\pi n} = \dfrac{60 \times 10^6 \times 25}{2\pi \times 360}$ → $\boxed{1\text{-Mark}}$

$M_t = 663145.60\,N\text{-}mm$

$\tau = \dfrac{16 M_t}{\pi d^3} \Rightarrow 50 = \dfrac{16\,(663145.60)}{\pi d^3}$ $d = 40.73 = 45\,mm$ $\boxed{1\text{-Mark}}$

Sleeve dimensions – $D = 2d + 13 = 2.0 \times 45 + 13 = 103 = 105\,mm$

$L = 3.5d = 3.5 \times (45) = 157.5 = 160\,mm$

Check sleeve for torsional shear stress $\tau = \dfrac{M_t . r}{J} = \dfrac{663145.60 \times 52.5}{11530626.79}$

$\tau = 3.02\,N/mm^2$ $\boxed{\tau < 16.67\,N/mm^2}$ $\boxed{1\text{-Mark}}$

Dimensions of key → [From D.D.H.B table 4.6 page No – 74] standard dimension is 14×9 mm. and length of key in each shaft is one half of the length of sleeve $l = \dfrac{L}{2} = \dfrac{160}{2} = 80\,mm$

dimensions of key $14 \times 9 \times 80\,mm$, $\sigma_c = \dfrac{4M_t}{dhl}$ $\boxed{1\text{-Mark}}$

$\tau = \dfrac{2M_t}{dbl} = \dfrac{2.0 \times 663145.60}{45 \times 14 \times 80} = 26.32\,N/mm^2$ $\sigma_c = \dfrac{4 \times 663145.60}{45 \times 9 \times 80} = 81.87\,N/mm^2$

Part-C

Ans:8 Given : $S_{yt} = 400\ N/mm^2$ $(fs) = 5$ $F = 5\ kN$

For lever, long arm $= 450$ mm short arm $= 100$ mm

$d/b = 3$, For pin $p = 10\ N/mm^2$ $l_1/d_1 = 1.25$.

Step.I Calculation of permissible stresses for the pin

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80\ N/mm^2 \quad (1)$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5 S_{yt}}{(fs)} = \frac{0.5(400)}{5} = 40\ N/mm^2$$

Step. II Calculation of forces acting on the lever

Taking moment about the axis of the fulcrum,

$$(5\times10^3)(100) = P \times 450 \quad \therefore P = 1111.11\ N$$

$$R = \sqrt{(5000)^2 + (1111.11)^2} = 5121.97\ N \quad (1)$$

Step. III. Diameter and length of fulcrum pin

$$R = p(d_1 \times l_1) \quad \text{or} \quad R = p(d_1 \times l_1)$$

$$\text{so} \quad 5121.97 = 10(d_1 \times 1.25 d_1) \quad (2)$$

$$\therefore d_1 = 20.24\ mm$$

$$l_1 = 1.25 d_1 = 1.25(20.24) = 25.30\ mm$$

Step IV: Shear stress in pin

The pin is subjected to double shear. The shear stress in the pin

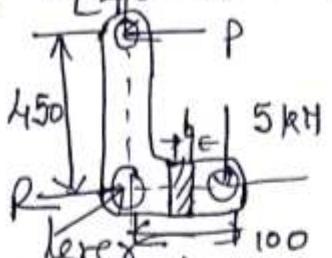
$$\tau = \frac{R}{2(\frac{\pi}{4}d_1^2)} = \frac{5121.97}{2[\frac{\pi}{4}(20.24)^2]} = 7.96\ N/mm^2 \quad (1)$$

Step V: Dimensions of the boss

inner dia $= 21$ mm,

outer dia $= 42$ mm (1)

length $= 26$ mm



Step VI: Dim of cross section of lever

$$d = 3b, \quad \sigma_b = \frac{M_b \cdot y}{I} \quad \text{or} \quad 80 = \frac{(5000 \times 100)(1.5b)}{\frac{1}{12}(b)(3b)^3}$$

$$b = 16.09\ mm$$

$$d = 3b = 3(16.09) = 48.27\ mm \quad (2)$$

Soln 9:

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5 S_{yt}}{(fs)} = \frac{0.5(300)}{3} = 50\, N/mm^2 \quad (1)$$

$$M_t = \frac{60\times10^6 (kW)}{2\pi n} = \frac{60\times10^6 (12.5)}{2\pi(300)} = 397\,887.36\, N\text{-}mm$$

$$M_t = (P_1 - P_2)R \quad \therefore 397887.36 = (P_1 - P_2)225$$

$$\therefore (P_1 - P_2) = 1768.39\, N \quad \text{and} \quad \left(\frac{P_1}{P_2}\right) = 2 \quad (2)$$

from eqn

$$P_1 = 3536.78\, N$$

$$P_2 = 1768.39\, N$$

$$\therefore (P_1 + P_2 + W) = 5605.17\, N$$

So $(P_1 + P_2 + W) \times 200 = R_6 \times 750$

$$\therefore R_6 = 1494.71\, N \quad (2)$$

$$M_b = R_6 \times 550 = 1494.71 \times 550 = 822091.6\, N\text{-}mm \quad (1)$$

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(M_b)^2 + (M_t)^2}$$

$$50 = \frac{16}{\pi d^3}\sqrt{(822091.6)^2 + (397887.36)^2}$$

$$\therefore d = 45.31\, mm \quad \text{(Ans)} \quad (2)$$

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B

I Mid Term Examination-2020

B.Tech./ Semester – III Yr./V Sem. **Branch: ME**
Subject: Design of Machine Elements-I **Subject Code: 5ME4-04**
Time: 1½ Hours **Maximum Marks: 50**

Q.No	Questions	Marks	CO	BL	PI
1. (a)	Explain the various mechanical properties of engineering materials.	8	CO1	L2	1.3.1
1. (b)	Explain IS designation of: **(i)** Grey Cast Iron having minimum tensile strength of 220MPa **(ii)** Plain carbon steel having average content of 0.4% carbon and 0.6% manganese	2	CO1	L2	3.1.4
2.	Design a spigot and socket type cotter joint for axial load of 75 kN which alternately changes from tensile to compressive. Allowable stresses for the material used are 50 MPa in tensions, 40 MPa in shear, and 105 MPa in crushing.	10	CO2	L3	1.4.1, 3.4.1
3.	Design a knuckle joint to connect two round rods subjected to a tensile load of 100 kN. The permissible stresses may be taken as 75 MPa in tension, 50 MPa in shear and 135 MPa in crushing.	10	CO2	L3	1.4.1, 3.4.1
4.	A lever-loaded safety valve is mounted on the boiler to blow off at a pressure of 1.5 MPa gauge. The effective diameter of the opening of the valve is 50 mm. The distance between the fulcrum and the dead weights on the lever is 1000 mm. The distance between the fulcrum and the pin connecting the valve spindle to the lever is 100 mm. The lever and the pin are made of plain carbon steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure at the pins in the lever is 25 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. Design a suitable lever for the safety valve.	10	CO3	L3	1.4.1, 3.4.1
5.	A locomotive spring has an overall length of 1.1 m and sustains a load a load of 75 kN at its center. The spring has 3 full length leaves and 15 graduated leaves with a central band of 100 mm wide. All leaves are to be stressed to 420 N/mm^2 when fully loaded. The ratio of total depth to width is to be 2. Take E=210 GPa and Bearing pressure (P_b)=10 N/mm^2, (i) determine the width and thickness of leaves (ii) determine the Nip (iii) what load is exerted on the band after spring is assembled? (iv) eye diameter using bear consideration only (v) maximum deflection.	10	CO3	L3	1.4.1, 3.4.1

BL – Bloom's Taxonomy Levels
(1- Remembering, 2- Understanding, 3 – Applying, 4 – Analyzing, 5 – Evaluating, 6 - Creating)

Solution of I Mid Term Exam-2020 (SET-B)

Ans. 1(a): Mechanical Properties of Metals

The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load. Mechanical properties are given as:

xiv. Strength. It is the ability of a material to resist the externally applied forces without breaking or yielding. The internal resistance offered by a part to an externally applied force is called stress.

xv. Stiffness or rigidity: It is the ability of the material to resist deformation under the action of an external load. The modulus of elasticity is the measure of stiffness.

xvi. Elasticity. It is the property of a material to regain its original shape after deformation when the external forces are removed. This property is desirable for materials used in tools and machines. It may be noted that steel is more elastic than rubber.

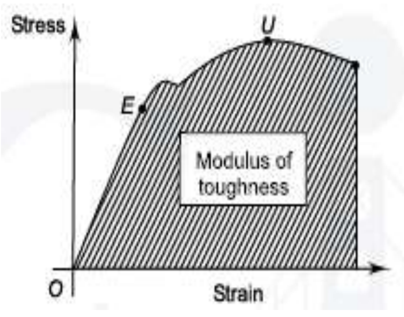
xvii. Plasticity. It is property of a material which retains the deformation produced under load permanently. This property of the material is necessary for forgings, in stamping images on coins and in ornamental work.

xviii. Ductility. It is the property of a material enabling it to be drawn into wire with the application of a tensile force. A ductile material must be both strong and plastic. The ductility is usually measured by the terms, percentage elongation and percentage reduction in area. The ductile material commonly used in engineering practice (in order of diminishing ductility) are mild steel, copper, aluminum, nickel, zinc, tin and lead.

xix. Brittleness. It is the property of a material which shows negligible plastic deformation before fracture takes place. Brittleness is the opposite to ductility. In ductile materials, failure takes place by yielding. Brittle components fail by sudden fracture. A tensile strain of 5% at fracture in a tension test is considered as the dividing line between ductile and brittle materials. Cast iron is a brittle material.

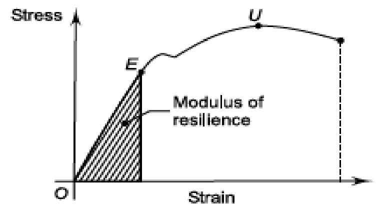
xx. Malleability. It is a special case of ductility which permits materials to be rolled or hammered into thin sheets. A malleable material should be plastic but it is not essential to be so strong. The malleable materials commonly used in engineering practice (in order of diminishing malleability) are lead, soft steel, wrought iron, copper and aluminum.

xxi. Toughness. It is the ability of the material to absorb energy before fracture takes place. This property is essential for machine components which are required to withstand impact loads. Tough materials have the ability to bend, twist or stretch before failure takes place. All structural steels are tough materials. Toughness is measured by a quantity called modulus of toughness. Modulus of toughness is the total area under stress–strain curve in a tension test, which also represents the work done to fracture the specimen. In practice, toughness is measured by the Izod and Charpy impact testing machines.



xxii. Machinability. It is the property of a material which refers to a relative case with which a material can be cut. The machinability of a material can be measured in a number of ways such as comparing the tool life for cutting different materials or thrust required to remove the material at some given rate or the energy required to remove a unit volume of the material. It may be noted that brass can be easily machined than steel.

xxiii. Resilience. It is the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded. Resilience is measured by a quantity, called modulus of resilience, which is the strain energy per unit volume. It is represented by the area under the stress–strain curve from the origin to the elastic limit point. This property is essential for spring materials.



xxiv. Creep. When a part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called creep. This property is considered in designing internal combustion engines, boilers and turbines.

xxv. Fatigue. When a material is subjected to fluctuating stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as fatigue. The failure is caused by means of a progressive crack formation which are usually fine and of microscopic size. This property is considered in designing shafts, connecting rods, springs, gears, etc.

xxvi. Hardness. It is defined as the resistance of the material to penetration or permanent deformation. It usually indicates resistance to abrasion, scratching, cutting or shaping. The hardness of a metal may be determined by the following tests:

(a) Brinell hardness test,

(b) Rockwell hardness test,

(c) Vickers hardness (also called Diamond Pyramid) test, and (**At least 8 mechanical properties, 8 marks)**

(d) Shore scleroscope.

Ans. 1(b): IS designation

(i) FG 220 **(1 mark)** (ii) 40C6 (**1 mark)**

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Ans(2): Given

$P = 75\,kN = 75 \times 1000\,N$; $\sigma_t = 50\,MPa$; $\sigma_c = 105\,MPa$

$\tau = 40\,MPa$

(I) Diameter of rods

$$d = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4 \times 75000}{\pi \times 50}} = 43.70\,mm \simeq 44\,mm$$ 1 Mark

(ii) Thickness of cotter

$$t = 0.31d = 0.31 \times 44 = 13.64\,mm \simeq 14\,mm$$ $\frac{1}{2}$ Mark

(III) Diameter of spigot (d_2):

$$P = \left(\frac{\pi}{4} d_2^2 - d_2 t\right) \sigma_t$$

$$75000 = \frac{\pi}{4} d_2^2 \times 50 - d_2 \times 14 \times 50$$

$$d_2 = 53.51\,mm \simeq 54\,mm$$ 1 Mark

(iv) outer diameter of socket (d_1):

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)\,t\right] \sigma_t$$

$$75000 = \left[\frac{\pi}{4}(d_1^2 - 54^2) - (d_1 - 54) \times 14\right] \times 50$$

$$\Rightarrow d_1 = 71.704\,mm \simeq 72\,mm$$ 1 mark

(V) Diameters of spigot collar (d_3) and socket collar (d_4)

$$d_3 = 1.5d = 1.5 \times 44 = 66\,mm$$

$$d_4 = 2.4d = 2.4 \times 44 = 105.6\,mm \simeq 106\,mm$$ 1 Mark

(VI) Dimensions a and c

$$a = c = 0.75d = 0.75 \times 44 = 33\,mm$$ $\frac{1}{2}$ mark

(VII) Width of cotter

$$b = \frac{P}{2\tau t} = \frac{75000}{2 \times 40 \times 14} = 66.96\,mm \simeq 67\,mm$$ (Shear consideration)

or $$b = \sqrt{\frac{3P}{t\sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]} = 84.41\,mm \simeq 85\,mm$$ (Bending consideration)

1.5 marks

Then. width of cotter is maximum in bending, so

$$b = 85\,mm$$

(VIII) check for crushing and shear stress in spigot end.

$$\sigma_c = \frac{P}{t d_2} = \frac{75000}{14 \times 54} = 99.206 < 105\,MPa$$ MPa

$$\tau = \frac{P}{2 a d_2} = \frac{75000}{2 \times 33 \times 54} = 21.044\,\frac{N}{mm^2} < 40\,\frac{N}{mm^2}$$ 1.5 marks

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(IX) Check for crushing and shear stresses in socket end

$$\sigma = \frac{P}{(d_4 - d_1)t} = \frac{75000}{(106-54)\times 14} = 103.022 \frac{N}{mm^2} < 105 \frac{N}{mm^2}$$

$$\tau = \frac{P}{2(d_4 - d_1)c} = \frac{75000}{2\times(106-54)\times 33} = 21.853 \frac{N}{mm^2} < 40 \frac{N}{mm^2}$$ 1.5 marks

The stresses induced in spigot and the socket ends are within limits.

(X) Thickness of spigot collar

$$t_1 = 0.45d = 0.45\times 44 = 19.8\,mm \simeq 20\,mm$$ ½ mark

The taper for the cotter is 1 in 32.

Ans.(3): Given

$P = 100\,kN = 100\times 10^3\,N$; $\sigma_t = 75\,MPa$; $\tau = 50\,MPa$; $\sigma_c = 135\,MPa$

(I) Diameter of rods (D)

$$D = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4\times 100\times 1000}{\pi\times 75}} = 41.20\,mm \simeq 42\,mm$$ 1 Mark

(II) Enlarged diameter of rods (D_1)

$$D_1 = 1.1D = 1.1\times 42 = 46.2\,mm \simeq 47\,mm$$ (½) mark

(III) Dimensions a and b

$$a = 0.75D = 0.75\times 42 = 31.5\,mm \simeq 32\,mm$$

$$b = 1.25D = 1.25\times 42 = 52.5\,mm \simeq 53\,mm$$ 1 mark

(iv) Diameter of pin (d) from shear and bending consideration

$$d = \sqrt{\frac{2P}{\pi\tau}} = \sqrt{\frac{2\times 100\,000}{\pi\times 50}} = 35.68\,mm \simeq 36\,mm$$

$$d = \sqrt[3]{\frac{32}{\pi\sigma_b}\times\frac{P}{2}\left[\frac{b}{4}+\frac{a}{3}\right]} \qquad \sigma_b = 75\frac{N}{mm^2}$$

$$d = 54.56\,mm \simeq 55\,mm$$ 2 marks

Diameter of pin is Larger in bending, so

$$d = 55\,mm$$

(V) Dimensions d_0 and d_1

$$d_0 = 2d = 2\times 55 = 110\,mm$$ 1 mark

$$d_1 = 1.5d = 1.5\times 55 = 82.5\,mm \simeq 83\,mm$$

(VI) Check for stresses in eye

$$\sigma_t = \frac{P}{b(d_0-d)} = \frac{100\,000}{53\times(110-55)} = 34.305\frac{N}{mm^2} < 75\frac{N}{mm^2}$$

$$\sigma_c = \frac{P}{bd} = \frac{100\,000}{53\times 55} = 34.305\frac{N}{mm^2} < 135\frac{N}{mm^2}$$

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$$\tau = \frac{P}{b(d_o-d)} = \frac{1,00,000}{53\times(110-55)} = 34.305\ \frac{N}{mm^2} < 50\ \frac{N}{mm^2}$$ 2 marks

(VII) Check for stresses in fork

$$\sigma_t = \frac{P}{2a(d_o-d)} = \frac{100000}{2\times32(110-55)} = 28.409\ \frac{N}{mm^2} < 75\ \frac{N}{mm^2}$$

$$\sigma_c = \frac{P}{2ad} = \frac{10,0000}{2\times32\times55} = 28.409\ \frac{N}{mm^2} < 135\ \frac{N}{mm^2}$$

$$\tau = \frac{P}{2a(d_o-d)} = \frac{100000}{2\times32\times(110-55)} = 28.409\ \frac{N}{mm^2} < 50\ \frac{N}{mm^2}$$ 2 marks

it is observed that stresses are within limits. so design is safe.

Ans (4): Given

$S_{yt} = 400\,MPa$ $FS = 5$

for valve: $d = 50mm$; $P_v = 1.5\,MPa$

for lever $l_1 = 1000mm$; $l_2 = 100mm$; $\frac{d}{b} = 3$; for pin $P_b = 25\ \frac{N}{mm^2}$

(I) Permissible stresses

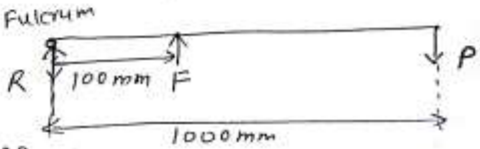
$$\sigma_t = \frac{S_{yt}}{F_s} = \frac{400}{5} = 80\ \frac{N}{mm^2}$$

$$\tau = \frac{0.5\,S_{yt}}{F_s} = \frac{0.5\times400}{5} = 40\ \frac{N}{mm^2}$$ 1 mark

(II) Forces acting on lever :

$$F = \frac{\pi}{4}d^2 P_v = \frac{\pi}{4}(50)^2\times1.5$$

$F = 2945.24N$ 1 mark



Taking moment about fulcrum

$$P\times l_1 = F\times l_2 \Rightarrow P\times1000 = 2945.24\times100$$

$\Rightarrow P = 294.52N$ 1 mark

Equilibrium of forces in vertical direction

$$F = R+P \Rightarrow R = F-P = 2650.72\,N$$ 1 mark

(III) Diameter and length of pin:
(d_1) (l_f)

considering $\frac{l_f}{d_1} = 1 \Rightarrow l_f = d_1$

Bearing Force $= l_f d_1 p = d_1^2\times25$

Bearing force $=$ max. of R, P, F so F = Bearing Force 2 marks

$$2945.24 = 25d_1^2 \Rightarrow d_1 = 10.85mm \simeq 12mm$$

Check for shear stress: $\tau < \tau_p$ $\tau = \frac{F}{2\times\frac{\pi}{4}d_1^2} = \frac{2945.24}{2\times\frac{\pi}{4}(12)^2} = 13.02\ \frac{N}{mm^2} < 40\ \frac{N}{mm^2}$ 1 mark

(iv) Cross-section of lever:

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

$M = M_{max} = P \times (1000-100) = 294.52 \times 900 = 265068$ N-mm

$I = \frac{bd^3}{12} \Rightarrow d = 3b$

$I = \frac{b(3b)^3}{12}$; $y = \frac{d}{2} = \frac{3b}{2} = 1.5b$

$\sigma_b = \frac{265068 \times 12}{b \times (3b)^3} \times 1.5b = 80$ 3 marks

$b = 13.02$ mm $\simeq 14$ mm ; $d = 3 \times 14 = 42$ mm Ans.

Ans.(5) Given:

$2L_1 = 1.1$ m $= 1100$ mm ; $2P = 75$ kN $\Rightarrow P = \frac{75}{2} = 37.5$ kN $= 37.5 \times 1000$ N

$n_f = 3$; $n_g = 15$; $l = 100$ mm ; $\sigma_b = 420 \frac{N}{mm^2}$; $\frac{nt}{b} = 2$

$E = 210$ GPa $= 210 \times 10^3$ MPa ; $p_b = 10 \frac{N}{mm^2}$

(i) Width and thickness of leaves

$\sigma_b = \frac{6PL}{nbt^2} \Rightarrow n = n_f + n_g = 3 + 15 = 18$; $2L = 2L_1 - l$

$2L = 1100 - 100$

$\frac{18t}{b} = 2$; $2L = 1000$; $L = 500$ mm

$420 = \frac{6 \times 37.5 \times 1000 \times 500}{18 \times 9t \times t^2} \Rightarrow b = 9t$

$\Rightarrow t^3 = 1653.44 \Rightarrow t = 11.82$ mm $\simeq 12$ mm 2 marks

$b = 9 \times 12 = 108$ mm

(ii) Nip (C):

$C = \frac{2PL^3}{Enbt^3} = \frac{2 \times 37.5 \times 1000 \times 500^3}{210 \times 10^3 \times 18 \times 108 \times 12^3} = 13.29$ mm 2 marks

(iii) Load is exerted on the band after spring is assembled

$P_i = \frac{2 n_f n_g P}{n(3n_f + 2n_g)} = \frac{2 \times 3 \times 15 \times 37.5 \times 1000}{18(3 \times 3 + 2 \times 15)} = 4807.692$ N 2 marks

(iv) Eye diameter (d) using bear consideration

$P = d\, b\, p_b$

$\Rightarrow 37.5 \times 1000 = d \times 108 \times 10 \Rightarrow d = 34.72$ mm $\simeq 35$ mm 2 marks

(v) Maximum deflection (δ)

$\delta = \frac{12PL^3}{Ebt^3(3n_f + 2n_g)} = \frac{12 \times 37.5 \times 1000 \times 500^3}{210 \times 10^3 \times 108 \times (12)^3 (3 \times 3 + 2 \times 15)} = 36.802$ mm 2 marks

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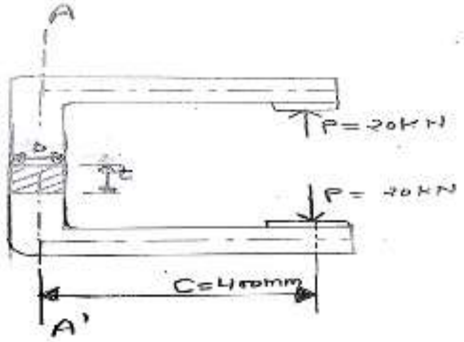
Tutorial Sheets (with EMD Analysis)

Sheet No.1	B.Tech. V Semester (2022-23) Basics of Mechanics of Solid & Design

1. Draw the Stress strain diagram of mild steel and show following points on it. Also give definition of these points.
   (i) Elastic limit, (ii) Proportional limit(iii) Yield point (iv) Ultimate Strength (v) Fracture point **(E)**
2. Define Proof strength and represent it on stress strain diagram. Explain the importance of proof strength with suitable example. **(M)**
3. Explain the various mechanical properties of engineering materials. **(E)**
4. The stresses induced at a critical point in a machine component made of steel (S_{yt}= 380 N/mm^2) are as follows
   $\sigma_x = 100\ MPa,\ \sigma_y = 40\ MPa, \tau_{xy} = 80 MPa$ calculate the factor of safety by
   (i) The maximum normal stress theory, (ii) The maximum shear stress theory(iii) The distortion energy theory **(D)**
5. A steel rod having circular cross section of 300mm^2 and length of 150 m is suspended vertically from one end. It supports a tensile load of 20kN at lower end. If unit mass of steel is 7850 kg/m^3 and E=200000MN/m^2, find elongation of rod. (Hint: elongation due to its own wt& due to 20kN). **(D)**
6. What punch force is required to punch a hole of 20 mm diameter in a plate of 25mm thickness? The shear strength is 350 MN/m^2. **(M)**
7. A bar of 50 mm diameter fixed at one end is subjected to a torque of 1kN-m. In addition to it an axial pull of 15kN and 3kN bending load at free end also acts on it. Length of bar = 250mm. Determine the following at critical points. (i) Maximum normal stress. (ii) Minimum normal stress (iii) Max. Shear stress. **(M)**
8. A solid shaft in a rolling mill transmit 20 kW at 2 Hz. Determine the diameter of the shaft if the shear stress is not exceed 40 MPa and the angle of twist is limited to 6^0 in a length of 3 m. G = 83 GPa. **(M)**
9. A flywheel weighing 500 Kg is mounted on a shaft of 75 mm and midway bearing 0.6 m apart. If the shaft is transmitting 30 kW at 360 rpm, calculate the principal stress and the maximum shear stress at the ends of a vertical diameter in a plane close to the flywheel. **(M)**
10. A load of 20kN acts on the frame of press as shown in figure. The frame has equal width and thickness at all sections. If width =150mm & thickness=25mm, determine the stress at section AA'. **(M)**



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Sheet No.2	B.Tech. V Semester (2022-23) Cotter and Knuckle Joint

1. Name the different types of cotter joint. Draw the sectional view of a Spigot-Socket Cotter joint and write its proportional dimensions. **(M)**
2. Write the design procedure for Spigot-Socket cotter joint with neat sketches. **(E)**
3. It is required to design a Spigot-Socket Cotter Joint to connect two steel rods of equal diameter. Each rod is subjected to an axial tensile force of 50 kN. Design the joint and specify its main dimensions. **(D)**
4. Two rods, made of plain carbon steel 40C8 (S_{yt} = 380N/mm^2), are to be connected by means of a cotter joint. The diameter of each rod is 50mm and the cotter is made from a steel plate of 15mm thickness. Calculate the dimension of the socket end making the following assumptions:
   i. The yield strength in compression is twice of the tensile yield strength; and
   ii. The yield strength in shear is 50% of the tensile yield strength.
   iii. The factor of safety is 6. **(M)**
5. Draw a neat sectional view of Knuckle joint and write its proportional dimensions. **(E)**
6. Write the design procedure of knuckle joint with neat sketches. **(E)**
7. A Trolley is connected to a Tractor by means of a Knuckle Joint. Knuckle Joint is subjected to an axial tensile force of 50 kN. In Joint, a small angular movement between axes is permissible. Design the joint and specify the dimensions of its components. Select suitable materials for the Knuckle Joint. **(D)**
8. Two rods are connected by means of a knuckle joint. The axial force P acting on the rods is 25kN. The rods and the pin are made of plain carbon steel 45C8 (S_{yt} = 380N/mm^2) and the factor of safety is 2.5. The yield strength in shear is 57.7% of the yield strength in tension. Calculate: (i) the diameters of the rods, and (ii) The diameter of the pin. **(D)**

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Sheet No.3	**B.Tech. V Semester (2022-23)** **Design of members in Bending (Beams, levers and laminated springs)**

1. A bracket, made of steel FeE 200 (S_{yt}= 200 N/mm^2) and subjected to a force of 5 kN acting at an angle of 30° to the vertical, is shown in **Fig.1**. The factor of safety is 4. Determine the dimensions of the cross-section of the bracket. **(M)**
2. A Cast iron beam having ultimate strength of 200 MPa is shown in **Fig.2** Determine the dimensions of cross-section using maximum principal theory. Take the factor of safety 2.5 **(D)**

RTU-2016

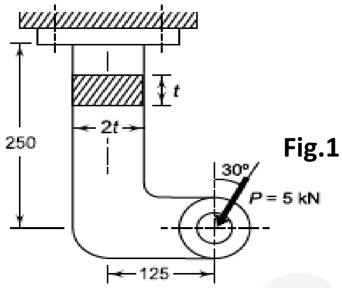

Fig.1.

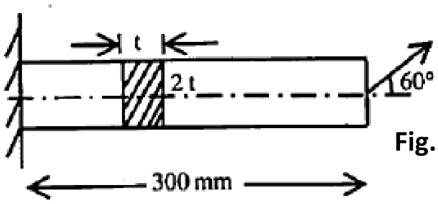

Fig.2

3. As imply supported beam of rectangular section carries a UDL of 12KN/m run over entire length and point load of 10KN at 3m from the left support. The length of beam is 8 m. If the depth is two times the width and the stress in the beam is not to exceed 8 N/mm^2.Find the suitable dimensions of the beam.**(M)**
4. A lever-loaded safety valve is mounted on the boiler to blow off at a pressure of 1.5 MPa gauge. The effective diameter of the opening of the valve is 50 mm. The distance between the fulcrum and the dead weights on the lever is 1000 mm. The distance between the fulcrum and the pin connecting the valve spindle to the lever is 100 mm. The lever and the pin are made of plain carbon steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure at the pins in the lever is 25 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. Design a suitable lever for the safety valve. **(D)**
5. Design a right angled bell crank lever having one arm 500 mm and the other 150 mm long. The load of 5 kN is to be raised acting on a pin at the end of 500 mm arm and the effort is applied at the end of 150 mm arm. The lever consists of a steel forging, turning on a point at the fulcrum. The permissible stresses for the pin and lever are 84 MPa in tension and compression and 70 MPa in shear. The bearing pressure on the pin is not to exceed 10 N/mm^2. **RTU-2014** **(M)**
6. A cranked lever, as shown in 15.10, has the following dimensions: Length of the handle = 300 mm; Length of the lever arm = 400 mm; Overhang of the journal = 100 mm; If the lever is operated by a single person exerting a maximum force of 400 N at a distance of 1/3rd length of the handle from its free end, find: (i). Diameter of the handle, (ii). Cross-section of the lever arm, and (iii). Diameter of the journal. The permissible bending stress for the lever material may be taken as 50 MPa and shear stressfor shaft material as 40 MPa. **RTU-2018** **(D)**
7. A bell crank lever is subjected to a force of 7.5 kN at the short arm end. The lengths of the short and long arms are 100 and 500 mm respectively. The arms are at right angles to each other. The lever and the pins are made of steel FeE 300 (S_{yt} = 300 N/mm^2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 4: 1. The length to diameter ratio of the fulcrum pin is 1.5: 1. Calculate: (i) the diameter and the length of the fulcrum pin (ii) the shear stress in the pin (iii) the dimensions of the boss of the lever at the fulcrum pin (iv) the dimensions of the cross-section of the lever Assume that the arm of the bending moment on the lever extends up to the axis of the fulcrum. **(M)**

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8. A locomotive spring has an overall length of 1.1 m and sustains a load a load of 75 kN at its center. The spring has 3 full length leaves and 15 graduated leaves with a central band of 100 mm wide. All leaves are to be stressed to 420 N/mm^2 when fully loaded. The ratio of the spring depth to width is to be 2. Take E=210 GPa and Bearing pressure (P_b)=10 N/mm^2, (i) determine the width and thickness of leaves (ii) determine the Nip (iii) what load is exerted on the band after spring is assembled? (iv) eye diameter using bear consideration only (v) maximum deflection. **RTU-2014** **(E)**
9. A semi elliptic laminated leaf spring has an eye to eye span of 1.2 m and supports a central load of 20 kN, for the purpose three full length and six graduated leaves including master leaf are used. Width of the central brand is 0.2 m. The width to thickness ratio for each leaf is 6. The allowable stresses are 200 N/mm^2. Take E = 2 x 105 N/mm^2. Calculate the section of leaves and deflection at full load if: (i) Leaves are not stressed initially (ii) Leaves are stressed initially for equalized stresses at maximum load.
RTU-2012 **(M)**
10. Design a leaf spring for the following specifications: Total load = 140 kN; Number of springs supporting the load = 4; Maximum number of leaves = 10; Span of the spring = 1000 mm; Permissible deflection = 80 mm. Take Young's modulus, E = 200 kN/mm^2 and allowable stress in spring material as 600 MPa. **(M)**

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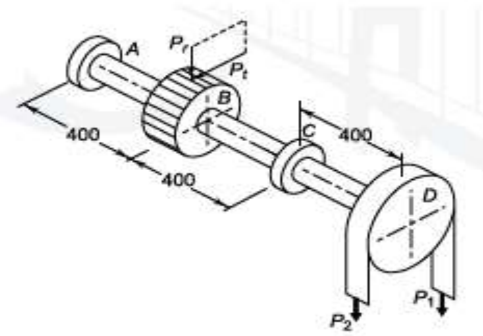
Sheet No.4	**B.Tech. V Semester (2022-23)** **Design of Shafts, Keys and Couplings**

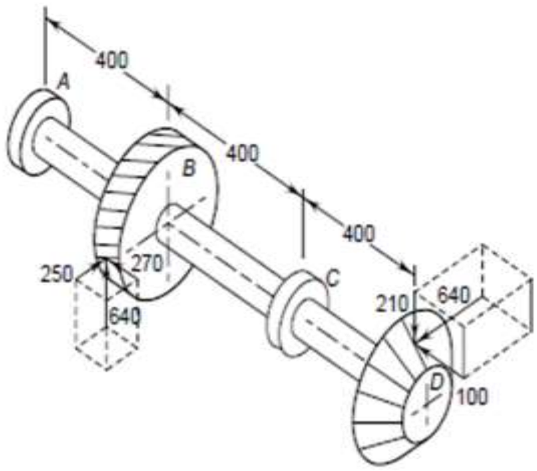
1. A line shaft rotating at 200 rpm is to transmit 20 kW power. The allowable shear stress for the shaft material is 42 N/mm^2. Determine the diameter of the shaft. **(E)**
2. Find the diameter of a solid steel shaft to transmit 20 kW at 200 r.p.m. The ultimate shear stress for the steel may be taken as 360 MPa and a factor of safety as 8. If a hollow shaft is to be used in place of the solid shaft, find the inside and outside diameter when the ratio of inside to outside diameter is 0.5.**(M)**
3. A solid circular shaft is subjected to a bending moment of 3000 N-m and a torque of 10 000 N-m. The shaft is made of 45C8 steel having ultimate tensile stress of 700 MPa and an ultimate shear stress of 500 MPa. Assuming a factor of safety as 6, determine the diameter of the shaft. **(M)**
4. A 15 kW, 960 r.p.m. motor has a mild steel shaft of 40 mm diameter and the extension being 75 mm. The permissible shear and crushing stresses for the mild steel key are 56 MPa and 112 MPa. Design the keyway in the motor shaft extension. Check the shear strength of the key against the normal strength of the shaft. **(M)**
5. A transmission shaft supporting a spur gears B and the pulley D is shown in Fig. The shaft is mounted on two bearings A and C. The diameter of the pulley and the pitch circle diameter of the gear are 450 mm and 300 mm respectively.The pulley transmits 20 kW power at 500 rpm to the gear. P1 and P2 are belt tensions in the tight and loose sides, while P_t and P_r are tangential and radial components of gear tooth force. Assume,



$$P_1 = 3P_2 \quad and \quad P_r = P_t \tan(20^\circ)$$

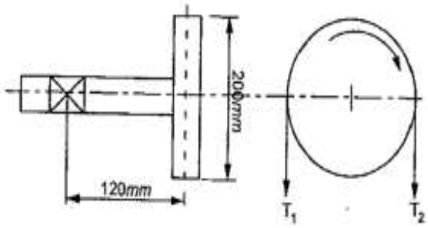
The gear and pulley are keyed to the shaft. The material of the shaft is steel 50C4 (S_{ut} = 700 and S_{yt} = 460 N/mm^2). The factors k_b and $k_t$of the ASME code are 1.5 each. Determine the shaft diameter using the ASME code.(All dimensions are in mm) **(D)**

6. A transmission shaft supporting a helical gear B and an overhang bevel gear D is shown in figure. The shaft is mounted on two bearings A and C. the pitch circle diameter of the helical gear is 450 mm and the diameter of the bevel gear at the forces is 450 mm. power is transmitted from helical gear to the bevel gear. The gears are keyed to the shaft. The material of the shaft is steel 45C8 ($S_{ut} = 600\ and\ S_{yt} = 380\ N/mm^2$). The factors $K_b\ and\ K_t$ of ASME code are 2.0 and 1.5, respectively. Determine the shaft diameter using ASME code. Also find the diameter based on torsional rigidity. Take



permissible angle of twist 3^0 per meter length and modulus of rigidity for the shaft material as 79300 N/mm^2. (All dimensions are in mm) **(D)**

7. A solid shaft is to transmit power from electric motor to a m/c through a vertical belt drive. Pulley has wt. of 250N and overhand of 120mm; Max. Power transmitted is 3kW @150rpm; Coff. of friction between belt and pulley is 0.25; Combined shock and fatigue factor in torsion is 1.5 & in bending 2.0; Permissible stress in shaft is 40MPa. Design the shaft diameter. **(E)**



8. Design a taper sunk key to transmit a power of 60kW @ 300 rpm through a shaft of 100mm dia. Allowable compressive strength is 175MPa. **(E)**
9. A square key of 10x10x75 mm is required to transmit a torque of 11000 N-mm from 60 mm solid shaft. The limiting shear & crushing stress are 60 & 170MPa, respectively. Determine whether the length of key is sufficient or not. **(M)**
10. Classify shaft couplings and also give their relative advantages disadvantages and applications. **(E)**
11. Design a split muff coupling to transmit 50kW power at 120 rpm. The shafts, key and clamping bolts are made of plain carbon steel 30C8. The yield strength in compression is 150% of the tensile yield strength. The factor of safety for key, shaft and bolts is 5. The no. of clamping bolts is 8. The coefficient of friction between sleeve halves and the shaft is 0.3. **(M)**
12. A rigid coupling is used to connect a 37.5kW, 180 rpm electric motor to a centrifugal pump. The service factor for application is 1.5. The shafts and key are made of plain carbon steel 30C8. The flange is made of gray cast iron FG 200. The factor of safety for the shaft and key is 4. For the sleeve, the factor of safety is 5 based on ultimate strength. The bolts are made of steel (S_{yt} = 380N/mm^2) and the factor of safety is 2.5. Design the coupling. Assume that the bolts are finger tight in reamed and ground holes. **(D)**
13. It is required to design a bushed pin type flexible coupling to connect the output shaft of an electric motor to the shaft of a centrifugal pump. The motor delivers 20 kW power at 720 rpm. The starting torque of the motor can be assumed to be 150% of the rated torque. Design the coupling and specify the dimensions of its components. **(D)**

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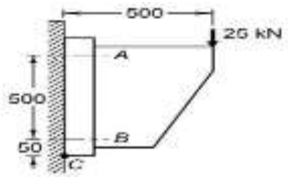
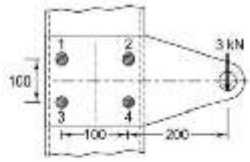
Sheet No.5	**B.Tech. V Semester (2022-23)** **Design of Threaded fasteners, Power screws and curved member**

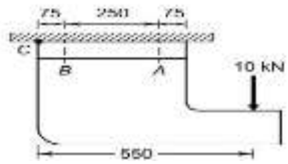
1. The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm^2. It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa. **(M)**



2. A wall bracket is attached to the wall by means of four identical bolts, two at A and two at B, as shown in Fig. Assuming that the bracket is held against the wall and prevented from tipping about the point C by all four bolts and using an allowable tensile stress in the bolts as 35 N/mm^2, determine the size of the bolts on the basis of maximum principal stress theory. **(E)**



3. A steel plate subjected to a force of 3 kN and fixed to a vertical channel by means of four identical bolts is shown in Fig. The bolts are made of plain carbon steel 45C8 (Syt = 380 N/mm^2) and the factor of safety is 2. Determine the diameter of the shank. **(E)**



4. A cast iron bracket, as shown in Fig., supports a load of 10 kN. It is fixed to the horizontal channel by means of four identical bolts, two at A and two at B. The bolts are made of steel 30C8 (Syt = 400 N/mm^2) and the factor of safety is 6. Determine the major diameter of the bolts if (d_c = 0.8d). **(M)**
5. The nominal diameter of a triple-threaded square screw is 50 mm, while the pitch is 8 mm. It is used with a collar having an outer diameter of 100 mm and inner diameter as 65 mm. The coefficient of friction at the thread surface as well as at the collar surface can be taken as 0.15. The screw is used to raise a load of 15 kN. Using the uniform wear theory for collar friction, calculate: (i) torque required to raise the load; (ii) torque required to lower the load; and (iii) the force required to raise the load, if applied at a radius of 500 mm. **(D)**
6. A double-threaded power screw, with ISO metric trapezoidal threads is used to raise a load of 300 kN. The nominal diameter is 100 mm and the pitch is 12 mm. The coefficient of friction at the screw threads is 0.15. Neglecting collar friction, calculate (i) torque required to raise the load; (ii) torque required to lower the load; and (iii) efficiency of the screw. **(D)**
7. A crane hook having an approximate trapezoidal cross-section is shown in Fig. It is made of plain carbon steel 45C8 (Syt = 380 N/mm^2) and the factor of safety is 3.5. Determine the load carrying capacity of the hook. **(E)**
8. The C-frame of a 100 kN capacity press is shown in Fig. The material of the frame is grey cast iron FG 200 and the factor of safety is 3. Determine the dimensions of the frame. **(M)**

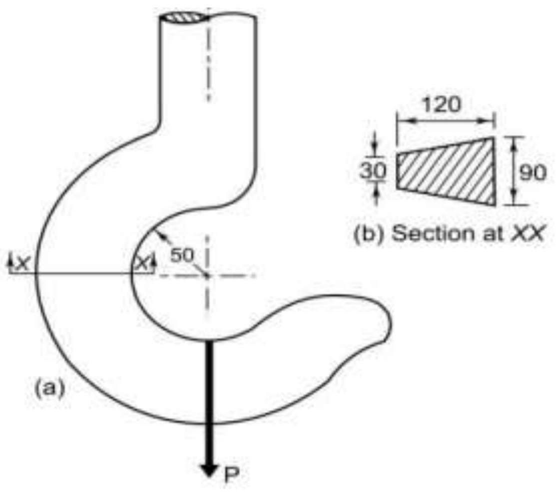

120
30
90
(b) Section at XX
X
X
50
(a)
P

Fig. Q.7

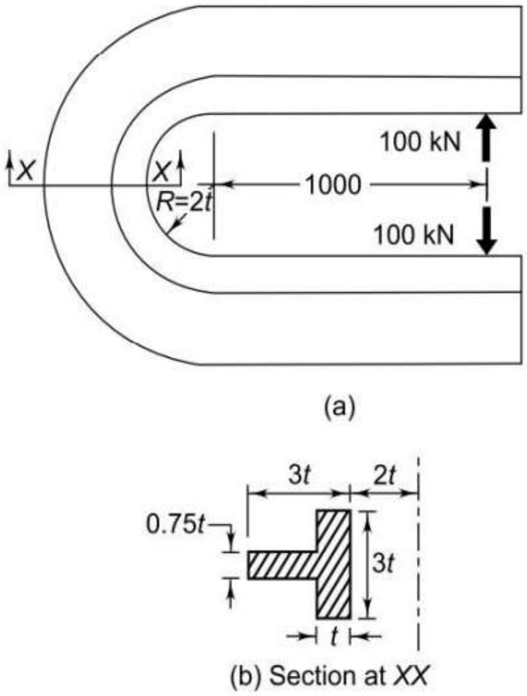
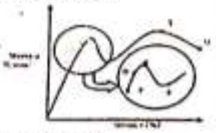

X
X
R=2t
1000
100 kN
100 kN
(a)
3t
2t
0.75t
3t
t
(b) Section at XX

Fig. Q.8

Gate Sheet-1

Q.1 The stress-strain curve for mild steel is shown in the figure.1 given below. Choose the correct option referring to both figure and table. **GATE-ME-2014**



Point on the graph	Description of the point
P	1. Upper Yield Point
Q	2. Ultimate Tensile Strength
R	3. Proportionality Limit
S	4. Elastic Limit
T	5. Lower Yield Point
U	6. Failure

figure.1

(A) P-1, Q-2, R-3, S-4, T-5, U-6 (B) P-3, Q-1, R-4, S-2, T-6, U-5
(C) P-3, Q-4, R-1, S-5, T-2, U-6 (D) P-4, Q-1, R-5, S-2, T-3, U-6

Q.2. A machine element XY, fixed at end X, is subjected to an axial load P, transverse load F, and a twisting moment T at its free end Y. The most critical point from the strength point of view is

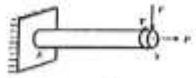


figure.2

GATE-ME-2016

(A) a point on the circumference at location Y (B) a point at the center at location Y
(C) a point on the circumference at location X (D) a point at the center at location X

Q.3. A cantilever beam with flexural rigidity of 200 Nm^2 is loaded a shown in the figure.3. The deflection (in mm) at the tip of the beam is _______. **GATE-ME-2015**

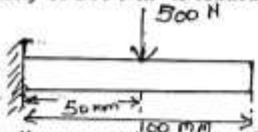


figure-3

Q. 4. Two identical circular rods of same diameter and same length are subjected to same magnitude of axial tensile force. One of the rods is made out of mild steel having the modulus of elasticity of 206 GPa. The other rod is made out of cast iron having the modulus of elasticity 100 GPa. Assume both the materials to be homogeneous and isotropic and the axial forces causes the same amount of uniform stress in both the rods. The stresses developed are with in the proportional limit of the respective materials. Evaluate that the Cast iron rod elongate more or less than the mild steel rod. **GATE-ME-2003**

Q.5. If the Posson's ratio of an elastic material is 0.4, the ratio of modulus of rigidity to Young's modulus is............ **GATE-ME-2014**

Q.6 A bolted joint has four bolts arranged as shown in figure.4. The cross sectional area of each bolt is 25 mm^2. A torque T = 200 N-m is acting on the joint. Neglecting friction due to clamping force, maximum shear stress in a bolt is .... MPa. **GATE-ME-2016**

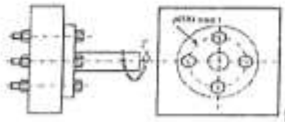
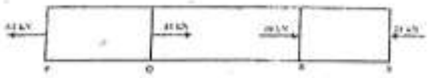


figure.4

Q.7. The state of stress at a point is $\sigma_x = \sigma_y = \sigma_z = \tau_{xy} = \tau_{yz} = 0$ and $\tau_{xy} = \tau_{yx} = 50$ MPa. The maximum normal stress (in MPa) at that point is............ **GATE-ME-2017**

Q.8. A bar in figure.5 having a cross-sectional area of 700 mm^2 is subjected to axial loads at the positions indicated. The value of stress in the segment QR is.................. **GATE-ME-2006**

63
35
98



49
21
28

figure.5

Q.9. Which one of following is NOT correct?
(A) Intermediate principal stress is ignored when applying the maximum principal stress theory
(B) The maximum shear stress theory gives the most accurate results amongst all the failure theories
(C) As per the maximum strain energy theory, failure occurs when the strain energy per unit volume exceeds a critical value
(D) As per the maximum distortion energy theory, failure occurs when the distortion energy per unit volume exceeds a critical value **GATE-ME-2014**

Q 10. A force P is applied at a distance x from the end of the beam as shown in the figure.6. What would be the value of x so that the displacement at 'A' is equal to zero ? **GATE-ME-2014**

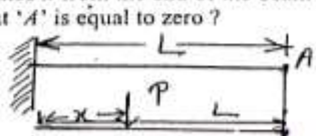


figure.6

Q.11. A steel bar is held by two fixed supports as shown in the figure.7 and is subjected to an increases of temperature ΔT=100°C. and 200GPa, respectively, the magnitude of thermal stress (in MPa) induced in the bar is ___________. **GATE-ME-2017**

figure.7

Q.12. For a loaded cantilever beam of uniform cross section, the bending moment (in N.mm) along the length is $M(x) = 5x^2 + 10x$, where x is the distance (in mm) measured from the free end of the beam. The magnitude of shear force (in N) in the cross-section at x = 10 mm is............... **GATE-ME-2017**

Q.13. The principal stresses at a point in a critical section of a machine component are $\sigma_1 = 60$MPa, $\sigma_2 = 5$MPa and $\sigma_3 = -40$MPa. For the material of the component, the tensile yield strength is $\sigma_y = 200$MPa. According to the maximum shear stress theory, the factor of safety is.......... **GATE-ME-2017**

Q.14. Two circular shafts made od same material, one solid (S) and hollow (H), have the same length and polar moment of inertia. Both are subjected to same torque. Here, θ_s is the twist and τ_s is the maximum shear stress in the solid shaft, whereas θ_H is the twist and τ_H is the maximum shear stress in the hollow shaft. Which one of the following is TRUE? **GATE-ME-2016**

(a) $\theta_S = \theta_H$ and $\tau_s = \tau_H$ (b) $\theta_S > \theta_H$ and $\tau_s > \tau_H$ (c) $\theta_S < \theta_H$ and $\tau_s < \tau_H$ (d) $\theta_S = \theta_H$ and $\tau_s < \tau_H$

Q.15. Consider a steel (Young's Modulus E = 200 GPa) column hinged on both sides. Its height is 1.0 and cross section is 10 mm x 20 mm. The lowest Euler critical buckling load (in N) is................ **GATE-ME-2015**

Q.16. A rod length L having uniform cross-sectional area A is subjected to a tensile strength force P as shown in the figure.8 below. If the Young's modulus of the material varies linearly from E_1 to E_2 along the length of the rod, what will be the normal stress developed at the section SS? **GATE-ME-2013**

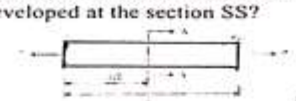


figure.8

Q.17. A long thin walled cylindrical shell, closed at both ends, is subjected to an internal pressure. The ratio of the hoop stress (circumferential stress) to longitudinal stress.................. **GATE-ME-2013**

Q.18. Consider a simply supported beam of length, $50h$, with a rectangular cross-section of depth, h, and width, $2h$. The beam carries a vertical point load, P, at its mid-point. What is the ratio of the maximum shear stress to the maximum bending stress in the beam? **GATE-ME-2014**

Q.19. Two solid circular shafts of radii R_1 and R_2 are subjected to same torque. The maximum shear stresses developed in the two shafts are τ_1 and τ_2. If $R_1/R_2=2$, then τ_2/τ_1 is _______ **GATE-ME-2014**

Q.20. A cylindrical tank with closed ends is filled with compressed air at a pressure of 500 kPa. The inner radius of the tank is 2 m, and it has wall thickness of 10 mm. The magnitude of maximum in-plane shear stress (in MPa) is____________ **GATE-ME-2015**

Q.21. A machine element is subjected to the following bi-axial state of stress: σ_x=80 MPa, σ_y=20 MPa and τ_{xy}=40 MPa. If the shear strength of the material is 100 MPa, the factor of safety as per Tresca's maximum shear stress theory is.................... **GATE-ME-2015**

Q.22. A gas is stored in a cylindrical tank of inner radius 7 m and wall thickness 50 mm. The gage pressure of the gas is 2 MPa. What will be the maximum shear stress (in MPa) developed in the wall? **GATE-ME-2015**

Q.23. The cross-sections of two solid bars made of the same material are shown in the figure.9. The square cross-section has flexural (bending) rigidity I_1, while the circular cross-section has flexural rigidity I_2. Both sections have the same cross-sectional area. The ratio I_1/I_2 is **GATE-ME-2016**

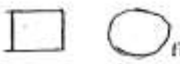


figure.9

Q.24. A shaft with a circular cross-section is subjected to pure twisting moment. What will be the ratio of the maximum shear stress to the largest principal stress? **GATE-ME-2016**

Q.25. A simply-supported beam of length $3L$ is subjected to the loading shown in the figure.10.

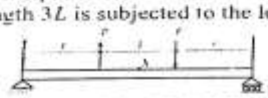


figure.10

It is given that P=1 N, L=1 m and Young's modulus E=200 GPa. The cross-section is a square with dimension 10 mm × 10 mm. The bending stress (in Pa) at the point A located at the top surface of the beam at a distance of 1.5 L from the left end is ____________ **GATE-ME-2016**

Q.26. A cantilever beam of length L and flexural modulus EI is subjected to a point load P at the free end. Find the value of elastic strain energy stored in the beam due to bending (neglecting transverse shear). **GATE-ME-2017**

Q.27. A simply supported beam of width 100 mm, height 200 mm and length 4 m is carrying a uniformly distributed load of intensity 10 kN/m. The maximum bending stress (in MPa) in the beam is __________ **GATE-ME-2018**

Q.28. A carpenter glues a pair of cylindrical wooden logs by bonding their end faces at an angle of θ=30 as shown in the figure.11.

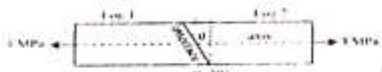


figure.11

The glue used at the interface fails if

Criterion 1: the maximum normal stress exceeds 2.5 MPa.

Criterion 2: the maximum shear stress exceeds 1.5 MPa.

Assume that the interface fails before the logs fail. When a uniform tensile stress of 4 MPa is applied, the interface will fail due to which criterion? **GATE-ME-2018**

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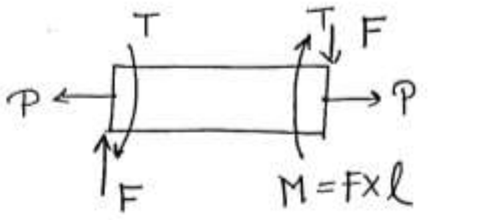
Solution of Gate Sheet-2

1 (c)

2. Solution



Free Body Diagram

At location Y

At circumference

→ Direct stress due to direct load P is $\sigma_0 = \frac{P}{A}$ where A is cross section area

→ Due to shear load F, $\tau_1 = \frac{F}{A}$

→ Due to torsion T is $\tau_2 = \frac{16T}{\pi d^3}$ $\{\because r = \frac{d}{2}\}$

→ Due to B.M F.dx is $\sigma_b = 0$

At centre of location Y

→ $\sigma_0 = \frac{P}{A}$

→ $\tau_1 = \frac{F}{A}$

→ $\tau_2 = 0$ $(\because r = 0)$ in relation $\frac{T}{J} = \frac{\tau}{r}$

→ $\sigma_b = 0$

At location X

At circumference

→ $\sigma_0 = \frac{P}{A}$

→ $\tau_1 = \frac{F}{A}$

→ $\tau_2 = \frac{16T}{\pi d^3}$ $(\because r = d/2)$

→ $\sigma_b = \frac{My}{I} = \frac{32M}{\pi d^3}$ $(\because y = d/2$ & $M = F \times l)$

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②

At centre

→ $\sigma_0 = P/A$

→ $\tau_1 = T/A$

→ $\tau_2 = 0 \; (\because r = 0)$

→ $\sigma_b = 0 \; (\because y = 0)$

∴ The most critical point is at circumference of location X. (C) Ans

Ans. 3 For the given configuration of cantilever loaded beam at mid-point the deflection at the point of load will be affected by load, further it will be deflected by the slope of loaded beam only. So, we have

↳ Deflection at end point = deflection at the point of load + Slope × distance

∴ Deflection at the point of load

$$= \frac{W \cdot a^3}{3EI} = \frac{Wa^3}{3IE}$$

where W = Load = 500 N

a = length = 0.05 m

EI = Flexural rigidity = 200 N·m²

$$\delta_{at\ point\ load} = \frac{500 \times (0.05)^3}{3 \times 200} = 0.10416 \text{ mm}$$

Deflection due to slope

$$= \text{slope} \times \cancel{\text{deflection}} \text{ distance}$$

$$= \frac{Wa^2}{2EI} \times 0.05$$

$$= \frac{500 \times (0.05)^2}{2 \times 200} \times 0.05 = 0.1562 \text{ mm}$$

3

Total deflection at end point is
$= 0.10416 + 0.1562$
$= 0.26041$ mm
So, total deflection is 0.2604 mm.

Ans(4) Given: $L_s = L_i$
$E_s = 206$ GPa,
$E_i = 100$ GPa,
$P_s = P_i$
$D_s = D_i$
$\Rightarrow A_s = A_i$
where subscript s is for steel and i is for iron rod.
We know that elongation is given by

$$\Delta L = \frac{PL}{AE} \quad \text{—(1)} \rightarrow \boxed{\delta L \propto \frac{1}{E}}$$

Now for steel rod

$$\Delta L_s = \frac{P_s L_s}{A_s E_s} \quad \text{—(2)}$$

for iron rod

$$\Delta L_i = \frac{P_i L_i}{A_i L_i} \quad \text{—(3)}$$

now eq 2 divides by eqn (3)

$$\frac{\Delta L_s}{\Delta L_i} = \frac{\cancel{P_s}\cancel{L_s}}{\cancel{A_s}E_s} \times \frac{\cancel{A_i}E_i}{\cancel{P_i}\cancel{L_i}}$$

$$= \frac{E_i}{E_s} \quad \text{—(4)}$$

Substitutes the values, we get

$$\frac{\Delta L_s}{\Delta L_i} = \frac{100}{206} = 0.485 < 1$$

or, $\Delta L_s < \Delta L_i \Rightarrow \Delta L_i > \Delta L_s$
so cast iron rod elongates more than the mild steel rod.

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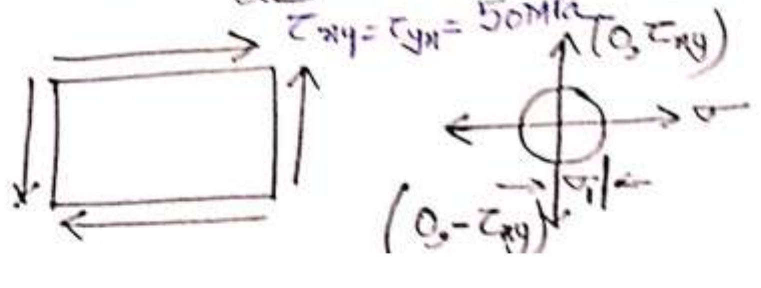
Ans:5 Given that in an elastic material
Poission's ratio of ~~mo~~ (μ) is 0.4.
We know that
$$E = 2G(1+\mu) \quad \text{---} (1)$$
Where E = Young's modulus
G = Modulus of rigidity
So $\frac{G}{E} = \frac{1}{2(1+\mu)}$ --- (2)
or $\frac{G}{E} = \frac{1}{2(1+\mu)} = \frac{1}{2(1+0.4)} = \frac{1}{2.8}$
$= 0.3571$ Ans

Ans:6 Let the resisting force in each bolt
$= F$ Newton
Net resisting torque $(T_R) = 4F \times 01$
$= 4F \times 50 \times 10^{-3}$ N-m
$\because$ Applied torque = Resisting torque (T_R)
$T = 4F \times 50 \times 10^{-3}$
$F = \frac{200 \times 10^{3}}{4 \times 50} = 1000$ N
$\therefore$ Let shear stress developed in each Bolt $= \tau$ MPa
It is given that resisting area (A_R)
$= 25$ mm^2
$F = \tau \times A_R$
$\tau = \frac{1000}{25} = 40$ MPa Ans

Ans:7



It is a pure torsion case in 2D (5

$\tau_{xy} = \tau_{yx} = 50$ MPa

$\therefore$ for pure torsion,

$\tau_{xy} = \tau_{yx} = \sigma_1$

So maximum Normal stress

$\sigma_1 = 50$ MPa

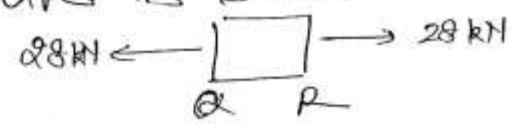
{Max. Principal stress theory}

$$\left\{ \sigma_1 = \frac{1}{2}(\sigma_x + \sigma_y) + \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2} \right.$$

$$\text{or } \sigma_1 = 0 + 0 + \cancel{\tfrac{1}{2}} \times \cancel{2}\tau_{xy}$$

$$\therefore \tau_{xy} = \tau_{yx} = \sigma_1 = 50 \text{ MPa} \}$$

<u>Ans: 8</u> The Force body diagram of segment QR is shown



Given $A = 700\ mm^2$

Force acting on QR, P = 28 kN (Tensile)

Stress in segment QR is given by,

$$\sigma = \frac{P}{Area} = \frac{28 \times 10^3}{700 \times 10^{-6}}$$

$$= 40 \text{ MPa} \quad \underline{\text{Ans}}$$

<u>Ans. 9</u> (A) In maximum principal stress theory, when the value of maximum principal stress is equal to that of yield point stress as found in a simple tensile test in which we don't consider intermediate principal stress.

(B) Maximum shear stress theory is well justified for ductile material because ductile material mostly fails by shear failure.

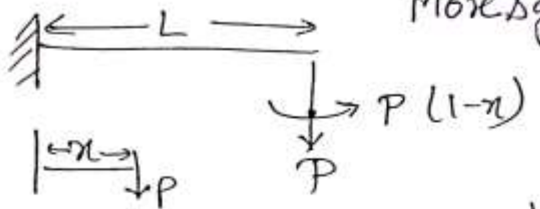
(C) Maximum strain energy over estimate the elastic strength of ductile material. (6)

(D) As per maximum distortion energy theory, failure occurs when the distortion energy per unit volume exceeds a critical value.

Ans is (B)

More safe (Conservative theory)

Ans. 10



where E: Modulus of elasticity
I: area moment of inertia
P: load

for deflection should be zero

$$\frac{P(l-x)\,l^2}{2EI} = \frac{P.l^3}{3EI}$$

$$\frac{(l-x)}{2} = \frac{l}{3}$$

or $3l - 3x - 2l = 0$

$l = 3x$

$x = l/3$ Ans

Ans 11. Thermal stress,

$\sigma_{th} = E\alpha\Delta T = 200\times10^3 \times 11\times10^{-6} \times 100$
$= 220\,MPa$ Ans

Ans. 12 $S.F = \frac{d}{dx}.(B.M)$

$S.F = \frac{d}{dx}(5x^2+10x)$

$S.F. = 10x+10 \Rightarrow [S.F.]_{x=10mm} = (10\times10)+10 = 110N$ Ans

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Ans. 13 According to the maximum shear stress theory.

$$\frac{\sigma_1 - \sigma_3}{2} \leq \frac{\sigma_{yield}}{2}$$

$$\Rightarrow \sigma_1 - \sigma_3 = \frac{\sigma_y}{FOS}$$

or $60 - (-40) = \frac{200}{FOS}$

or $FOS = 2$ Ans

Ans. 14 According to pure torsion equation

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{l}$$

Let d_s and d_h are diameter of solid shaft and outer diameter of hollow shaft

$$\frac{\frac{T_s}{J_s} = \frac{\tau_s}{R_s}}{\frac{T_h}{J_h} = \frac{\tau_h}{R_h}} \Rightarrow \frac{\tau_s}{R_s} = \frac{\tau_h}{R_h}$$

$$\Rightarrow \frac{\tau_s}{d_s} = \frac{\tau_h}{d_h} \Rightarrow \tau_s = \tau_h \cdot \frac{d_s}{d_h}$$

Since $J_s = J_h$

∴ d_h must be greater than d_s

∴ $\frac{d_s}{d_h}$ must be less than 1 $\Rightarrow \boxed{\tau_s < \tau_h}$ and Ans

$$\frac{T}{J} = \frac{G\theta}{l} \Rightarrow \frac{\frac{T_s}{J_s} = \frac{G_s\theta_s}{l_s}}{\frac{T_h}{J_h} = \frac{G_h\theta_h}{l_h}}$$

Ans (d)

$$\Rightarrow \boxed{\theta_s = \theta_h}$$ Ans

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Ans: 15 Euler critical buckling load : (8)

A long, slender column becomes unstable when its axial compressive load reaches a value called the critical buckling load.

- At this value, the structure is in equilibrium regardless of the magnitude of the angle.

It is given by $P_{cr} = \frac{\pi^2 E I}{l_c^2}$

Where P_{cr} = Euler's critical buckling load

E = Young's Modulus ($E = 200$ GPa)

I = Moment of Inertia

l_c = Equivalent length of column (let $L_e = l$ for both hinged side)

Note that the least moment of inertia (I) should be taken in order to find lowest critical stress.

Cross section of beam = $(10 \times 20)\ mm^2$

Least moment of inertia

$$= \frac{bh^3}{12} = \frac{20 \times (10)^3}{12}\ mm^4 = I_{min}$$

$$P_{(cr)min} = \frac{\pi^2 \times 200 \times 10^9}{(1)^2} \frac{\times 20 \times 10^3}{12} \times (10^{-3})^4$$

$$= 3289.868\ N$$

So lowest Euler critical buckling load of given beam is 3289.868 N Ans

Ans.16 The normal stress is given by (9

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

same tensile load P at all the cross sections

We see that normal stress only depends upon on force and area and it does not depends on E.

Ans.17 Hoop stress or circumferential stress is

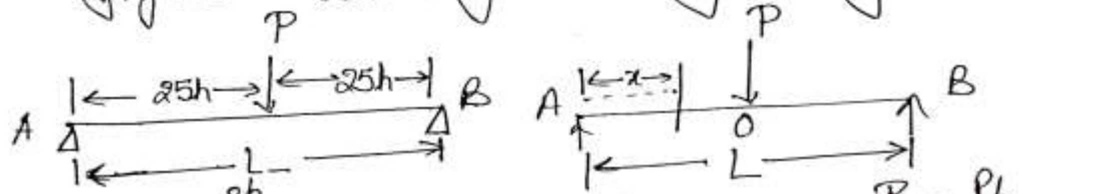
$$\sigma_1 = \frac{p.r}{t}$$

and longitudinal or axial stress is

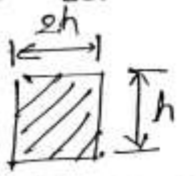
$$\sigma_2 = \frac{p.r}{2t}$$

$$\text{Ratio } \frac{\sigma_1}{\sigma_2} = \frac{pr}{t} \times \frac{2t}{pr} = 2 \text{ Ans}$$

Ans.18 A simply supported beam is shown in figure with free body diagram



$R_1 = P/2$ $\quad R_2 = P/2$



$$\boxed{\tau_{max} = 1.5\, \tau_{avg}} = 1.5 \times \frac{Load}{Area}$$

At a distance x from 'A' shear force

$$= \frac{P - P/2}{A}$$

Maximum shear stress

$$= \frac{P/2}{A}$$

Bending moment $= (P - P/2) \cdot x$

Maximum bending moment will be at $x = l/2$ as shear force direction will change from that point.

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where M – Max bending moment
I : Area moment of inertia of beam
y : max distance from mean line of beam

Maximum bending moment $= \left(\frac{P}{2}\right)\left(\frac{L}{2}\right) = \frac{P.L}{4}$

Maximum bending stress $(\sigma) = \frac{M.y}{I}$

$$= \frac{(PL/4)(h/2)}{\left[\frac{(2h)(h)^3}{12}\right]}$$

$$= \frac{3}{4}\frac{P.L}{h^3}$$

Ratio of maximum shear stress (τ) to maximum bending stress is

$$\frac{\tau}{\sigma} = \frac{(P/2)/A}{3PL/4h^3} \times 1.5$$

$$= \frac{(P/2)/2h^2}{3PL/4h^3} = \frac{h}{3L}$$

$$L = 50h$$

$$\frac{\tau}{\sigma} = \frac{h}{3 \times 50h}$$

$$= \frac{h}{150h} = \frac{1}{150} = 0.0066 \times 1.5$$

$$= 0.01 \text{ Ay}$$

Ans:19 We know that for solid circular shaft max shear stress developed can be expressed as

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

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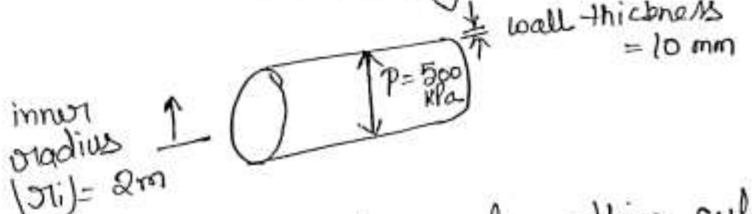
∴ for two shaft of radii R_1 and R_2 ratio of (11)
Shear stress for same torque will be.

$$\frac{\tau_2}{\tau_1} = \frac{16T/\pi d_2^3}{16T/\pi d_1^3}$$

$$\text{or } \frac{\tau_2}{\tau_1} = \frac{d_1^3}{d_2^3} = \left(\frac{R_1}{R_2}\right)^3$$

$$\text{or } \frac{\tau_2}{\tau_1} = (2)^3 = \underline{8 \text{ Ans}}$$

Ans.20: For a thin cylinder



longitudinal stress for thin cylinder is given by $(\sigma_L) = \frac{P r_i}{2t} = \frac{p d_i}{4t}$

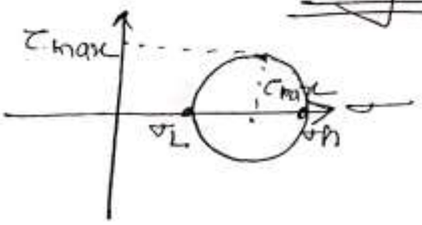
$$\frac{500 \times 10^3 \times 2}{2 \times 10 \times 10^{-3}} = 50 \text{ MPa}$$

Hoop stress for thin cylinder is given by

$$(\sigma_h) = \frac{\sigma . r_i}{t} = \frac{p r_i}{t} = \frac{p . d_i}{2t}$$

$$= \frac{500 \times 10^3 \times 2}{10 \times 10^{-3}} = 100 \text{ MPa}$$

Maximum in-plane shear stress is given by $\left(\frac{\sigma_1 - \sigma_2}{2}\right)$ as shown in Mohr's circle diagram.



$$\tau_{max} = \frac{\sigma_h - \sigma_L}{2}$$

$$\tau_{max} = \frac{100 - 50}{2} = 25 \text{ MPa}$$

So maximum in plane - shear stress in given cylinder is 25 MPa

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Ans. 21 For a machine element subjected to a bi-axial state of stress (12)

$$\sigma_x = 80\,MPa$$
$$\sigma_y = 20\,MPa$$
$$\tau_{xy} = 40\,MPa$$

maximum shear stress will be given by

$$\tau_{max} = \pm\sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2+\tau_{xy}^2} = \frac{\sigma_1-\sigma_2}{2}$$

$$\tau_{max} = \pm\sqrt{\left(\frac{80-20}{2}\right)^2+(40)^2}$$

$$\tau_{max} = \pm 50\,MPa$$

$$\text{Factor of safety} = \frac{\text{Shear strength of Material}}{\text{Maximum Shear Stress}}$$

$$= \frac{100}{50} = 2$$

So factor of safety according to Tresca's maximum shear stress theory is 2.

Ans 22. Maximum shear stress in the wall in form of longitudinal stress

$$\tau_{max} = \frac{\sigma_l-\sigma_c}{2} \qquad \sigma_x = \tau_{max} = \frac{P \cdot r_i}{2t}$$

$$= \frac{2\times 7}{2\times 50\times 10^{-3}} = 140\,MPa$$

Ans. 23. Flexural rigidity $= E \cdot I$

Both have same cross-section area

$$\Rightarrow a^2 = \frac{\pi}{4}d^2$$

where a is side of square and d is diameter of circle

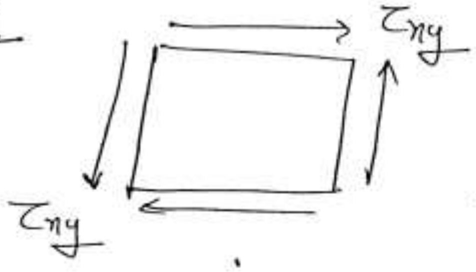
(15)

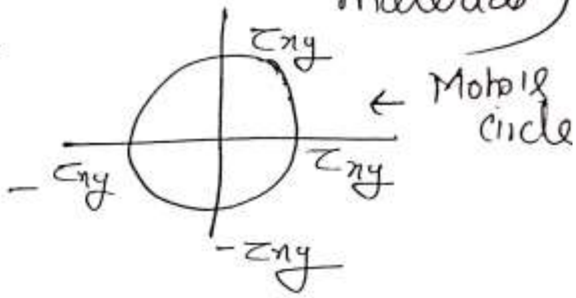
$$\Rightarrow a^4 = \frac{\pi^2}{16} d^4 \Rightarrow \frac{I_1}{I_2} = \frac{E_1 \times \frac{a^4}{12}}{E_2 \times \frac{\pi d^4}{64}}$$

$$\frac{E_1 I_1}{E_2 I_2} = \frac{I_1}{I_2}$$ Flexural rigidity

$$\Rightarrow \frac{\frac{\pi^2 d^4}{12 \times 16}}{\frac{\pi}{64} d^4} = \frac{\pi}{3}$$ ($E_1 = E_2$ because Ae of same material)

Ans: 24



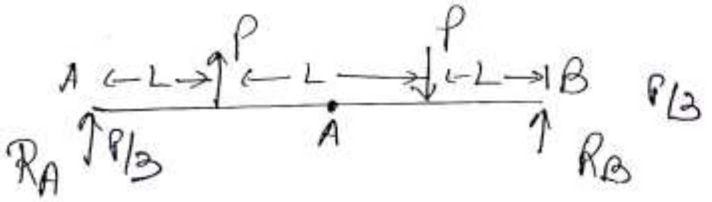


$\because \tau_{max} = \tau_{xy}$

$\because \sigma_1 = \tau_{xy}$

$\therefore \boxed{\frac{\tau_{max}}{\sigma_1} = 1}$ Ans.

Ans: 25



Taking moment about B

$$\Sigma M_B = 0 \Rightarrow R_A \times 3L + P \times 2L - PL = 0$$

$$\Rightarrow R_A = -P/3$$

$$\Sigma F_y = 0 \Rightarrow R_B + R_A = 0 \Rightarrow R_B = P/3$$

Taking moment about A

$$\Sigma M_A = 0 \quad -\frac{P}{2} \times 1.5L + P \times 0.5L = 0$$

(14)

$\Sigma M_A = 0$

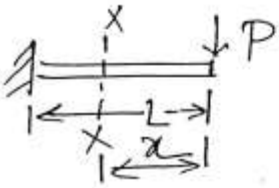
$R_A \times 1.5L + 0.5PL = M_A'$ (Assuming M_A' anticlockwise)

$\Rightarrow -\frac{P}{3} \times 1.5L + 0.5PL = M_A'$

$\Rightarrow M_A' = 0$

$\therefore \frac{M}{I} = \frac{\sigma_b}{y} \Rightarrow \sigma_b$ (Bending stress) $= 0$ Since $\boxed{M_A' = 0}$

Ans 26:



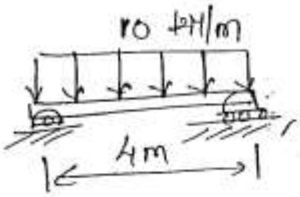
$M_x = -P.x$

$\therefore U = \int_0^L \frac{M_x^2 dx}{2EI} = \int_0^L \frac{P^2 x^2 dx}{2EI}$

$\frac{P^2}{2EI} \times \frac{x^3}{3} \Big|_0^L$

$\boxed{U = \frac{P^2 L^3}{6EI}}$ Ans

Ans: 27:



$\sigma = \frac{M}{I} y,$

$M = \frac{wL^2}{8} = \frac{10 \times 4^2}{8} = 20$ kN-m

$I = \frac{bd^3}{12} = \frac{100 \times 200^3}{12}$

$= 66.67 \times 10^6$ mm^4

$y = \frac{200}{2} = 100;$

$\sigma = \frac{20 \times 10^6}{66.67 \times 10^6} \times \left(\frac{200}{2}\right)$

9828866622

Ans: 28 $\theta = 30°$

$\Rightarrow \sigma = \frac{P}{A} = 4$ MPa

$\sigma_\theta = \frac{P}{A} \cos^2\theta$

$= 4\cos^2 30$

$= 3$ MPa > 2.5 MPa

$\therefore \tau_\theta = \frac{P}{2A} \sin 2\theta$

$\frac{\pi}{32} d^4$, $\frac{3.14}{32}$, $\frac{3.14}{64}$

$\Rightarrow \tau_\theta = \frac{4}{2} \sin 60°$

$= 1.732 > 1.5$ MPa

Fails by both criterion 1 and 2.

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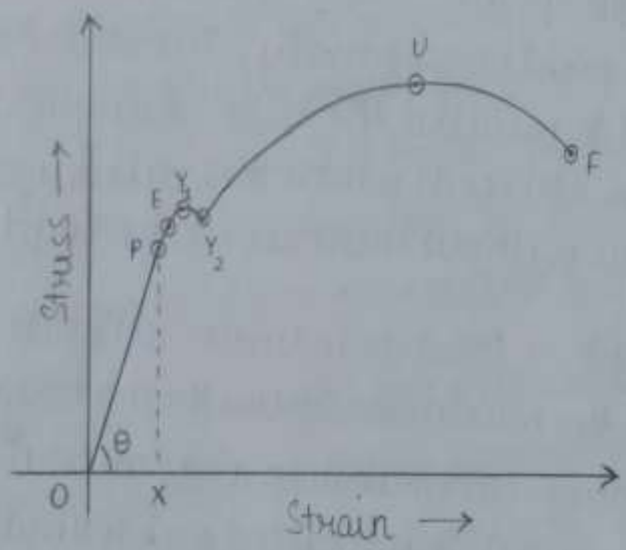
Solution of Sheet-1

Q.1

Ans.



Stress-Strain diagram of a Ductile Material (mild steel):

(i) Proportional Limit → It is observed from the diagram that stress-strain relationship is linear from the point O to P. OP is a straight line and after the point P, the curve begins to deviate from the straight line. Hooke's law states that stress is directly proportional to strain, Therefore, it is applicable only upto the point P. The term proportional limit is defined as the stress at which the stress-strain curve begins to deviate from the straight line. Point 'P' indicates the proportional limit.

(ii) Elastic Limit → It is the limiting value of the stress upto which if the material is stressed and released (unloaded), the strain disappears completely, and the original length is regained. This point is slightly beyond the limit of proportionality. Point 'E' indicates elastic limit on the graph.

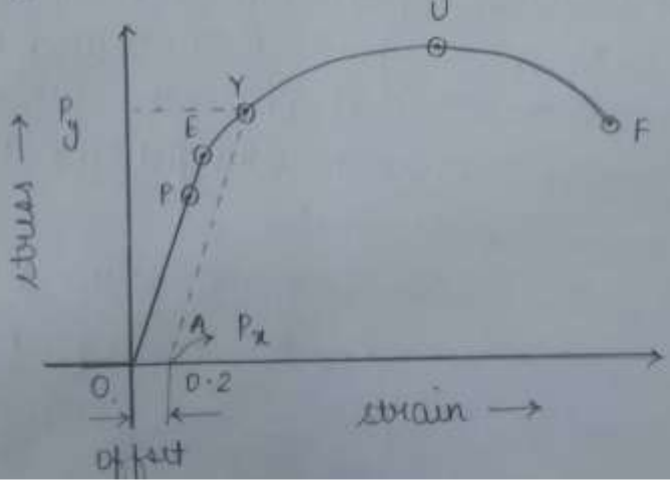
(iii) Yield point → When the specimen is stressed beyond the point 'E', plastic deformation occurs and the material starts yielding. During this stage, it is not possible to recover the initial size and shape of the specimen, on the removal of load. In the graph, points Y_1 and Y_2, are called the upper and lower yield points respectively. The stress corresponding to the yield point Y_1 is called the Yield strength. It is defined as the maximum stress at which a marked increase in elongation occurs without increase in the load.

(iv) Ultimate Strength → Point 'U' indicates ultimate strength on the graph. This is the maximum stress the material can resist. At this stage, the cross-sectional area at a particular section starts reducing very fast. This is called neck formation. After this stage, the resisted load, and hence the developed stress, starts reducing.

(v) Fracture Point → Point 'F' indicates the fracture point on the graph. The value of stress at which, Breaking point in the graph. The value of stress at which, the specimen finally breaks, is called the Fracture point.

Q.2

Ans.



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Proof strength :→ Many varieties of steel, especially heat-treated steels and cold-drawn steels, do not have a well-defined Yield point. For such materials, the yield strength is defined as the stress corresponding to a permanent set of 0.2% of gauge length. In such cases, the yield strength is determined by offset method.

A distance OA equal to 0.002 mm/mm strain (corresponding to 0.2% gauge length) is marked on the x-axis. A line is constructed from A, parallel to OP. The point of intersection of this line with the curve is called Yield strength point, and the corresponding stress is called 0.2% Yield strength.

Proof load or Proof strength are frequently used terms in the design of fasteners. Proof strength is similar to Yield strength. It is also determined by the offset method; however the offset in this case is 0.001 mm/mm, corresponding to a 0.1% of gauge length. 0.1% Proof strength ($R_p0.1$), is defined as the stress which will produce a permanent extension of 0.1% in the gauge length of the test specimen.

Q.3
Ans. Mechanical Properties of Engineering Materials :

(i) Strength → It is defined as the ability of the material to resist, without rupture, external forces causing various types of stresses. Strength is measured by different quantities.

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(iii) Elasticity → Elasticity is defined as the ability of the material to regain its original shape and size after the deformation, when the external forces are removed. All the engineering materials are elastic, but the degree of elasticity varies.

(iv) Plasticity → It is defined as the ability of the material to retain the deformation produced under the load on a permanent basis. In this case, the external forces deform the metal to such an extent, that it cannot fully recover its original dimensions.

(iv) Stiffness → Stiffness or rigidity is defined as the ability of the material to resist deformation under the action of an external load. Modulus of elasticity is the measure of stiffness. The values of the modulus of elasticity for aluminium alloy and carbon steel are 71000 and 207000 N/mm^2 respectively. Therefore, carbon steel is stiffer than aluminium alloy.

(v) Resilience → Resilience is defined as the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded. Resilience is measured by a quantity called Modulus of Resilience, which is the strain energy per unit volume that is required to stress the specimen in a tension test to the elastic limit point. It is represented by the area under the stress-strain curve from the origin to the elastic limit point.

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) Toughness → It is defined as the ability of the material to absorb energy before fracture takes place. This property is essential for machine components which are required to withstand impact loads. Toughness is measured by a quantity called Modulus of Toughness. It is the total area under stress-strain curve in a tension test, which also represents the work done to fracture the specimen.

(vii) Malleability → It is defined as the ability of a material to deform to a greater extent before the sign of crack, when it is subjected to compressive force. Malleable metals can be rolled, forged or extruded because these processes involve shaping under compressive force.

(viii) Ductility → Ductility is defined as the ability of a material to deform to a greater extent before the sign of crack, when it is subjected to tensile force. Ductile metals can be formed, drawn or bent because these processes involve shaping under tension.

(ix) Brittleness → It is the property of a material which shows negligible plastic deformation before fracture takes place. Brittleness is the opposite to ductility.

(x) Hardness → It is defined as the resistance of the material to penetration or permanent deformation. It usually indicates resistance to abrasion, scratching, cutting or shaping. Hardness is an important property in the selection of material for pinion, gear, cam and follower, rail parts, etc.

Q.4

Ans. Given: $S_{yt} = 380\ N/mm^2$

$\sigma_x = 100\ MPa$

$\sigma_y = 40\ MPa$

$\tau_{xy} = 80\ MPa$

(i) Maximum normal stress theory:

$$\sigma_1 = \left(\frac{\sigma_x + \sigma_y}{2}\right) + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \left(\frac{100+40}{2}\right) + \sqrt{\left(\frac{100-40}{2}\right)^2 + (80)^2}$$

$$\sigma_1 = 155.44$$

$$\therefore F.O.S = \frac{S_{yt}}{\sigma_1} = \frac{380}{155.44}$$

$$\boxed{FOS = 2.44}$$

(ii) Maximum shear stress theory:

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2}$$

$$\sigma_1 = \left(\frac{\sigma_x + \sigma_y}{2}\right) + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 155.44\ MPa$$

$$\sigma_2 = \left(\frac{\sigma_x + \sigma_y}{2}\right) - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = -15.44\ MPa$$

$$\tau_{max} = \frac{155.44 - (-15.44)}{2} = 85.44$$

Now, $F.O.S = \frac{\sigma_{yt}}{\sigma_t}$ $\Rightarrow \frac{\sigma_t}{2} \le \tau_{max}$

$\sigma_t \le (\sigma_1 - \sigma_2)$

$\therefore$ $FOS = \frac{\sigma_{yt}}{\sigma_1 - \sigma_2} = \frac{380}{170.88} = 4.44$

$$\boxed{FOS = 4.44}$$

(iii) The distortion energy theory :

$\sigma_t = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2}$

$\sigma_t = \sqrt{(155.44)^2 + (-15.44)^2 - (155.44)(-15.44)}$

$\sigma_t = 163.70$ MPa

$\therefore$ $FOS = \frac{\sigma_{yt}}{\sigma_t} = \frac{380}{163.70}$

$$\boxed{FOS = 2.32}$$

Q.5

Ans. Elongation due to its own weight

$\delta_1 = \frac{PL}{AE}$

where, $P = W$

$P = \frac{7850 \times 9.81 \times 300 \times 150 \times 1000}{1000^3}$

$P = 3465.3825$ N

$L = 75 \times 1000 = 75000$ mm

$A = 300$ mm^2

$E = 200000$ MPa

$$\therefore \delta_1 = \frac{3465.3825 \times 75000}{300(200000)}$$

$$\delta_1 = 4.33\,mm$$

Elongation due to applied load :

$$\delta_2 = \frac{PL}{AE}$$

Here, $P = 20KN = 20000\,N$

$L = 150\,m = 150 \times 10^3\,mm$

$A = 300\,mm^2$

$E = 2 \times 10^5\,MPa$

$$\therefore \delta_2 = \frac{2 \times 10^4 \times 150 \times 10^3}{300 \times 2 \times 10^5}$$

$$\delta_2 = 50\,mm$$

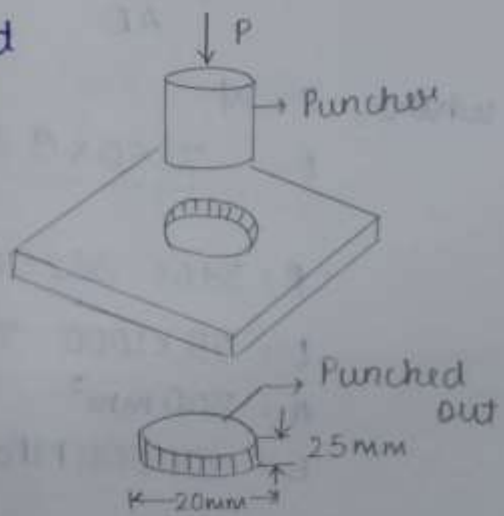
so, Total Elongation : $\delta = \delta_1 + \delta_2$

$$\delta = 4.33 + 50$$

$$\boxed{\delta = 54.33\,mm}$$

Q.6

Ans. The resisting area is the shaded area along the perimeter and the shear force 'V' is equal to the punching force 'P'



$$V = \tau A$$

$$V = P = 350[\pi(20)(25)]$$

$$P = 549778.7\,N$$

$$\boxed{P = 549.778\,KN}$$

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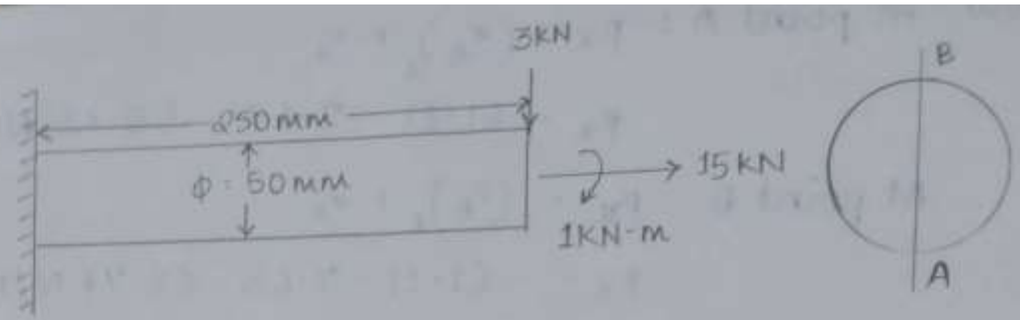
Q.7
Ans.



Given: diameter : d = 50mm

Torque : $T = 1kN\text{-}m = 1000\,N\text{-}m = 10^6\,N\text{-}mm$

Axial Thrust : $P = 15kN = 15\times10^3\,N$

Bending Load : $W = 3kN = 3\times10^3\,N$

Length of Bar : $l = 250\,mm$

we know that, Axial stress : $\sigma_A = \frac{4P}{\pi d^2}$

$$\sigma_A = \frac{4\times15\times10^3}{\pi\times50^2} = 7.63\,N/mm^2 \text{ (Compressive)}$$

Now, Bending Moment : $M = W\times l = 3\times10^3\times250$

$M = 75\times10^4\,N\text{-}mm$

Bending stress : $\sigma_B = \frac{M\cdot y}{I} = \frac{75\times10^4\times d/2}{\frac{\pi}{64}\times d^4}$

$\therefore$ at point A : $(\sigma_B)_A = 61.11\,N/mm^2$ (Tensile)

at point B : $(\sigma_B)_B = -61.11\,N/mm^2$ (Compressive)

Now, shear stress : $\tau = \frac{TR}{J} = \frac{T\times d/2}{\frac{\pi}{32}\times d^4} = \frac{16T}{\pi d^3}$

$$\tau = 40.74\,N/mm^2$$

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Now, At point A : $p_x = (\sigma_B)_A + \sigma_A$

$$p_x = 61.11 - 7.63 = 53.48 \text{ N/mm}^2$$

At point B : $p_x = (\sigma_B)_B + \sigma_A$

$$p_x = -61.11 - 7.63 = -68.74 \text{ N/mm}^2$$

and $p_y = 0$ at 'A' as well as 'B'.

Here, we are considering critical points as 'A' and 'B'. critical points will be the points where torsional shear is maximum, bending is maximum, all the points lying on the periphery will be critical.

At point A : (i) Maximum Normal Stress :

$$\sigma_1 = \frac{p_x + p_y}{2} + \sqrt{\left(\frac{p_x - p_y}{2}\right)^2 + \tau^2}$$

$$\sigma_1 = \frac{53.48}{2} + \sqrt{\left(\frac{53.48}{2}\right)^2 + (40.74)^2}$$

$$\boxed{\sigma_1 = 75.47 \text{ N/mm}^2}$$

(ii) Minimum normal stress :

$$\sigma_2 = \frac{p_x + p_y}{2} - \sqrt{\left(\frac{p_x - p_y}{2}\right)^2 + \tau^2}$$

$$\sigma_2 = \frac{53.48}{2} - \sqrt{\left(\frac{53.48}{2}\right)^2 + (40.74)^2}$$

$$\boxed{\sigma_2 = -21.99 \text{ N/mm}^2}$$

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Maximum Shear Stress :

$$\tau_{max} = \sqrt{\left(\frac{P_x - P_y}{2}\right)^2 + \tau^2}$$

$$\tau_{max} = \sqrt{\left(\frac{53.48}{2}\right)^2 + (40.74)^2}$$

$$\boxed{\tau_{max} = 48.73\ N/mm^2}$$

At point B : (i) Maximum Normal Stress :

$$\sigma_1 = \frac{P_x + P_y}{2} + \sqrt{\left(\frac{P_x - P_y}{2}\right)^2 + \tau^2}$$

$$\sigma_1 = -\frac{68.74}{2} + \sqrt{\left(-\frac{68.74}{2}\right)^2 + (40.74)^2}$$

$$\boxed{\sigma_1 = 18.93\ N/mm^2}$$

(ii) Minimum Normal Stress :

$$\sigma_2 = \frac{P_x + P_y}{2} - \sqrt{\left(\frac{P_x - P_y}{2}\right)^2 + \tau^2}$$

$$\sigma_2 = -\frac{68.74}{2} - \sqrt{\left(-\frac{68.74}{2}\right)^2 + (40.74)^2}$$

$$\boxed{\sigma_2 = -87.67\ N/mm^2}$$

(iii) Maximum shear stress :

$$\tau_{max} = \sqrt{\left(\frac{P_x - P_y}{2}\right)^2 + \tau^2}$$

$$\tau_{max} = \sqrt{\left(-\frac{68.74}{2}\right)^2 + (40.74)^2}$$

$$\boxed{\tau_{max} = 53.30\ N/mm^2}$$

Q.8

= Ans. Given : Power : $P = 20kW = 20\times10^3 W$

Frequency : $f = 2Hz$

Shear stress : $\tau = 40MPa = 40 N/mm^2$

we know that : Torque : $T = \frac{P}{2\pi f}$

$$T = \frac{20\times10^3}{2\pi\times2} = 1591.54 \text{ N-m}$$

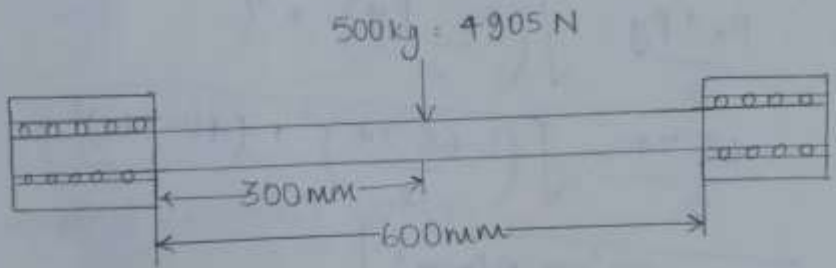
also, by Torsion formula : $\tau = \frac{16T}{\pi d^3}$

$$d^3 = \frac{16\times1591.54}{\pi\times40\times10^6}$$

$$\boxed{\therefore d = 0.05873 m = 58.73 mm}$$

Q.9

= Ans.



Moment : $M = \frac{WL}{4} = \frac{4905\times600}{4}$

$$M = 7.5\times10^5 \text{ N-mm}$$

Also : $P = \frac{2\pi NT}{60} = \frac{2\pi\times300\times T}{60} = 30\times10^6$

$$\therefore T = \frac{30\times10^6\times60}{2\pi\times300} = 9.549\times10^5 \text{ N-mm}$$

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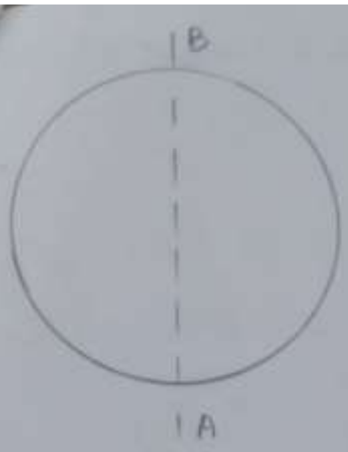


Maximum bending and shearing stress occur at A and B along vertical diameter.

Now, Bending Stress : $\sigma_B = \frac{M \cdot y}{I}$

$$\sigma_B = \frac{M \cdot d/2}{\frac{\pi}{64} \cdot d^4} = \frac{32M}{\pi d^3}$$

At point A : $p_x = \sigma_B = \frac{32M}{\pi d^3}$; $p_y = 0$

shearing stress : $\tau = \frac{TR}{J} = \frac{T \times d/2}{\frac{\pi}{32} \cdot d^4} = \frac{16T}{\pi d^3}$

Maximum Normal stress :

$$p_1 = \frac{p_x + p_y}{2} + \sqrt{\left(\frac{p_x - p_y}{2}\right)^2 + \tau^2}$$

$$p_1 = \frac{16M}{\pi d^3} + \sqrt{\left(\frac{32M}{2\pi d^3}\right)^2 + \left(\frac{16T}{\pi d^3}\right)^2}$$

$$p_1 = \frac{16}{\pi d^3}\left\{M + \sqrt{M^2 + T^2}\right\}$$

$$p_1 = \frac{16}{\pi \times 75^3}\left\{7.5 \times 10^5 + \sqrt{(7.5 \times 10^5)^2 + (9.5 \times 10^5)^2}\right\}$$

$$p_1 = \frac{16 \times 10^5}{\pi \times 75^3}(7.5 + 12.10)$$

$$\boxed{p_1 = 23.66 \text{ N/mm}^2}$$

Minimum Normal Stress :

$$P_2 = \frac{P_x + P_y}{2} - \sqrt{\left(\frac{P_x - P_y}{2}\right)^2 + \tau^2}$$

$$P_2 = \frac{16 \times 10^5}{\pi \times 75^3}(7.5 - 12.10)$$

$$\boxed{P_2 = -5.55 \text{ N/mm}^2}$$

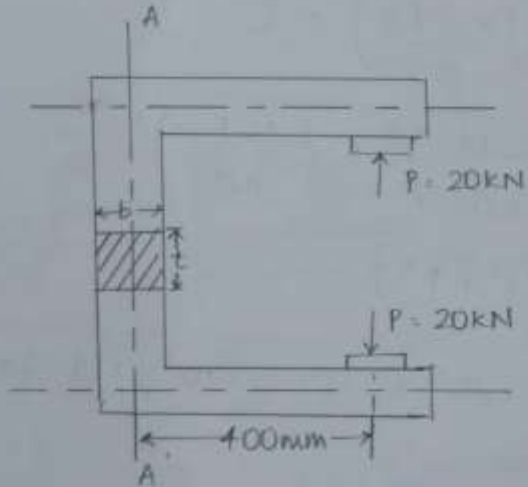
Maximum Shear Stress :

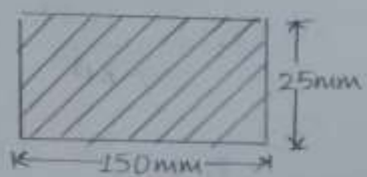
$$\tau_{max} = \sqrt{\left(\frac{P_x - P_y}{2}\right)^2 + \tau^2}$$

$$\tau_{max} = \frac{16 \times 10^5}{\pi \times 75^3} \times 12.10$$

$$\boxed{\tau_{max} = 14.60 \text{ N/mm}^2}$$

Q.16
Ans.





Given: $P = 20\text{kN} = 20 \times 10^3 \text{ N}$

$b = 150 \text{ mm}$

$t = 25 \text{ mm}$

$\text{Area} = b \times t = 3750 \text{ mm}^2$

Direct stresses acting on section A-A':

$$\sigma_D = \frac{P}{A} = \frac{20 \times 10^3}{3750}$$

$$\sigma_D = 5.34 \text{ N/mm}^2$$

Bending stress on section A-A':

$$\sigma_B = \frac{M}{Z}$$

Here, M = Bending moment (N-mm)

$$M = P \times l = 20 \times 10^3 \times 400$$

$$M = 8 \times 10^6 \text{ N-mm}$$

also, Z = Section modulus $(mm^3) = \frac{I}{y}$

$$Z = \frac{tb^3}{12 \cdot b/2} = \frac{tb^2}{6} = 93750 \text{ mm}^3$$

$$\therefore \sigma_B = \frac{8 \times 10^6}{93750} = 85.34 \text{ N/mm}^2$$

so, total stress on A-A' section:

$$\sigma_t = \sigma_D + \sigma_B$$

$$\boxed{\sigma_t = 90.68 \text{ N/mm}^2}$$

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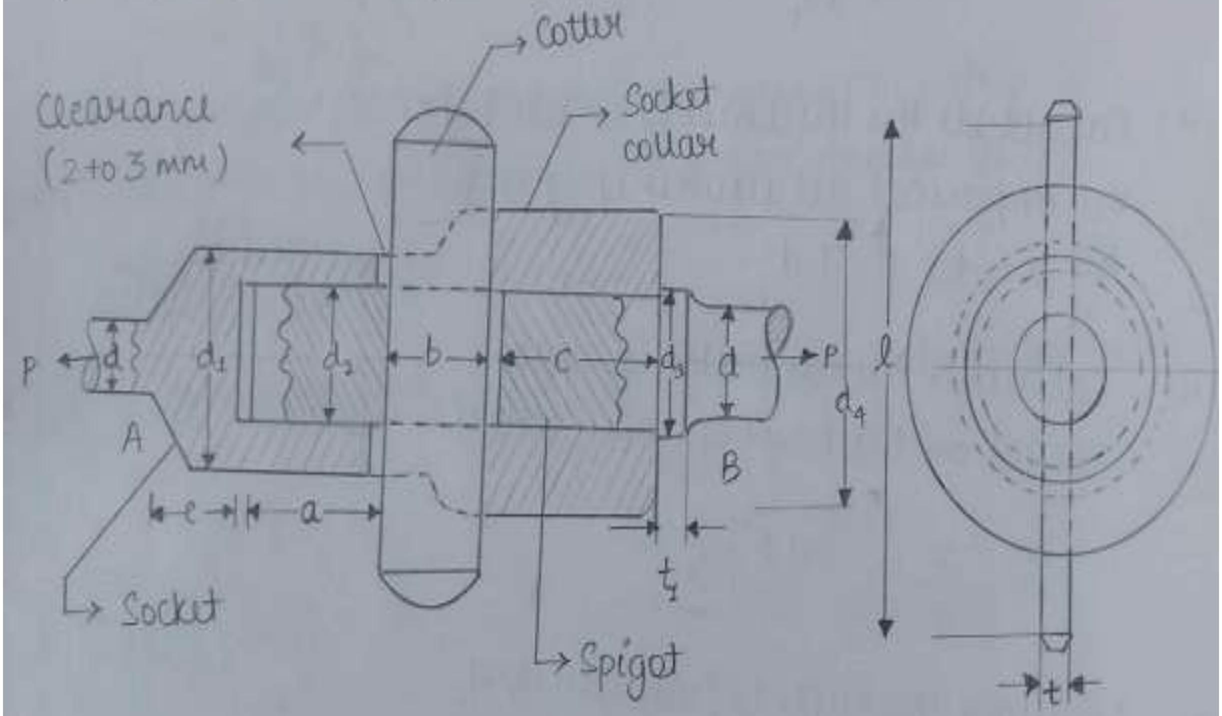
Solution of Sheet-2

Q.1

Ans. The different types of cotter joints are as follows:

(i) Socket and Spigot cotter
(ii) Sleeve and Cotter joints
(iii) Gib and Cotter joints.



DESIGN OF SPIGOT AND SOCKET COLLAR

Here, P = Load carried by the rods
d = diameter of the rods
d_1 = outside diameter of socket
d_2 = diameter of spigot or inside diameter of socket
d_3 = outside diameter of spigot collar

t_1 = thickness of spigot collar
d_4 = diameter of socket collar
c = thickness of socket collar
b = Mean width of cotter
t = thickness of cotter
l = length of cotter
a = distance from the end of the slot to the end of the rod

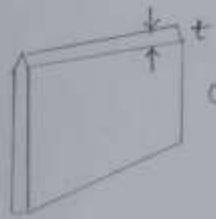
Q.2
Ans. Design procedure for spigot socket cotter joint :

(i) Calculate the diameter of each rod

$$d = \sqrt{\frac{4P}{\pi \sigma_t}}$$


Rod

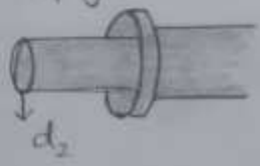
(ii) Calculate the thickness of cotter by the empirical relationship given by : $t = 0.31\, d$



cotter

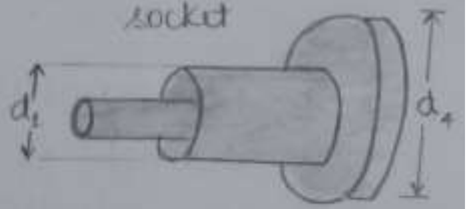
(iii) Calculate the diameter d_2 of the spigot on the basis of tensile stress :

$$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right]\sigma_t$$

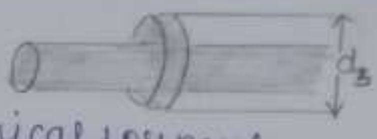
spigot


(iv) Calculate the outside diameter d_1 of the socket on the basis of tensile stress in the socket

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t$$

socket


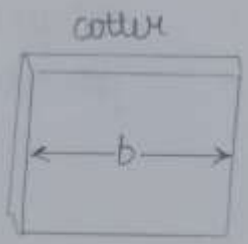
The diameter of the spigot collar d_3 and the diameter of the socket collar d_4 are calculated by the following empirical formula:



$$d_3 = 1.5d$$
$$d_4 = 2.4d$$

vi) The dimensions 'a' and 'c' are calculated by the following empirical formula: $a = c = 0.75d$

vii) Calculate the width 'b' of the cotter by shear consideration, by bending consideration: $b = \frac{P}{2\tau t}$



$$b = \sqrt{\frac{3P}{t\sigma_b}\left(\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right)}$$

From these two values select the maximum value

(viii) Check the crushing and and shear stresses in the spigot end: $\sigma_c = \frac{P}{td_2}$; $\tau = \frac{P}{2ad_2}$

(ix) Check the crushing and shear stresses in the socket end:

$$\sigma_c = \frac{P}{(d_4 - d_2)t} \quad ; \quad \tau = \frac{P}{2(d_4 - d_2)\cdot c}$$

(x) Calculate the thickness t_1 of the spigot collar by the following empirical relationship:

$$t_1 = 0.45d$$

The yield strength in compression is twice the yield strength in tension. The shear strength is half the yield strength in torsion.

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Q.3

Ans. Selection of Materials :→ on the basis of strength, the material of the two rods and the cotter is selected as plain carbon steel of grade 30C8 (σ_{yt} = 400 N/mm²). The factor of safety for the rods, spigot end and socket end is assumed as 6. while for other, it is taken as 4.

Calculation of Permissible stresses :→ The permissible for rods, spigot end and socket end are as follows :

$$\sigma_t = \frac{\sigma_{yt}}{FOS} = \frac{400}{6} = 66.67\ N/mm^2$$

$$\sigma_c = \frac{\sigma_{yc}}{FOS} = \frac{2 \cdot \sigma_{yt}}{FOS} = 133.33\ N/mm^2$$

$$\tau = \frac{\sigma_{sy}}{FOS} = \frac{0.5\,\sigma_{yt}}{FOS} = 33.33\ N/mm^2$$

Permissible stresses for the cotter are as follows :

$$\sigma_t = \frac{\sigma_{yt}}{FOS} = \frac{400}{4} = 100\ N/mm^2$$

$$\tau = \frac{\sigma_{sy}}{FOS} = \frac{0.5\,\sigma_{yt}}{FOS} = 50\ N/mm^2$$

Calculation of dimensions :→ The dimensions of the cotter joint are determined as :

Step 1 :→ Diameter of Rods :

$$d = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4 \times 50 \times 10^3}{\pi (66.67)}} = 30.90 \approx 32\ mm$$

Step : 2 :→ Thickness of cotter

$$t = 0.31\,d = 0.31 \times 32$$

$$\therefore t = 9.92 \approx 10 \text{ mm}$$

Step : 3 :→ Diameter (d_2) of spigot

$$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right] \sigma_t$$

$$50 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times 10\right](66.67)$$

$$d_2^2 - 12.73\,d_2 - 954.88 = 0$$

$$\therefore d_2 = 37.91 \approx 40 \text{ mm}$$

Step : 4 :→ outer diameter (d_1) of socket

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right] \sigma_t$$

$$50 \times 10^3 = \left[\frac{\pi}{4}(d_1^2 - 40^2) - (d_1 - 40)(10)\right](66.67)$$

$$d_1^2 - 12.73\,d_1 - 2045.59 = 0$$

$$\therefore d_1 = 52.04 \approx 55 \text{ mm}$$

Step : 5 :→ Diameter of spigot collar (d_3) and socket collar (d_4)

$$d_3 = 1.5d = 1.5 \times (32) = 48 \text{ mm}$$

$$d_4 = 2.4\,d = 2.4 \times (32) = 76.8 \approx 80 \text{ mm}$$

Step : 6 :→ Dimensions 'a' and 'c'

$$a = c = 0.75\,d = 0.75 \times (32) = 24 \text{ mm}$$

Step : 7 :→ Width of cotter

$$b = \frac{P}{2\tau t} = \frac{50 \times 10^3}{2 \times 50 \times 10} = 50 \text{ mm} \quad \text{— (a)}$$

$$\text{or } b = \sqrt{\frac{3P}{t\sigma_b}\left(\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right)} = 50 \text{mm} \quad \text{— (b)}$$

$\therefore$ from (a) and (b) : $b = 50$ mm

Step: 8 :→ Check for crushing and shear stresses in spigot

$$\sigma_c = \frac{P}{t d_2} = \frac{50 \times 10^3}{10 \times 40} = 125 \text{ N/mm}^2$$

$$\tau = \frac{P}{2 a d_2} = \frac{50 \times 10^3}{2 \times 24 \times 40} = 26.04 \text{ N/mm}^2$$

$\therefore \sigma_c < 133.33 \text{ N/mm}^2$ and $\tau < 33.33 \text{ N/mm}^2$

Step: 9 :→ Check for crushing and shear stress in socket end

$$\sigma_c = \frac{P}{(d_4 - d_2)t}$$

$$\sigma_c = \frac{50 \times 10^3}{(80-40)(10)} = 125 \text{ N/mm}^2$$

$$\tau = \frac{P}{2(d_4 - d_2)c}$$

$$\tau = \frac{50 \times 10^3}{2(80-40)(24)} = 26.04 \text{ N/mm}^2$$

$\therefore \sigma_c < 133.33 \text{ N/mm}^2$ and $\tau < 33.33 \text{ N/mm}^2$

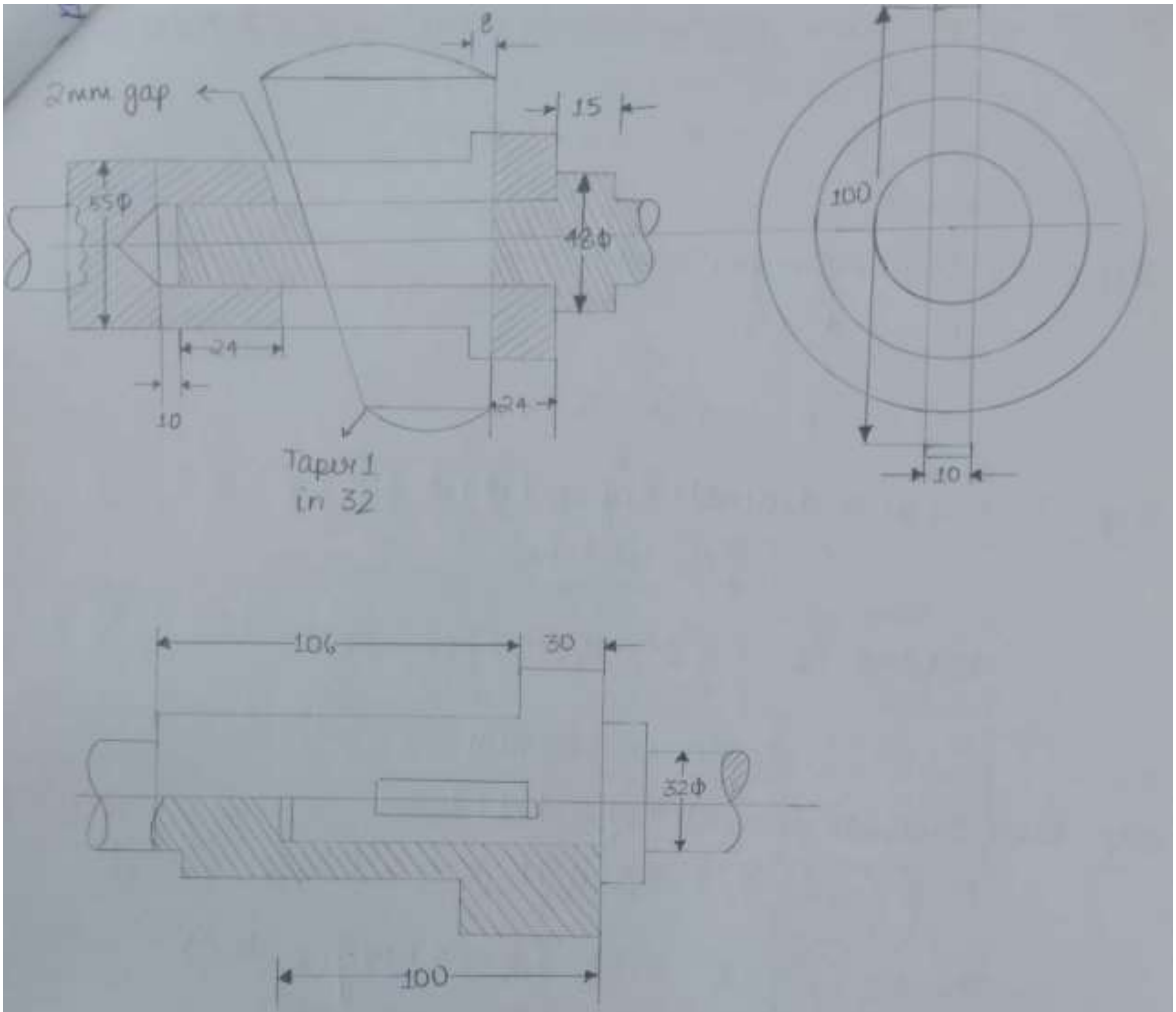
The stresses induced in the spigot ends and socket ends are within limits.

Step: 10 :→ Thickness of spigot collar

$$t_1 = 0.45d = 0.45 \times 32$$

$$\therefore t_1 = 14.4 \approx 15 \text{ mm}$$

The taper for the cotter is 1 in 32.



DIMENSIONED SKETCH OF COTTER JOINT

Q.4

Ans. Given: $\sigma_{yt} = 380 N/mm^2$

FOS = 6

t = 15 mm

d = 50 mm

Step: 1 :→ Permissible Stresses :

$$\sigma_c = \frac{S_{yc}}{FOS} = \frac{2\sigma_{yt}}{FOS} = \frac{2 \times 380}{6} = 126.67 \ N/mm^2$$

$$\tau = \frac{\sigma_{sy}}{FOS} = \frac{0.5\sigma_{yt}}{FOS} = \frac{0.5 \times 380}{6} = 31.67 \ N/mm^2$$

$$\sigma_t = \frac{\sigma_{yt}}{FOS} = \frac{380}{6} = 63.33 \ N/mm^2$$

Step : 2 :→ Load Acting on rods

$$P = \frac{\pi}{4} d^2 \times \sigma_t = \frac{\pi}{4}(50)^2 \times 63.33$$

$$\therefore P = 124348.16 \ N$$

Step : 3 :→ Inside diameter of socket (d_2)

$$P = \left(\frac{\pi}{4} d_2^2 - d_2 t\right)\sigma_t$$

$$124348.16 = \left[\frac{\pi}{4} d_2^2 - d_2(15)\right](63.33)$$

$$\therefore d_2 = 60.45 \approx 65 \ mm$$

Step : 4 :→ Outside diameter of socket (d_1)

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t$$

$$124348.16 = \left[\frac{\pi}{4}(d_1^2 - 65^2) - (d_1 - 65)15\right](63.33)$$

$$\therefore d_1 = 84.21 \approx 85 \ mm$$

Step : 5 :→ Diameter of socket collar (d_4)

$$\sigma_c = \frac{P}{(d_4 - d_2)t}$$

$$126.67 = \frac{124348.16}{(d_4 - 65)(15)}$$

$$\therefore d_4 = 130.44 \approx 135 \ mm$$

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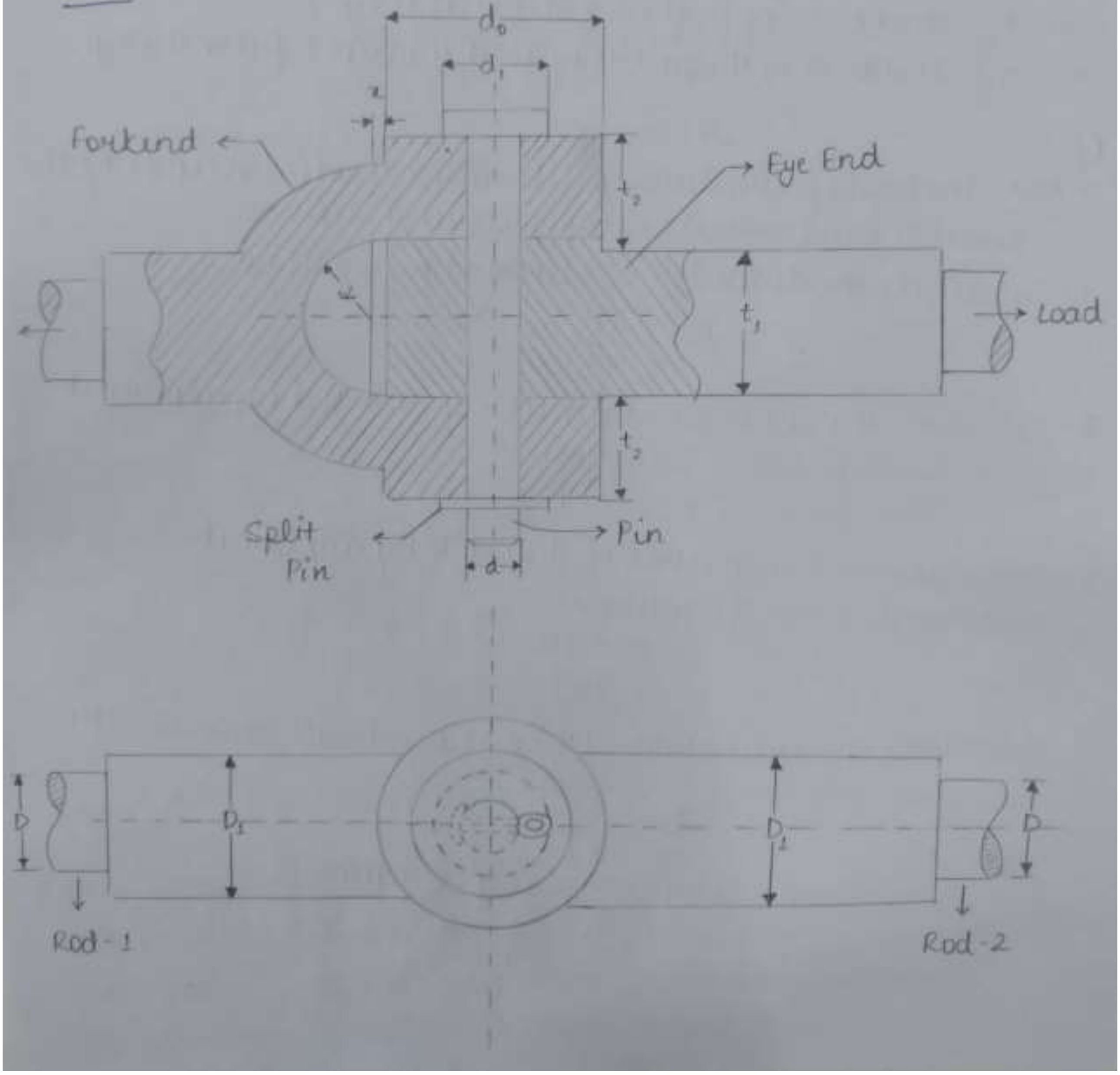
Step 6 :→ Dimensions of a and c

$$a = \frac{P}{2 d_2 \tau} = \frac{124348.16}{2 \times 65 \times 31.67} = 30.20 \approx 35 \text{ mm}$$

$$c = \frac{P}{2(d_4 - d_2)\tau} = \frac{124348.16}{2(135-65)\,31.67} = 28.04 \approx 30 \text{ mm}$$

Q.5
→ Ans.



Notations used in design :

(i) P = Tension in rod (Load on the joint)
(ii) D = Diameter of rod
(iii) D_1 = Enlarged diameter of rod
(iv) d = diameter of pin
(v) d_1 = diameter of pin ahead
(vi) d_0 = outer diameter of eye or fork
(vii) t_1 = thickness of eye
(viii) t_2 = thickness of forked end (double eye)
(ix) x = distance of the centre of fork radius R from the eye.

Q.6

Ans. The basic procedure to determine the dimensions of the knuckle joint consists of the following steps :

1. Calculate the diameter of each rod by equation :

$$D = \sqrt{\frac{4P}{\pi \sigma_t}}$$

2. Calculate the enlarged diameter of each rod by empirical relationship using equation :

$$D_1 = 1.1 D$$

3. Calculate the dimensions of 'a' and 'b' by empirical relationship using equation :

$$a = 0.75 D$$
$$b = 1.25 D$$

4. Calculate the diameters of the pin by shear consideration using equation :

$$d = \sqrt{\frac{2P}{\pi \tau}}$$

and bending consideration using equation :

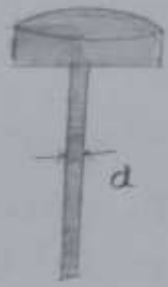
$$d = \sqrt[3]{\frac{32}{\pi \sigma_t} \times \frac{P}{2}\left(\frac{b}{4} + \frac{a}{3}\right)}$$

Calculate the dimensions d_0 and d_1 by empirical relationship using equations respectively :

$$d_0 = 2d \quad \text{and} \quad d_1 = 1.5d$$

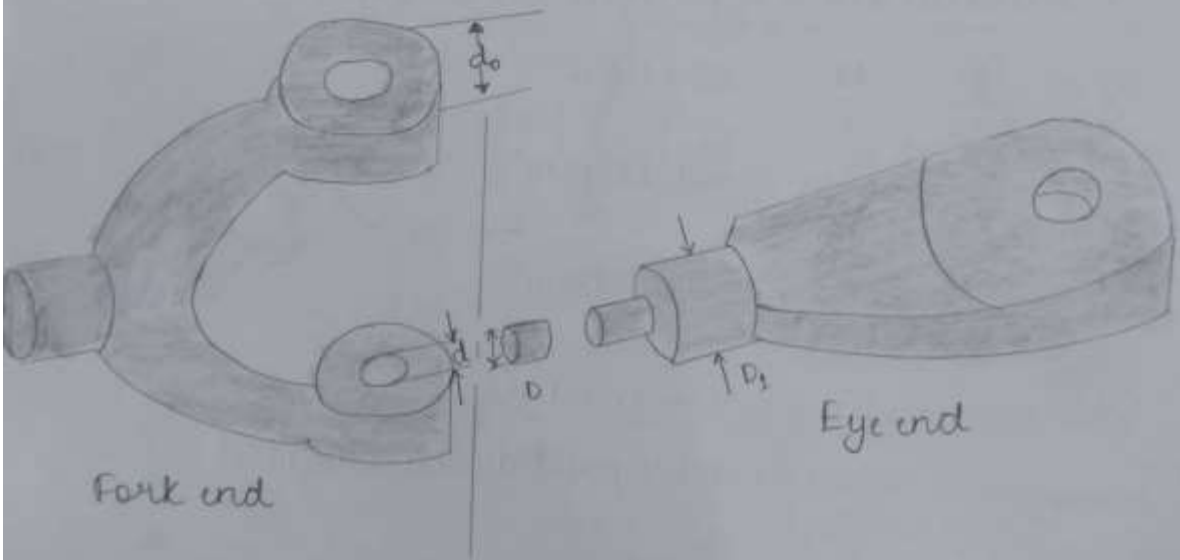
6. Check the tensile, crushing and shear stresses in the eye by using equations respectively :

$$\sigma_t = \frac{P}{b(d_0 - d)} \; ; \; \sigma_c = \frac{P}{bd} \; ; \; \tau = \frac{P}{b(d_0 - d)}$$



7. Check the tensile, crushing and shear stresses in the fork by using three equations respectively:

$$\sigma_t = \frac{P}{2a(d_0 - d)} \; ; \; \sigma_c = \frac{P}{2ad} \; ; \; \tau = \frac{P}{2a(d_0 - d)}$$



Q.7

Ans. Given: $P = (50 \times 10^3)$ N

Part : 1 → Selection of Material

The knuckle joint is subjected to an axial tensile force. Therefore, yield strength is the criteria required for the selection of material for the knuckle joint. The pin is shear stresses and bending stresses. Therefore, strength is also the criteria for the material selection of the pin. On strength basis, the material for two rods and pin is selected as plain carbon steel of Grade 30C8 ($S_{yt} = 400 N/mm^2$). It is further assumed that the yield strength in compression is equal to yield strength in tension. In practice, the compressive strength of steel is much higher than its tensile strength.

Part : 2 → Selection of Factor of Safety :

In stress analysis of knuckle joint, the effect of stress concentration is neglected. To account for this effect, a higher factor of safety 5 is assumed in the present design.

Part : 3 → Calculation of permissible stresses :

$$\sigma_t = \frac{S_{yt}}{FOS} = \frac{400}{5} = 80 N/mm^2$$

$$\sigma_c = \frac{S_{yc}}{FOS} = \frac{S_{yt}}{FOS} = 80 N/mm^2$$

$$\sigma = \frac{S_{sy}}{FOS} = \frac{0.5 S_{yt}}{FOS} = 40 N/mm^2$$

Part : 4 → Calculation of dimensions :

The dimensions of the knuckle joint are calculated by the procedure :

Step : 1 → Diameter of rods

$$D = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4 \times 50 \times 10^3}{\pi (80)}} = 28.21 \approx 30 mm$$

Step : 2 → Enlarged diameter of rods (D_1)

$$D_1 = 1.1 D = 1.1 \times 30 = 33 \approx 35 mm$$

Step : 3 → Dimensions of 'a' and 'b'

$a = 0.75D = 0.75 \times 30 = 22.5 \approx 25\ mm$

$b = 1.25D = 1.25 \times 30 = 37.5 \approx 40\ mm$

Step : 4 → Diameter of pin

$$D = \sqrt{\frac{2P}{\pi \tau}} = \sqrt{\frac{2 \times 50 \times 10^3}{\pi (40)}} = 28.21 \approx 30\ mm$$

also, $$d = \sqrt[3]{\frac{32}{\pi \sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

$d = 38.79 \approx 40\ mm$

$\therefore d = 40\ mm$

Step : 5 → Dimensions d_o and d_1

$d_o = 2d = 2 \times 40 = 80\ mm$

$d_1 = 1.5d = 1.5 \times 40 = 60\ mm$

Step : 6 → Check for stresses in eye

$$\sigma_t = \frac{P}{b(d_o - d)} = \frac{50 \times 10^3}{40(80-40)} = 31.25\ N/mm^2$$

$\therefore \sigma_t < 80\ N/mm^2$

$$\sigma_c = \frac{P}{bd} = \frac{50 \times 10^3}{40 \times 40} = 31.25\ N/mm^2$$

$\therefore \sigma_c < 80\ N/mm^2$

$$\tau = \frac{P}{b(d_o - d)} = \frac{50 \times 10^3}{40(80-40)} = 31.25\ N/mm^2$$

$\therefore \tau < 80\ N/mm^2$

Step : 7 → Check for stresses in fork

$$\sigma_t = \frac{P}{2a(d_o - d)} = \frac{50 \times 10^3}{2(25)(80-40)} = 25\ N/mm^2$$

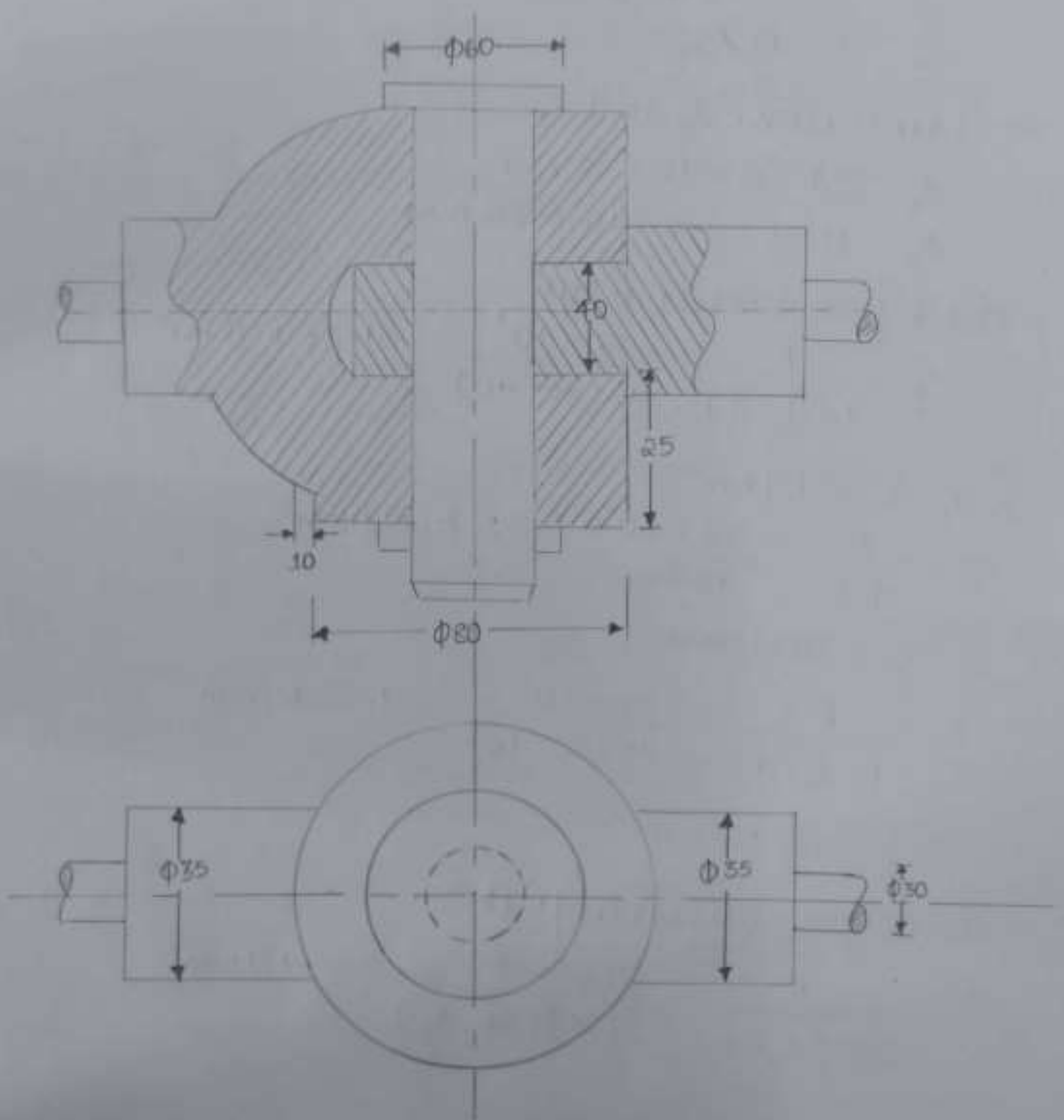
$\therefore \sigma_t < 80 \, N/mm^2$

$$\sigma_c = \frac{P}{2ad} = \frac{50 \times 10^3}{2 \times 25 \times 40} = 25 \, N/mm^2$$

$\therefore \sigma_c < 80 \, N/mm^2$

$$\tau = \frac{P}{2a(d_0 - d)} = \frac{50 \times 10^3}{2(25)(40)} = 25 \, N/mm^2$$

$\therefore \tau < 40 \, N/mm^2$



Step: 6 → Check for stresses in eye

$$\sigma_t = \frac{P}{b(d_0-d)} = \frac{25\times10^3}{19(40-20)} = 65.78\ N/mm^2$$

$$\therefore \sigma_t < 152\ N/mm^2$$

$$\sigma_c = \frac{P}{bd} = \frac{25\times10^3}{19(20)} = 65.78\ N/mm^2$$

$$\therefore \sigma_c < 152\ N/mm^2$$

$$\tau = \frac{P}{b(d_0-d)} = \frac{25\times10^3}{19(40-20)} = 65.78\ N/mm^2$$

$$\therefore \tau < 87.70\ N/mm^2$$

Step: 7 → Check stresses in fork

$$\sigma_t = \frac{P}{2a(d_0-d)} = \frac{25\times10^3}{2\times12(40-20)} = 52.08\ N/mm^2$$

$$\therefore \sigma_t < 152\ N/mm^2$$

$$\sigma_c = \frac{P}{2ad} = \frac{25\times10^3}{2\times12\times20} = 52.08\ N/mm^2$$

$$\therefore \sigma_c < 152\ N/mm^2$$

$$\tau = \frac{P}{2a(d_0-d)} = \frac{25\times10^3}{2(12)(40-20)} = 52.08\ N/mm^2$$

$$\therefore \tau < 87.70\ N/mm^2$$

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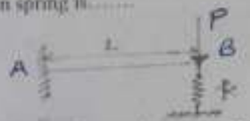
Gate Sheet-2

1. Two equal length of steel wire of the same diameter are made into springs ... and 60 mm respectively. The stiffness ratio of S1 to S_2 is:
(A) $(60/75)^2$ (B) $(60/75)^3$ (C) $(75/60)^2$ (D) $(75/60)^3$

2. A closely coiled helical spring of 20 cm mean diameter is having 25 coils of 2 cm diameter rod. The modulus of rigidity of the material is 10^7 N/cm². What is the stiffness for the spring in N/cm?
(A) 50 (B) 100 (C) 250 (D) 500

3. A steel strip of length L=200 mm is fixed at end A and rests at B an vertical spring of stiffness k= 2 N/mm. The steel strip is 5 mm wide and 10 mm thick. A vertical load P= 50 N is applied at B as shown. Considering E=200 GPa the force developed (in Newtons) in spring is.......



4. A compression spring is made of music wire of 2 mm diameter with a shear strength and shear modulus of 800 MPa and 80 GPa respectively. The mean coil diameter is 20 mm. free length is 40 mm and the number of active coils are 10. If the mean diameter is reduced to 10mm, the stiffness of the spring is.
(A) decreased by 8 times (B) decreased by 2 times (C) increased by 2 times (D) increased by 8 times

5. A helical compression spring made of wire of circular cross-section is subjected to a compressive load. The maximum shear stress induced in the cross-section of the wire is 24 MPa. For the same compressive load, if both the wire diameter and the mean coil diameter are doubled, the maximum shear stress (in MPa) induced in the cross-section of the wire is .... **GATE 2017**

6. The spring constant of a helical compression spring DOES NOT depends on **GATE 2016**
(A) coil diameter (B) material strength (C) number of active turns (D) wire diameter

7. Cracks in helical springs used in Railway carriages usually start on the inner side of the coil because of fact that **IES 1994**
(A) it is subjected to higher stress than the outer side
(B) It is subjected to higher cyclic loading than the outer side
(C) it is more stretched than the outer side during the manufacturing process
(D) it has lower curvature than the outer side

8. Two helical springs of same material and of equal circular cross-section and length and number of turns, but having radii 20 mm and 40 mm, kept concentrically (smaller radius spring within the layer of larger radius spring), are compressed between two planes with a load P. The inner spring carry a load equal to. **IES 1994**
(A) P/2 (B) 2P/3 (C) P/9 (D) 8P/9

9. The maximum shear stress occurs on the outer-most fibers of a circular shaft under torsion. In a closely coiled helical spring, the maximum shear stress occurs on the **IES 1999**
(A) outermost fibers (B) fibers at mean diameter (C) innermost fibers (D) end coils

10. When closely coiled helical spring is subjected to a couple about its axis, the stress induced in the wire of the spring is
(A) bending stress only (B) direct stress only
(C) a combination of torsional shear stress and bending stress
(D) a combination of bending stress and direct shear stress

Belts

11. The percentages improvement of power capacity of a flat belt drive when the wrap angle at the driving pulley is increased from 150° to 210° by an idler arrangement for a friction coefficient of 0.3 is. **GATE 1997**
(A) 25.21 (B) 33.92 (C) 40.17 (D) 67.85

12. The difference between tension on tight and slack side of a belt drive is 3000 N. If the belt speed is 15 m/sec, the transmitted power in kW is. **GATE 1998**
(A) 45 (B) 22.5 (C) 90 (D) 100

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13. A wire rope is designated as 6 x 9 standard hoisting. The number 6 x 9 represent **GATE 2003**
(A) Diameter in mm x length in m (B) Diameter in cm x length in m
(C) Number of strands x number of wire in each strand (D) Number of wires in each strand x number of strands

14. The ratio of tension on the tight side to that on the slack side in a flat belt drive is **GATE 2000**
(A) Proportional to the product of coefficient of friction and lap angle
(B) An exponential function of the product of coefficient of friction and lap angle
(C) Proportional to the lap angle
(D) Proportional to the coefficient of friction

15. A belt drive has an angle of wrap of 160° on the smaller pulley adding idler increases the wrap angle to 200°. The slack side tension is the same in both cases and the centrifugal force is negligible. By what.... percentage is the torque capacity of the belt drive increased by adding the idler? Use $\mu = 0.3$ **GATE 2001**

16. With regard to belt drives with given pulley diameters, centre distance and the coefficient of friction between the pulley and the belt materials, which of the statements below is false? **GATE 1999**
(A) a crossed flat belt configuration can transmit more power than an open flat belt configuration
(B a V- belt has greater power transmission capacity than an open belt.
(C) power transmission is greater when belt tension is higher due to centrifugal effects than the same belt drive when centrifugal affects are absent.
(D) power transmission is the greatest just before the point of slipping is reached.

17. A flat belt open drive transmits 1.5 kW of power. The coefficient of friction between the belt and drive pulley is 0.25 and the lap angle is 159°. The drive pulley is rotating in the clock wise direction with the linear belt speed of 3.75 m/sec. Determine X and Y components of the reaction force on the drive pulley shaft. **GATE 1999**

18. In the assembly design of shaft, pulley and key the weakest number is. **IES 1998**
(A) pulley (B) key (C) shaft (D) none

19. Source of power loss in a chain drive are given below: **IES 1995**
(A) friction between chain and sprocket teeth (B) overcoming the chain stiffness.
(C) overcoming the friction in shaft bearing (D) friction resistance to the motion of chain in air or lubricant

20. Which one of the following pairs is not correctly matched? **IES 1995**
(A) positive drive.....Belt drive (B) high velocity ratio.....Worm gearing
(C) to connect non-parallel and non-intersecting shafts.....Spiral gearing
(D) finished noise and smooth operation..........Helical gear

21. A 50 kW motor using six V-belts is used in a pump mill. If one of the belts breaks after a month of continuous running, then
(A) the broken belt is to be replaced by a similar belt (B) all the belts are to be replaced
(C) the broken belt and two adjacent belts are to be replaced
(D) the broken belt and one adjacent belt is to be replaced

22. When compared to a rod of same diameter and material, a wire rope **IES 1994**
(A) is less flexible (B) has a much smaller load carrying capacity
(C) does not provide much warning before failure (D) provides much greater time for remedial action

23. In a multiple V-belt drive, when a single belt is damaged, it is preferable to change the complete set to **IES 1993**
(A) reduce vibration (B) reduce slip (C) ensure uniform loading (D) ensure proper alignment

24. When a belt drive is transmitting maximum power **IES 1996**
(A) effective tension is equal to centrifugal tension (B) effective tension is half of centrifugal tension
(C) driving tension on slack side is equal to the centrifugal tension
(D) driving tension on tight side is twice the centrifugal tension

25. A flat belt transmits 112 kW at a speed of 24 m/s between two pulleys of diameters 250 mm and 400 mm, having pulley center distance of 1000 mm. The allowable belt stress is 8.6 MN/mm² and the belts are available having thickness to-width ratio of 0.1 and a material density of 1100 kg/m³. If coefficient of friction between the belt and pulleys is 0.3, determine the minimum required width of the belt. **GATE 1994**

Solution of Gate Sheet-2

Ans-1 :- We know that $\delta = \frac{8WD^3 n}{Gd^4}$

Given:-

→ Lenth of both spring are equal

$d_1 = d_2 = d$

$D_1 = 75\,mm$

$D_2 = 60\,mm$

$$\frac{\delta}{W} = \frac{8D^3 n}{Gd^4} \qquad S = \frac{W}{\delta}$$

$$\frac{\delta}{W} = \frac{1}{S}$$

$$\frac{1}{S} = \frac{8 \cdot D^3 n}{Gd^4}$$

$$S = \frac{Gd^4}{8 \cdot D^3 n} \quad -(i)$$

[where S is stifness]

Equal length of wire, means $\pi D_1 n_1 = \pi D_2 n_2$

or $\frac{n_1}{n_2} = \frac{D_2}{D_1}$

$$\frac{S_1}{S_2} = \frac{\frac{Gd_1^4}{8D_1^3 n_1}}{\frac{Gd_2^4}{8 \cdot D_2^3 n_2}} \quad -(i)$$

$$= \frac{D_2^3 n_2}{D_1^3 n_1} = \frac{D_2^3}{D_1^3} \times \left(\frac{D_1}{D_2}\right) = \frac{D_2^2}{D_1^2}$$

$$\frac{S_1}{S_2} = \left(\frac{D_2}{D_1}\right)^3 \qquad \cancel{\frac{S_1}{S_2} = \left(\frac{75}{60}\right)^3} \qquad \left(\frac{S_1}{S_2}\right) = \left(\frac{60}{75}\right)^2$$

So ~~(d)~~ is correct answer (A) $\left(\frac{60}{75}\right)^2$

Ans2.-

$$\text{Stifness of spring } (K) = \frac{Gd^4}{8D^3 n}$$

$$K = \frac{10^7 (N/cm^2) \times 2^4 (cm^4)}{8 \times 20^3 (cm^3) \times 25} = 100\,N/cm$$

So B is correct answer

Ans 3 → Given → $L = 200\,mm$, $K = 2\,N/mm$, width of beam = 5 mm, thickness = 10 mm

$P = 50\,N$, $E = 200\,GPa$

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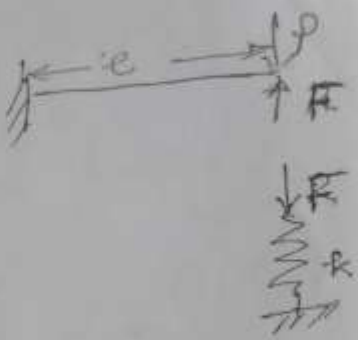
for a cantilever ...

$$\delta_e = \frac{PL^3}{3EI}$$



Deflection of spring $\delta_s = \frac{F}{K}$

Put at point B $\delta_e = \delta_s$

$$\frac{(P-F)L^3}{3EI} = \frac{F}{K}$$

$$\frac{(50-F)\times(200)^3}{3.0\times 200\times 10^3\times \frac{5\times 10^3}{2}} = \frac{F}{2}$$

$\boxed{F = \cancel{3.2}N}$ 3.007 N

Ans-4 $k_1 = \frac{Gd_1^4}{8D_1^3 \cdot n_1}$, $k_2 = \frac{Gd_2^4}{8D_2^3 \cdot n_2}$

where k_1 is stiffness

$d_1 = d_2 = 2\,mm$

$G = 80\,GPa$

$D_1 = 20$, $D_2 = 10\,mm$ $\qquad \frac{D_2}{D_1} = \frac{10}{20} = \frac{1}{2}$

$n_1 = n_2 = 10$

$$\frac{k_1}{k_2} = \left(\frac{D_2}{D_1}\right)^3 = \frac{1}{8}$$

$$k_2 = 8k_1$$

So (D) is correct answer

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~~Ans~~

$\delta = \frac{8FD^3n}{Gd^4}$

Here $\delta_1 = \delta_2$

$\frac{8W_1(2\times20)^3 n}{Gd^4} = \frac{8W_2(2\times40)^3 n}{Gd^4}$

$\therefore W_1 = 8W_2$

Ans-8:-

To be explained properly

$\frac{W_o}{W_i} = \frac{R_i^3}{R_o^3} = \left(\frac{20}{40}\right)^3 = \frac{1}{8}$

$W_o = \frac{W_i}{8}$ so $W_i + \frac{W_i}{8} = P$

$W = W_1 + W_2 =$

$P = W_1 + \frac{W_1}{8} = \frac{9}{8}$

$\therefore W_1 = \frac{8P}{9}$

$\boxed{W_1 = \frac{8P}{9}}$ Ans

So → d is correct answer

Ans-10 — Ans A → Bending stress only.

24

Belt Grade Sheet solution:

Ans-11 Case-1 $\frac{T_1}{T_2} = e^{\mu\theta}$ $\frac{T_1}{T_2} = e^{0.3\times\left(\frac{5\pi}{6}\right)}$ $T_1 = 2.193 T_2$

$P = (T_1 - T_2) \cdot v$ watt

$P_1 = 1.193 T_2 v$

Case-II $\frac{T_1}{T_2} = e^{\mu\theta} \Rightarrow \frac{T_1}{T_2} = e^{0.3\times(7\pi/6)}$

$T_1 = 3.003 T_2$ $P_1 = 2.003 T_2 v$ watt

Improvement in power capacity $= \frac{P_2 - P_1}{P_1} \times 100\%$

67.8%.

option d is correct

Ans-12 $P = (T_1 - T_2) \cdot V$

$= 3000 \times 15 = 45000 = 45 kW$ Ans

Ans 13 → (C) Number of stands × number of wire in each stand.

Ans-14 :→ Ans - b $\frac{T_1}{T_2} = e^{\mu\theta}$

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Ans-16 Tension in Tight side of the belt

$T_1 = T - T_c$

$\boxed{P = (T_1 - T_c)\, v}$ [So power transmission capacity reduce due to centrifugal effects]

Ans - C is ~~comm~~ false

Ans-17

Ans-18:- (B) key

Ans-19 - Power loss in desending order takes place as 1,2,3 and 4

Ans-20:- Ans-(a) Positive drive -- Belt drive due to slip present, the velocity ration will not remain constant and hence, belt drive is not a positive drive.

Ans-21 (B) All the belts are to replaced.

Ans-22 - (B) Has a much smaller load carrying capacity.

Ans-23 - (C) Ensure uniform loading.

Ans-24: (D) Driving tension on tight side is twice the centrifugal tension.

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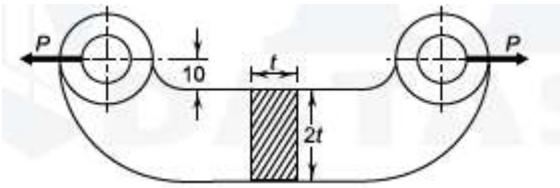
Technical Quiz Papers

Design of Machine Element-I

Unit Test-I

1. According to Indian standard specifications, a grey cast iron designated by 'FG 200' means that
   a) Carbon content is 2% (c) maximum compressive strength is 200 N/mm^2
   b) minimum tensile strength is 200 N/mm^2 (d) maximum shear strength is 200 N/mm^2
2. An offset link subjected to a force of 25 kN is shown in Fig. It is made of grey cast iron FG300 and the factor of safety is 3. Determine the dimensions of the cross-section of the link. (Hint; cross-section subjected to direct stress and bending stress)



   a) 26 mm, 52 mm c) 20 mm, 40 mm
   b) 16 mm, 32 mm d) 36 mm, 72 mm
3. The stresses induced at a critical point in a machine component made of steel 45C8 (Syt = 380 N/mm^2) are as follows: σx=100 MPa; σy=40 MPa ; τxy=80 MPa Calculate the factor of safety by using following theories (i) the maximum normal stress theory, (ii) the maximum shear stress theory, and (iii) the distortion energy theory.
   a) 3.44, 3.22, 3.32 c) 2.44, 2.22, 2.32
   b) 1.44, 1.22, 1.32 d) 4.44, 4.22, 4.32
4. The stress which vary from a minimum value to a maximum value of the same nature (i.e. tensile or compressive) is called
   a) repeated stress c) yield stress
   b) fluctuating stress d) alternating stress
5. Stress concentration factor is defined as the ratio of
   a) maximum stress to the endurance limit c) nominal stress to the endurance limit
   b) maximum stress to the nominal stress d) nominal stress to the maximum stress
6. Two rods are connected by means of a knuckle joint. The axial force P acting on the rods is 25 kN. The rods and the pin are made of plain carbon steel 45C8 (Syt = 380 N/mm^2) and the factor of safety is 2.5. The yield strength in shear is 57.7% of the yield strength in tension. Calculate: (i) the diameter of the rods, and (ii) the diameter of the pin.
   a) 16.47, 15.47 c) 14.47, 13.47
   b) 18.47, 17.47 d) 10.47, 11.47
7. Two mild steel rods are connected by cotter joint subjected to a pull of 30 kN and the cotter is made from a steel plate of 10 mm thickness. The maximum permissible stresses are 55 MPa in

tension; 40 MPa in shear and 110 MPa in crushing. Calculate: (i) the diameter of the rods, (ii) the diameter of spigot (iii) outside diameter of socket

a) 24 mm, 34 mm, 42 mm
b) 27 mm, 34 mm, 45 mm
c) 32 mm, 38 mm, 48 mm
d) 30 mm, 38 mm, 40 mm

8. Refer above question of cotter joint, Calculate the width of the cotter
   a) 38 mm
   b) 58 mm
   c) 28 mm
   d) 48 mm
9. Refer above question of cotter joint, find out the shear stress and crushing stress in spigot end
   a) 21.01 MPa, 88.24 MPa
   b) 31.01 MPa, 68.24 MPa
   c) 18.01 MPa, 78.24 MPa
   d) 41.01 MPa, 98.24 MPa
10. Refer above question of cotter joint, find out the shear stress and crushing stress in socket end
    a) 33.04 MPa, 76.77 MPa
    b) 38.03 MPa, 106.77 MPa
    c) 23.04 MPa, 96.77 mm
    d) 35.03 MPa, 100.77 MPa

Assignments (As Per RTU QP Format)

SKIT– JAIPUR DEPARTMENT OF MECHANICAL ENGINEERING	
B.Tech. V Semester (2022-23)	**DESIGN OF MACHINE ELEMENTS – I (5ME04-4)**
Assignment No. 1	

PART A (Short Answer Questions, upto 25 words)

1. What are the steps involved in design of a machine element?
2. What are the economic aspects form the selection of material?
   RTU-2019
3. Define the term interchangeability and standardization.
4. Give IS designation of: **(a)** Grey Cast Iron having minimum tensile strength of 220MPa **(b)** Plain carbon steel having average content of 0.5% carbon and 0.8% manganese.
5. What are unilateral and bilateral tolerances?
6. A shaft of size Ø 49 ± 0.05 mm is to be fitted in a bush bearing having bore of Ø50 ± 0.02 mm. Using hole-basis system, Determine: (a) Hole tolerance & shaft tolerance (b) Minimum & maximum clearance.
7. What is factor of safety? Why is it necessary to use factor of safety?
8. State the followings:**(a)** maximum principal stress theory of failure **(b)** Maximum shear stress theory of failure **(c)** distortion energy theory of failure.
9. What is fatigue failure.
10. What is endurance limit?

PART B (Analysis/Application/Problem Solving Questions)

11. Explain the design considerations of casting products and machined parts with the neat sketches. RTU-2016
12. Explain the stress concentration causes and mitigation with suitable figures.
13. **(a)** Describe the various types of fits with neat diagrams and examples.
    RTU-2015
    (b) Calculate the tolerances, limits of sizes, and allowance for the shaft and hole assembly designated as 40 H6/f7. What type of fits will be established? Take the suitable data from design hand data book.
14. Write the design procedure for Spigot-Socket cotter joint with neat sketches
15. Explain the various mechanical properties of engineering materials.

RTU-2019

PART C (Descriptive / Design questions)

16. A cantilever rod of 40 mm diameter is loaded as shown in the Fig.1. If the tensile yield strength of the material is 300 MPa determine the factor of safety at point A using (i) Maximum principal stress theory (ii) Maximum shear stress theory (iii) Maximum distortion energy theory.

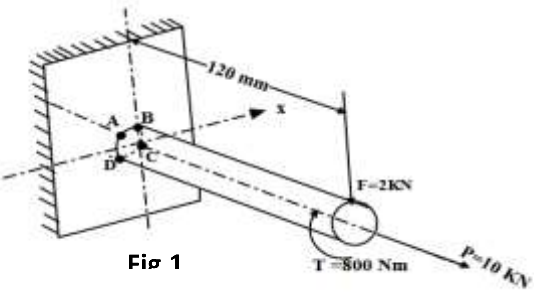


Fig 1

17. Design a spigot and socket type cotter Joint for axial load of 75 kN which alternately changes from tensile to compressive. Allowable stresses for the material used are 50 MPa in tensions, 40 MPa in shear, and 100 MPa in crushing. RTU-2014
18. Design a knuckle joint to connect two round rods subjected to a tensile load of 100 kN. The permissible stresses may be taken as 75 MPa in tension, 50 MPa in shear and 135 MPa in crushing.
    RTU-2015

19. **(a)** A wall bracket with a rectangular cross-section is shown in **Fig.2**. The depth of the cross-section is twice of the width. The force P acting on the bracket at 60^0 to the vertical is 5 kN. The material of the bracket is grey cast iron FG 200 and the factor of safety is 3.5. Determine the dimensions of the cross-section of the bracket. Assume maximum normal stress theory of failure.

(b) A beam of rectangular section is to support a load of 20 KN uniformly distributed over a span of 3.6 m when beam is simply supported. If the depth is to be twice the breadth, and the stress in beam is not exceed 7 N/mm^2, find the dimensions of the cross section. How could you modify the dimensions with 20 KN of concentrated load is present at centre with same breadth and depth ratio.

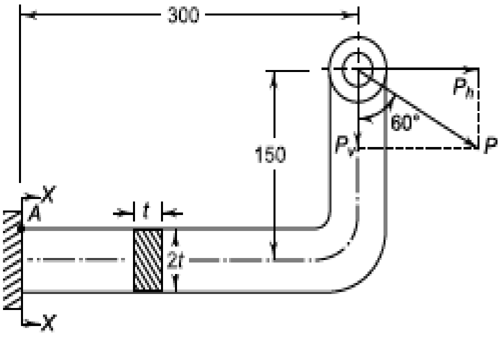


Fig.2 *Wall Bracket*

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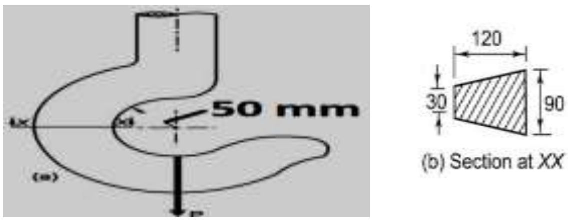
SKIT– JAIPUR	**DEPARTMENT OF MECHANICAL ENGINEERING**
B.Tech. V semester (2022-23)	**DESIGN OF MACHINE ELEMENTS – I (5ME4-04)**
Assignment No. 2	

PART A (Short Answer Questions, upto 25 words)

1. Which theories of failure are applicable for shaft design? RTU-2019
2. Explain the failure of key. RTU-2017
3. What do you understand by torsional rigidity? RTU-2016
4. What is the purpose of the rubber bush in bushed-pin flexible coupling?
5. How will you designate ISO metric coarse threads? RTU-2019
6. What do mean by bolt of uniform strength? RTU-2016
7. What is ‘overhauling’ of power screw?
8. What is a curved beam? Give practical examples of machine components made of curved beams.
9. What is self-locking screw?
10. What is difference between rigid and flexible coupling?

PART B (Analysis/Application/Problem Solving Questions)

11. A truck spring has 12 number of leaves, two of which are full length leaves. The spring supports are 1.05 m apart and the central band is 85 mm wide. The central load is to be 5.4 kN with a permissible stress of 280 MPa. Determine the thickness and width of the steel spring leaves. The ratio of the total depth to the width of the spring is 3. Also determine the deflectionof the spring. RTU-2018
12. The standard cross-section for a flat key, which is fitted on a 50 mm diameter shaft, is 16 X10 mm. The key is transmitting 475 N-m torque from the shaft to the hub. The key is made of commercial steel (S_{yt} = S_{yc} = 230 N/mm^2). Determine the length of the key, if the factor of safety is 3. RTU-2019
13. Design a muff coupling to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 (S_{yt} = S_{yc} = 400 N/mm^2). The sleeve is made of grey cast iron FG 200 (S_{ut} = 200 N/mm^2). The factorof safety for the shafts and key is 4. For the sleeve, the factor of safety is 6 based on ultimate strength. RTU-2017
14. The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm^2. It is held in position by means of 12 bolts.A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa. RTU-2018
15. A crane hook (curved beam) having an approximate trapezoidal cross section is shown in fig. It is made of plain carbon steel 45C8 (S_{yt}=380 N/mm^2) and factor of safety is 3.5. Determine the load carrying capacity of the hook (all dimensions in mm).

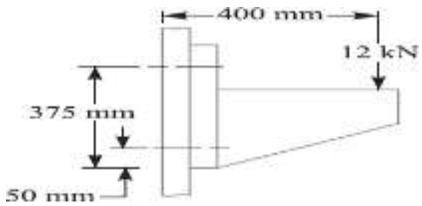


PART C (Descriptive / Design questions)

16. A right-angled bell-crank lever is to be designed to raise a load of 5 kN at the short arm end. The lengths of short and long arms are 100 and 450 mm respectively. The lever and the pins are made of steel 30C8 (Syt = 400 N/mm2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm2. The lever has a rectangularcross-section and the ratio of width to thickness is 3:1. The length to diameter ratio of the fulcrum pin is 1.25:1. Calculate (i)The diameter and the length of the fulcrum pin (ii) The shear stress in the pin (iii) The dimensions of the boss of the lever at the fulcrum (iv) The dimensions of the cross-section of the lever, assume that the arm of the bending moment on the lever extends up to the axis of the fulcrum. RTU-2019
17. A shaft is supported by two bearings placed 1 m apart. A 600 mm diameter pulley is mounted at a distance of 300 mm to theright of left hand bearing and this drives a pulley directly below it with the

help of belt having maximum tension of 2.25 kN. Another pulley 400 mm diameter is placed 200 mm to the left of right hand bearing and is driven with the help of electric motor and belt, which is placed horizontally to the right. The angle of contact for both the pulleys is 180° and μ= 0.24. Determine the suitable diameter for a solid shaft, allowing working stress of 63 MPa in tension and 42 MPa in shear for the material of shaft. Assume that the torque on one pulley is equal to that on the other pulley. RTU-2018

18. For supporting the travelling crane in a workshop, the brackets are fixed on steel columnas shown in Fig. The maximum load that comes on the bracket is 12 kN acting verticallyat a distance of 400 mm from the face of the column. The vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at a distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible valueof the tensile stress for bolt material is 84 MPa. Also find the cross-section of the arm ofthe bracket which is rectangular, take depth of arm of bracket as 250 mm.



19. Design a cast iron protective type flange coupling to transmit 15 kW at 900 from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used as shear stress for shaft, bolt and keymaterial = 40 MPa; crushing stress for bolt and key = 80 MPa; shear stress for cast iron = 8 MPa.

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SKIT– JAIPUR DEPARTMENT OF MECHANICAL ENGINEERING	
B.Tech. V Semester (2021-2022)	**DESIGN OF MACHINE ELEMENTS – I (5ME04-4)**
Assignment No. 1	

PART A (Short Answer Questions, upto 25 words)

11. What are the steps involved in design of a machine element?
12. What are the economic aspects form the selection of material? RTU-2019
13. Define the term interchangeability and standardization.
14. Give IS designation of: **(a)** Grey Cast Iron having minimum tensile strength of 220MPa **(b)** Plain carbon steel having average content of 0.5% carbon and 0.8% manganese.
15. What are unilateral and bilateral tolerances?
16. A shaft of size Ø 49 ± 0.05 mm is to be fitted in a bush bearing having bore of Ø50 ± 0.02 mm. Using hole-basis system, Determine: (a) Hole tolerance & shaft tolerance (b) Minimum & maximum clearance.
17. What is factor of safety? Why is it necessary to use factor of safety?
18. State the followings:**(a)** maximum principal stress theory of failure **(b)** Maximum shear stress theory of failure **(c)** distortion energy theory of failure.
19. What is fatigue failure.
20. What is endurance limit?

PART B (Analysis/Application/Problem Solving Questions)

14. Explain the design considerations of casting products and machined parts with the neat sketches. RTU-2016
15. Explain the stress concentration causes and mitigation with suitable figures.
16. **(a)** Describe the various types of fits with neat diagrams and examples. RTU-2015
    (b) Calculate the tolerances, limits of sizes, and allowance for the shaft and hole assembly designated as 40 H6/f7. What type of fits will be established? Take the suitable data from design hand data book.
15. Write the design procedure for Spigot-Socket cotter joint with neat sketches
16. Explain the various mechanical properties of engineering materials. RTU-2019

PART C (Descriptive / Design questions)

20. A cantilever rod of 40 mm diameter is loaded as shown in the Fig.1. If the tensile yield strength of the material is 300 MPa determine the factor of safety using (i) Maximum principal stress theory (ii) Maximum shear stress theory (iii) Maximum distortion energy theory.

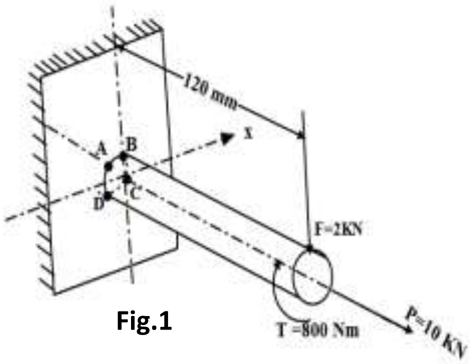


Fig.1

21. Design a spigot and socket type cotter Joint for axial load of 75 kN which alternately changes from tensile to compressive. Allowable stresses for the material used are 50 MPa in tensions, 40 MPa in shear, and 100 MPa in crushing. RTU-2014
22. Design a knuckle joint to connect two round rods subjected to a tensile load of 100 kN. The permissible stresses may be taken as 75 MPa in tension, 50 MPa in shear and 135 MPa in crushing. RTU-2015

23. **(a)** A wall bracket with a rectangular cross-section is shown in **Fig.2**. The depth of the cross-section is twice of the width. The force P acting on the bracket at 60^0 to the vertical is 5 kN. The material of the bracket is grey cast iron FG 200 and the factor of safety is 3.5. Determine the dimensions of the cross-section of the bracket. Assume maximum normal stress theory of failure.

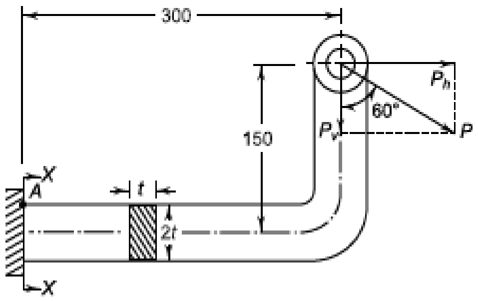


Fig.2 *Wall Bracket*

(b) A beam of rectangular section is to support a load of 20 KN uniformly distributed over a span of 3.6 m when beam is simply supported. If the depth is to be twice the breadth, and the stress in beam is not exceed 7 N/mm^2, find the dimensions of the cross section. How could you modify the dimensions with 20 KN of concentrated load is present at centre with same breadth and depth ratio.

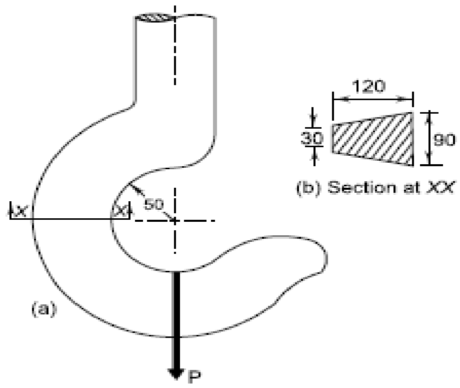
SKIT– JAIPUR	DEPARTMENT OF MECHANICAL ENGINEERING
B.Tech. V semester (2021-2022)	DESIGN OF MACHINE ELEMENTS – I (5ME4-04)
Assignment No. 2	

PART A (Short Answer Questions, upto 25 words)

1. Which theories of failure are applicable for shaft design? RTU-2019
2. Explain the failure of key. RTU-2017
3. What do you understand by torsional rigidity? RTU-2016
4. What is the purpose of the rubber bush in bushed-pin flexible coupling?
5. How will you designate ISO metric coarse threads? RTU-2019
6. What do mean by bolt of uniform strength? RTU-2016
7. What is 'overhauling' of power screw?
8. What is a curved beam? Give practical examples of machine components made of curved beams.
9. What is self-locking screw?
10. What is difference between rigid and flexible coupling?

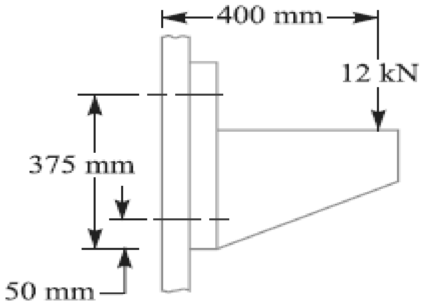
PART B (Analysis/Application/Problem Solving Questions)

11. The standard cross-section for a flat key, which is fitted on a 50 mm diameter shaft, is 16 X10 mm. The key is transmitting 475 N-m torque from the shaft to the hub. The key is made of commercial steel ($S_{yt} = S_{yc} = 230$ N/mm^2). Determine the length of the key, if the factor of safety is 3. RTU-2019
12. Design a muff coupling to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 ($S_{yt} = S_{yc} = 400$ N/mm^2). The sleeve is made of grey cast iron FG 200 ($S_{ut} = 200$ N/mm^2). The factor of safety for the shafts and key is 4. For the sleeve, the factor of safety is 6 based on ultimate strength. RTU-2017
13. A double-threaded power screw, with ISO metric trapezoidal threads is used to raise a load of 300 kN. The nominal diameter is 100 mm and the pitch is 12 mm. The coefficient of friction at the screw threads is 0.15. Neglecting collar friction, calculate **(i)** torque required to raise the load; **(ii)** torque required to lower the load; and **(iii)** efficiency of the screw.
14. The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm^2. It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa. RTU-2018
15. A crane hook (curved beam) having an approximate trapezoidal cross section is shown in fig. It is made of plain carbon steel 45C8 (S_{yt}=380 N/mm^2) and factor of safety is 3.5. Determine the load carrying capacity of the hook.



PART C (Descriptive / Design questions)

16. A shaft is supported by two bearings placed 1 m apart. A 600 mm diameter pulley is mounted at a distance of 300 mm to the right of left hand bearing and this drives a pulley directly below it with the help of belt having maximum tension of 2.25 kN. Another pulley 400 mm diameter is placed 200 mm to the left of right hand bearing and is driven with the help of electric motor and belt, which is placed horizontally to the right. The angle of contact for both the pulleys is180° and μ= 0.24. Determine the suitable diameter for a solid shaft, allowing working stress of 63 MPa in tension and 42 MPa in shear for the material of shaft. Assume that the torque on one pulley is equal to that on the other pulley. RTU-2018
17. For supporting the travelling crane in a workshop, the brackets are fixed on steel column as shown in Fig. The maximum load that comes on the bracket is 12 kN acting vertically at a distance of 400 mm from the face of the column. The vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at a distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible value of the tensile stress for bolt material is 84 MPa. Also find the cross-section of the arm of the bracket which is rectangular, take depth of arm of bracket as 250 mm.



18. Design a cast iron protective type flange coupling to transmit 15 kW at 900 from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used as shear stress for shaft, bolt and key material = 40 MPa; crushing stress for bolt and key = 80 MPa; shear stress for cast iron = 8 MPa.
19. The lead screw of a lathe has single-start ISO metric trapezoidal threads of 52 mm nominal diameter and 8 mm pitch. The screw is required to exert an axial force of 2 kN in order to drive the tool carriage during turning operation. The thrust is carried on a collar of 100 mm outer diameter and 60 mm inner diameter. The values of coefficient of friction at the screw threads and the collar are 0.15 and 0.12 respectively. The lead screw rotates at 30 rpm. Calculate **(i)** the power required to drive the lead screw; and **(ii)** the efficiency of the screw.

Details of Efforts Made to Fill Gap Between COs and POs (Expert Lecture/Workshop/Seminar/Extra Coverage in Lab etc.)

Analysis of different members using ANSYS

1. Stress and deformation analysis of cantilever beam using ANSYS
2. Stress analysis of the cotter joint subjected to 20 kN using ANSYS software.
3. Stress analysis of the knuckle joint subjected to 10 kN using ANSYS software.
4. Stress analysis of a crank hook, its supports, and bearing for a 20 kN force using ANSYS software.

Course Note: Design of machine Elements-I (5ME4-04)

Unit-I (L1): Introduction

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

Types of design: There may be several types of design such as

- Adaptive design: This is based on existing design, for example, standard products or systems adopted for a new application. Conveyor belts, control system of machines are some of the examples where existing design systems are adapted for a particular use.
- Developmental design: it starts with an existing design, finally a modified design is obtained. A new model of a car is a typical example of a developmental design.
- New design: This type of design is an entirely new one but based on existing scientific principles. No scientific invention is involved but requires creative thinking to solve a problem. Examples of this type of design may include designing a small vehicle for transportation of men and material on board a ship or in a desert. Some research activity may be necessary.

Types of design based upon the methods:

- Rational design. This is based on determining the stresses and strains of components and thereby deciding their dimensions.
- Empirical design. This type of design depends upon empirical formulae based on experience and experiments.
- Industrial design. This type of design depends upon the production aspects to manufacture any machine component in the industry.
- Computer aided design. This type of design depends upon the use of computer systems to assist in the creation, modification, analysis and optimization of a design.

General Procedure in Machine Design:

The general procedure to solve a design problem is as follows:

- Recognition of need. First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed.
- Synthesis (Mechanisms). Select the possible mechanism or group of mechanisms which will give the desired motion.
- Analysis of forces. Find the forces acting on each member of the machine and the energy transmitted by each member.
- Material selection. Select the material best suited for each member of the machine.
- Design of elements. The design of machine elements is an important step in a design process. It consists of the following steps: (i) Determine the forces acting on the component (ii) Select proper material for the component depending upon the functional requirements such as strength, rigidity, hardness and wear Resistance (iii) Determine the likely mode of failure for the component and depending upon it, select the criterion of failure, such as yield strength, ultimate tensile strength, endurance limit

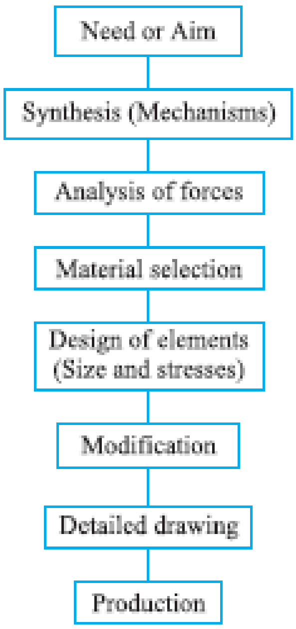


Fig.1: Design Procedure

or permissible deflection (iv) Determine the geometric dimensions of the component using a suitable factor of safety

- Modification. Modify the dimensions from assembly and manufacturing considerations. The modification may also be necessary by consideration of manufacturing to reduce overall cost.
- Detailed drawing. Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested.
- Production. The component, as per the drawing, is manufactured in the workshop. The flow chart for the general procedure in machine design is shown in Fig.1.

Basic requirements of Machine elements:

A machine consists of machine elements. Each part of a machine, which has motion with respect to some other part, is called a machine element. The machine element should satisfy the following basic requirements:

(i) Strength: A machine part should not fail under the effect of the forces that act on it. It should have sufficient strength to avoid failure either due to fracture or due to general yielding.

(ii) Rigidity: A machine component should be rigid, that is, it should not deflect or bend too much due to forces or moments that act on it. A transmission shaft in many times designed on the basis of lateral and torsional rigidities. In these cases, maximum permissible deflection and permissible angle of twist are the criteria for design.

(iii) Wear Resistance: Wear is the main reason for putting the machine part out of order. It reduces useful life of the component. Wear also leads to the loss of accuracy of machine tools. There are different types of wear such as abrasive wear, corrosive wear and pitting. Surface hardening can increase the wear resistance of the machine
components, such as gears and cams.

(iv) Minimum Dimensions and Weight: A machine part should be sufficiently strong, rigid and wear resistant
and at the same time, with minimum possible dimensions and weight. This will result in minimum material cost.

(v) Manufacturability: Manufacturability is the ease of fabrication and assembly. The shape and material of the machine part should be selected in such a way that it can be produced with minimum labor cost.

(vi) Safety: The shape and dimensions of the machine parts should ensure safety to the operator
of the machine. The designer should assume the worst possible conditions and apply 'fail-safe' or 'redundancy' principles in such cases.

(vii) Conformance to Standards: A machine part should conform to the national or international standard covering its profile, dimensions, grade and material.

(viii) Reliability: Reliability is the probability that a machine part will perform its intended functions under desired operating conditions over a specified period of time. A machine part should be reliable, that is, it should perform its function satisfactorily over its lifetime.

(ix) Maintainability: A machine part should be maintainable. Maintainability is the ease with which a machine part can be serviced or repaired.

(x) Minimum Life-cycle Cost: Life-cycle cost of the machine part is the total cost to be paid by the purchaser for purchasing the part and operating and maintaining it over its life span

Unit 2 (Part-1): L2: Materials and their properties

The machine elements should be made of such a material which has properties suitable for the conditions of operation.

Classification of Engineering Materials:

The engineering materials are mainly classified as:

Metals and their alloys, such as iron, steel, copper, aluminum, etc.

Non-metals, such as glass, rubber, plastic, etc.

The metals may be further classified as:

Ferrous metals: The ferrous metals are those which have the iron as main component, such as cast iron, wrought iron and steel.

Non-ferrous metals: The non-ferrous metals are those which have a metal other than iron as main component, such as copper, aluminum, brass, tin, zinc, etc.

Ferrous materials

Cast iron- It is an alloy of iron, carbon and silicon and it is hard and brittle. Carbon content may be more than 2%. Carbon may be present as free carbon or iron carbide Fe3C.

In general, the types of cast iron are:

(a) Grey cast iron- Carbon is mainly in the form of graphite. This type of cast iron is inexpensive and has high compressive strength. Graphite is an excellent solid lubricant and this makes it easily machined but brittle. Some examples of this type of cast iron are FG20, FG35 or FG35Si15. The numbers indicate ultimate tensile strength in MPa and 15 indicates 0.15% silicon. Grey cast iron is used for automotive components such as cylinder block, brake drum, clutch plate, cylinder and cylinder head, gears and housing of gear box, flywheel and machine frame, bed and guide.

(b) White cast iron- In these cast irons, carbon is present in the form of iron carbide (Fe3C) which is hard and brittle. The presence of iron carbide increases hardness and makes it difficult to machine. Consequently, these cast irons are abrasion resistant.

(c) Malleable cast iron- Malleable cast iron is first cast as white cast iron and then converted into malleable cast iron by heat treatment. In malleable cast iron, the carbon is present in the form of irregularly shaped nodules of graphite called 'temper' carbon. These are tougher than grey cast iron and they can be twisted or bent without fracture. They have excellent machining properties and are inexpensive. Malleable cast iron is used for making parts where forging is expensive such as hubs for wagon wheels, brake supports.

There are three basic types of malleable cast iron—blackheart, pearlite and white heart—which are designated by symbols BM, PM and WM, respectively and followed by
minimum tensile strength in N/mm^2. For example, BM 350 is blackheart malleable cast iron with a minimum tensile strength of 350 N/mm^2; PM 600 is pearlite malleable cast iron with a minimum tensile strength of 600 N/mm^2; and WM 400 is white heart malleable cast iron with a minimum tensile strength of 400 N/mm^2.

Blackheart malleable cast iron has excellent castability and machinability. It is used for brake shoes, pedal, levers, wheel hub, axle housing and door hinges.

Whiteheart malleable cast iron is particularly suitable for the manufacture of thin castings which require ductility. It is used for pipe fittings, switch gear equipment, fittings for bicycles and motorcycle frames.

Pearlitic malleable iron castings can be hardened by heat treatment. It is used for general engineering components with specific dimensional tolerances.

Ductile cast iron or spheroidal or nodular graphite cast iron: Ductile cast iron is also called nodular cast iron or spheroidal graphite cast iron. In ductile cast iron, carbon is present in the form of spherical nodules called 'spherules' or 'globules' in a relatively ductile matrix. When a component of ductile cast iron is broken; the fractured surface has a bright appearance like steel. Ductile cast iron is designated by the symbol SG (spheroidal graphite) followed by the minimum tensile strength in N/mm^2 and minimum elongation in per cent. For example, SG 800/2 is spheroidal graphite cast iron with a minimum tensile strength of 800 N/mm^2 and a minimum elongation of 2%. Ductile cast iron is used for crankshaft, heavy duty gears and automobile door hinges. Ductile cast iron combines the processing advantages of grey cast iron with the engineering properties of steel. Spheroidal graphite cast iron is dimensionally stable at high temperatures and, therefore, used for furnace doors, furnace components and steam plants.

Wrought iron- This is a very pure iron where the iron content is of the order of 99.5%. It is produced by re-melting pig iron and contains some small amount of silicon, Sulphur, or phosphorus. The wrought iron is a tough, malleable and ductile material. It cannot stand sudden and excessive shocks. It can be easily forged or welded. It is used for chains, crane hooks, railway couplings, water and steam pipes.

Steel- It is an alloy of iron and carbon, with carbon content up to a maximum of 1.5%. The carbon occurs in the form of iron carbide, because of its ability to increase the hardness and strength of the steel. Other elements e.g. silicon, Sulphur, phosphorus and manganese are also present to greater or lesser amount to impart certain desired properties to it.

Two main categories of steel are (a) Plain carbon steel and (b) alloy steel.

(a) Plain carbon steel- The properties of plain carbon steel depend mainly on the carbon percentages and other alloying elements are not usually present in more than 0.5 to 1% such as 0.5% Si or 1% Mn etc. There is a large variety of plane carbon steel and they are designated as C01, C14, C45, C70 and so on where the number indicates the carbon percentage.

Depending upon the percentage of carbon, plain carbon steels are classified into the following three groups: (i) Low Carbon Steel Low carbon steel contains less than 0.3% carbon. It is popular as 'mild steel'. Low carbon steels are soft and very ductile. They can be easily machined and easily welded. However, due to low carbon content, they are unresponsive to heat treatment.

(ii) Medium Carbon Steel Medium carbon steel has a carbon content in the range of 0.3% to 0.8%. It is popular as machinery steel. Medium carbon steel is easily hardened by heat treatment. Medium carbon steels are stronger and tougher as compared with low carbon steels. They can be machined well and they respond readily to heat treatment.

(iii) High Carbon Steel High carbon steel contains more than 0.8% carbon. They are called hard steels or tool steels. High carbon steels respond readily to heat treatments. When heat treated, high carbon steels have very high strength combined with hardness. They do not have much ductility as compared with low and medium carbon steels. High carbon steels are difficult to weld.

(b) Alloy steel- these are steels in which elements other than carbon are added in sufficient quantities to impart desired properties, such as wear resistance, corrosion resistance, electric or magnetic properties. Chief alloying elements added are usually as:

nickel for strength and toughness, chromium for hardness and strength, tungsten for hardness at elevated temperature, vanadium for tensile strength, manganese for high strength in hot rolled and heat treated condition, silicon for high elastic limit, cobalt for hardness and molybdenum for extra tensile strength. Some examples of alloy steels are 35Ni1Cr60, 30Ni4Cr1, 40Cr1Mo28, 37Mn2. Stainless steel is one such alloy steel that gives good corrosion resistance. One important type of stainless steel is often described as 18/8 steel where chromium and nickel percentages are 18 and 8 respectively.

Free cutting steels: Steels of this group include carbon steel and carbon–manganese steel with a small percentage of Sulphur. Due to addition of Sulphur, the machinability of these steels is improved.

Machinability is defined as the ease with which a component can be machined. It involves three factors—the ease of chip formation, the ability to achieve a good surface finish and ability to achieve an economical tool life. Machinability is an important consideration for parts made by automatic machine tools. Typical applications of free cutting steels are studs, bolts and nuts.

BIS or IS system of designation of materials:

Steels Designated on the Basis of Mechanical Properties: According to Indian standard **IS: 1570 (Part–I)-1978 (Reaffirmed 1993), these steels are designated by a symbol ‘Fe’ or ‘Fe E’ depending on whether the steel has been specified on the basis of minimum tensile strength or yield strength, followed by the figure indicating the minimum tensile strength or yield stress in N/mm^2. For example, ‘Fe 290’ means a steel having minimum tensile strength of 290 N/mm^2 and ‘Fe E 220’ means a steel having yield strength of 220 N/mm^2.

Steels Designated on the Basis of Chemical Composition: According to Indian standard, IS: 1570 (Part II/Sec I)-1979 (Reaffirmed 1991), The designation of plain carbon steel consists of the following three quantities:

(i) a figure indicating 100 times the average percentage of carbon; (ii) a letter C; and (iii) a figure indicating 10 times the average percentage of manganese.

As an example, 55C4 indicates a plain carbon steel with 0.55% carbon and 0.4% manganese. A steel with 0.35–0.45% carbon and 0.7–0.9% manganese is designated as 40C8.

The designation of unalloyed free cutting steels: it consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon; (ii) a letter C; (iii) a figure indicating 10 times the average percentage of manganese; (iv) a symbol ‘S’, ‘Se’, ‘Te’ or ‘Pb’ depending

upon the element that is present and which makes the steel free cutting; and (v) a figure indicating 100 times the average percentage of the above element that makes the steel free cutting.

As an example, 25C12S14 indicates a free cutting steel with 0.25% carbon, 1.2% manganese and 0.14% Sulphur. Similarly, a free cutting steel with an average of 0.20% carbon, 1.2% manganese and 0.15% lead is designated as 20C12Pb15.

Indian Standard Designation of Low and Medium Alloy Steels:

The term ‘alloy’ steel is used for low and medium alloy steels containing total alloying elements not exceeding 10%. The designation of alloy steels consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon; and (ii) chemical symbols for alloying elements each followed by the figure for its average percentage content multiplied by a factor.

The multiplying/Division factor depends upon the upon the alloying element. The values of this factor are as follows:

Element	*Multiplying factor*
Cr, Co, Ni, Mn, Si and W	4
Al, Be, V, Pb, Cu, Nb, Ti, Ta, Zr and Mo	10
P, S and N	100

In alloy steels, ‘Mn and silicon (Si)’ is included only if the content of manganese is equal to or greater than 1%. The chemical symbols and their figures are arranged in descending order of their percentage content. As an example, 25Cr4Mo2 is an alloy steel having average 0.25% of carbon, 1% chromium and 0.2% molybdenum. Similarly, 40Ni8Cr8V2 is an alloy steel containing average 0.4% of carbon, 2% nickel, 2% chromium and 0.2% vanadium.

Consider an alloy steel with the following composition:

carbon = 0.12–0.18%

silicon = 0.15–0.35%

manganese = 0.40–0.60%
chromium = 0.50–0.80%
The average percentage of carbon is 0.15%, which is denoted by the number (0.15×100) or 15. The percentage content of silicon and manganese is negligible and, as such, they are deleted from the designation. The significant element is chromium and its average percentage is 0.65. The multiplying factor for chromium is 4 and (0.65×4) is 2.6, which is rounded to 3. Therefore, the complete designation of steel is 15Cr3. As a second example, consider a steel with the following chemical composition:
carbon = 0.12–0.20%
silicon = 0.15–0.35%
manganese = 0.60–1.00%
nickel = 0.60–1.00%
chromium = 0.40–0.80%
The average percentage of carbon is 0.16% and multiplying this value by 100, the first figure in the designation of steel is 16. The average percentage of silicon and manganese is very small and, as such, the symbols Si and Mn are deleted. Average percentages of nickel and chromium are 0.8 and 0.6, respectively, and the multiplying factor for both elements is 4. Therefore, nickel: 0.8 × 4 = 3.2 rounded to 3 or Ni3 chromium: 0.6×4 = 2.4 rounded to 2 or Cr2. The complete designation of steel is 16Ni3Cr2.

Indian Standard Designation of high Alloy Steels:

The term 'high alloy steels' is used for alloy steels containing more than 10% of alloying elements. The designation of high alloy steels consists of the following quantities:
(i) a letter 'X';
(ii) a figure indicating 100 times the average percentage of carbon;
(iii) chemical symbol for alloying elements each followed by the figure for its average percentage content rounded off to the nearest integer; and
(iv) chemical symbol to indicate a specially added element to attain desired properties, if any.
As an example, X15Cr25Ni12 is a high alloy steel with 0.15% carbon, 25% chromium and 12% nickel. As a second example, consider a steel with the following chemical composition:
carbon = 0.15–0.25%
silicon = 0.10–0.50%
manganese = 0.30–0.50%
nickel = 1.5–2.5%
chromium = 16–20%
The average content of carbon is 0.20%, which is denoted by a number (0.20×100) or 20. The major alloying elements are chromium (average 18%) and nickel (average 2%). Hence, the designation of steel is X20Cr18Ni2.

Selection of material:

Selection of a proper material for the machine component is one of the most important steps in the process of machine design. The best material is one which will serve the desired purpose at minimum cost. The factors which should be considered while selecting the material for a machine component are as follows:
(i) Availability; The material should be readily available in the market, in large enough quantities to meet the requirement. Cast iron and aluminum alloys are always available in abundance while shortage of lead and copper alloys is a common experience.
(ii) Cost: For every application, there is a limiting cost beyond which the designer cannot go. When this limit is exceeded; the designer has to consider other alternative materials. In cost analysis, there

are two factors, namely, cost of material and the cost of processing the material into finished goods. It is likely that the cost of material might be low, but the processing may involve costly manufacturing operations.

(iii) Mechanical Properties: Mechanical properties are the most important technical factor governing the selection of material. They include strength under static and fluctuating loads, elasticity, plasticity, stiffness, resilience, toughness, ductility, malleability and hardness. Depending upon the service conditions and the functional requirement different mechanical properties are considered and a suitable material is selected. For example, the material for the connecting rod of an internal combustion engine should be capable to withstand fluctuating stresses induced due to combustion of fuel. In this case, endurance limit becomes the criterion of selection. The piston rings should have a hard surface to resist wear due to rubbing action with the cylinder surface, and surface hardness is the selection criterion. In case of bearing materials, a low coefficient of friction is desirable while clutch or brake lining requires a high coefficient of friction. The material for automobile bodies and hoods should have the ability to be extensively deformed in plastic range without fracture, and plasticity is the criterion of material selection.

(iv) Manufacturing Considerations; In some applications, machinability of material is an important consideration in selection. Sometimes, an expensive material is more economical than a low priced one, which is difficult to machine. Free cutting steels have excellent machinability, which is an important factor in their selection for high strength bolts, axles and shafts. Where the product is of complex shape, castability or ability of the molten metal to flow into intricate passages is the criterion of material selection. In fabricated assemblies of plates and rods, weldability becomes the governing factor. The manufacturing processes, such as casting, forging, extrusion, welding and machining governs the selection of material.

Properties of Materials:

The important properties, which determine the utility of the material, are physical, chemical and mechanical properties. Physical and mechanical properties are important for design.

Physical Properties of Metals

The physical properties of the metals include luster, color, size and shape, density, electric and thermal conductivity, and melting point.

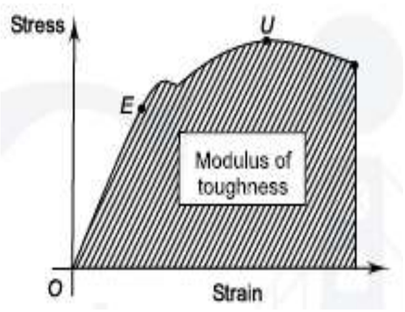
Mechanical Properties of Metals

The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load. Mechanical properties are given as:

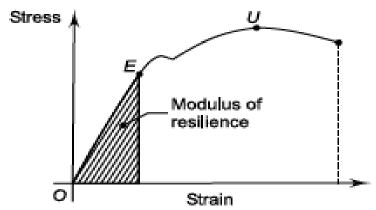
- Strength. It is the ability of a material to resist the externally applied forces without breaking or yielding. The internal resistance offered by a part to an externally applied force is called stress.
- Stiffness or rigidity: It is the ability of the material to resist deformation under the action of an external load. The modulus of elasticity is the measure of stiffness.
- Elasticity. It is the property of a material to regain its original shape after deformation when the external forces are removed. This property is desirable for materials used in tools and machines. It may be noted that steel is more elastic than rubber.
- Plasticity. It is property of a material which retains the deformation produced under load permanently. This property of the material is necessary for forgings, in stamping images on coins and in ornamental work.
- Ductility. It is the property of a material enabling it to be drawn into wire with the application of a tensile force. A ductile material must be both strong and plastic. The ductility is usually measured by the terms, percentage elongation and percentage reduction in area. The ductile

material commonly used in engineering practice (in order of diminishing ductility) are mild steel, copper, aluminum, nickel, zinc, tin and lead.

- Brittleness. It is the property of a material which shows negligible plastic deformation before fracture takes place. Brittleness is the opposite to ductility. In ductile materials, failure takes place by yielding. Brittle components fail by sudden fracture. A tensile strain of 5% at fracture in a tension test is considered as the dividing line between ductile and brittle materials. Cast iron is a brittle material.
- Malleability. It is a special case of ductility which permits materials to be rolled or hammered into thin sheets. A malleable material should be plastic but it is not essential to be so strong. The malleable materials commonly used in engineering practice (in order of diminishing malleability) are lead, soft steel, wrought iron, copper and aluminum.
- Toughness. It is the ability of the material to absorb energy before fracture takes place. This property is essential for machine components which are required to withstand impact loads. Tough materials have the ability to bend, twist or stretch before failure takes place. All structural steels are tough materials. Toughness is measured by a quantity called modulus of toughness. Modulus of toughness is the total area under stress–strain curve in a tension test, which also represents the work done to fracture the specimen. In practice, toughness is measured by the Izod and Charpy impact testing machines.



- Machinability. It is the property of a material which refers to a relative case with which a material can be cut. The machinability of a material can be measured in a number of ways such as comparing the tool life for cutting different materials or thrust required to remove the material at some given rate or the energy required to remove a unit volume of the material. It may be noted that brass can be easily machined than steel.
- Resilience. It is the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded. Resilience is measured by a quantity, called modulus of resilience, which is the strain energy per unit volume. It is represented by the area under the stress–strain curve from the origin to the elastic limit point. This property is essential for spring materials.

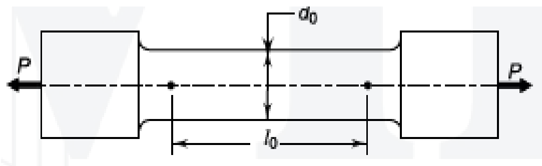


- Creep. When a part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called creep. This property is considered in designing internal combustion engines, boilers and turbines.
- Fatigue. When a material is subjected to fluctuating stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as fatigue. The failure is caused by means of a progressive crack formation which are usually fine and of microscopic size. This property is considered in designing shafts, connecting rods, springs, gears, etc.
- Hardness. It is defined as the resistance of the material to penetration or permanent deformation. It usually indicates resistance to abrasion, scratching, cutting or shaping. The hardness of a metal may be determined by the following tests:

(a) Brinell hardness test,
(b) Rockwell hardness test,
(c) Vickers hardness (also called Diamond Pyramid) test, and
(d) Shore scleroscope.

Stress–strain diagrams:

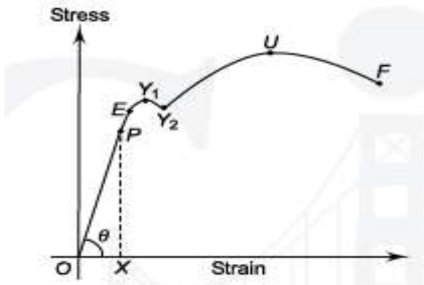
A stress–strain diagram for a material gives the relationship between stress and strain. It is obtained by making a tension test. A tension test is one of the simplest and basic tests and determines values of number of parameters concerned with mechanical properties of materials like strength, ductility and toughness. The specimen used in a tension test is illustrated in Fig. The shape and dimensions of this specimen are standardized. They should conform to IS 1608 : 19721. The cross-section of the specimen can be circular, square or rectangular. The standard gauge length l_o is given by,



$$l_0 = 5.65\sqrt{A_0},$$

where A_o is the cross-sectional area of the specimen. For circular cross-section, $l_0 = 5d_0$

In a tension test, the specimen is subjected to axial tension force, which is gradually increased and the corresponding deformation is measured. Initially, the gauge length is marked on the specimen and initial dimensions d_o and l_o are measured before starting the test. The specimen is then mounted on the machine and gripped in the jaws. It is then subjected to an axial tension force, which is increased by suitable increments. After each increment, the amount by which the gauge length l_o increases, i.e., deformation of gauge length, is measured by an extensometer. The procedure of measuring the tension force and corresponding deformation is continued till fracture occurs and the specimen is broken into two pieces. The tensile force divided by the original cross-sectional area of the specimen gives stress, while the deformation divided by gauge length gives the strain in the specimen. Therefore, the results of a tension test are expressed by means of stress–strain relationship and plotted in the form of a graph. A typical stress–strain diagram for ductile materials like mild steel is shown in Fig. The following properties of a material can be obtained from this diagram:



(i) Proportional Limit: It is observed from the diagram that stress–strain relationship is linear from the point O to P. OP is a straight line and after the point P, the curve begins to deviate from the straight line. Hooke's law states that stress is directly proportional to strain. Therefore, it is applicable only up to the point P. The term proportional limit is defined as the stress at which the stress–strain curve begins to deviate from the straight line. Point P indicates the proportional limit.

(ii) Modulus of Elasticity: The modulus of elasticity or Young's modulus (E) is the ratio of stress to strain up to the point P. It is given by the slope of the line OP. Therefore,

$$E = \tan\theta = \frac{PX}{OX} = \frac{\text{stress}}{\text{strain}}$$

(iii) Elastic Limit; Even if the specimen is stressed beyond the point P and up to the point E, it will regain its initial size and shape when the load is removed. This indicates that the material is in elastic stage up to the point E. Therefore, E is called the elastic limit. The elastic limit of the material is defined as the maximum stress without any permanent deformation. The proportional limit and elastic limit are very close to each other, and it is difficult to distinguish between points P and E on the stress–strain diagram. In practice, many times, these two limits are taken to be equal.

(iv) Yield Strength: When the specimen is stressed beyond the point E, plastic deformation occurs and

the material starts yielding. During this stage, it is not possible to recover the initial size and shape of the specimen on the removal of the load. It is seen from the diagram that beyond the point E, the strain increases at a faster rate up to the point Y_1. In other words, there is an appreciable increase in strain without much increase in stress. In case of mild steel, it is observed that there is a small reduction in load and the curve drops down to the point Y_2 immediately after yielding starts. The points Y_1 and Y_2 are called the upper and lower yield points, respectively. For many materials, the points Y_1 and Y_2 are very close to each other and in such cases, the two points are considered as same and denoted by Y. The stress corresponding to the yield point Y is called the yield strength. The yield strength is defined as the maximum stress at which a marked increase in elongation occurs without increase in the load.

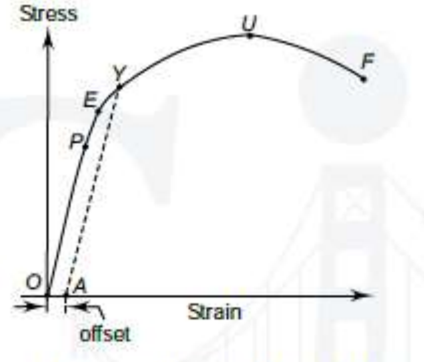


Fig. *Yield Stress by Offset Method*

Many varieties of steel, especially heat-treated steels and cold-drawn steels, do not have a well-defined yield point on the stress–strain diagram. As shown in Fig. For such materials, which do not exhibit a well-defined yield point, the yield strength is defined as the stress corresponding to a permanent set of 0.2% of gauge length. In such cases, the yield strength is determined by the offset method. A distance OA equal to 0.002 mm/mm strain (corresponding to 0.2% of gauge length) is marked on the X-axis. A line is constructed from the point A parallel to the straight line portion OP of the stress–strain curve. The point of intersection of this line and the stress–strain curve is called Y or the yield point and the corresponding stress is called 0.2% yield strength.

The terms proof load or proof strength are frequently used in the design of fasteners. Proof strength is similar to yield strength. It is determined by the offset method; however, the offset in this case is 0.001 mm/mm corresponding to a permanent set of 0.1% of gauge length. 0.1% Proof strength, denoted by symbol Rp0.1, is defined as the stress which will produce a permanent extension of 0.1% in the gauge length of the test specimen. The proof load is the force corresponding to proof stress.

(v) Ultimate Tensile Strength: After the yield point Y_2, plastic deformation of the specimen increases. The material becomes stronger due to strain hardening, and higher and higher load is required to deform the material. Finally, the load and corresponding stress reach a maximum value, as given by the point U. The stress corresponding to the point U is called the ultimate strength. The ultimate tensile

strength is the maximum stress that can be reached in the tension test. For ductile materials, the diameter of the specimen begins to decrease rapidly beyond the maximum load point U. There is a localized reduction in the cross-sectional area, called necking.

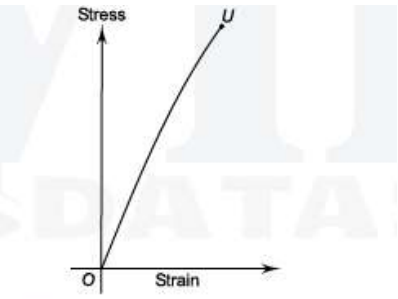


Fig. *Stress-Strain Diagram of Brittle Materials*

As the test progresses, the cross-sectional area at the neck decreases rapidly and fracture takes place at the narrowest cross-section of the neck. This fracture is shown by the point F on the diagram. The stress at the time of fracture is called breaking strength. It is observed from the stress–strain diagram that there is a downward trend after the maximum stress has been reached. The breaking strength is slightly lower than the ultimate tensile strength. The stress–strain diagram for brittle materials like cast iron is shown in Fig. It is observed that such materials do not exhibit the yield point. The deviation of the stress–strain curve from straight line begins very early and fracture occurs suddenly at the point U with very small plastic deformation and without necking. Therefore, ultimate tensile strength is considered as failure criterion in brittle materials.

(vi) Percentage Elongation: After the fracture, the two halves of the broken test specimen are fitted together as shown in Fig.(b) and the extended gauge length l is measured. The percentage elongation is defined as the ratio of the increase in the length of the gauge section of the specimen to original gauge length, expressed in per cent. Therefore,

$$\text{percentage elongation} = \left(\frac{l - l_0}{l_0}\right) \times 100$$

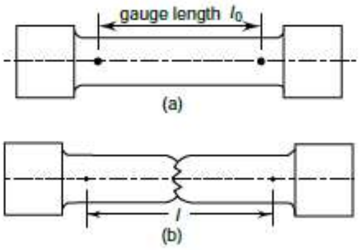


Fig. *Determination of Percentage Elongation: (a) Original Test Piece (b) Broken Test Piece*

Ductility is measured by percentage elongation. If percentage elongation exceeds 15% the material is ductile and if it is less than 5% the material is brittle.

(vii) Percentage Reduction in Area: Percentage reduction in area is defined as the ratio of decrease in cross-sectional area of the specimen after fracture to the original cross-sectional area, expressed in per cent. Therefore,

$$\text{percentage reduction in area} = \left(\frac{A_0 - A}{A_0}\right) \times 100$$

where, A_o = original cross-sectional area of the test specimen: A = final cross-sectional area after fracture. Percentage reduction in area, like percentage elongation, is a measure of the ductility of the material.

Unit 2 (Part-II) Manufacturing Considerations in Design

Manufacturing:

Definition:

It is the processing of raw materials into finished goods through the use of tools and processes

The manufacturing processes can be broadly classified into the following three categories:

Casting Processes:

In these processes, molten metals such as cast iron, copper, aluminum or non-metals like plastic are poured into the mould and solidified into the desired shape. e.g., housing of gear box, flywheel with rim and spokes, machine tool beds and guides.

Deformation Processes

In these processes, a metal, either hot or cold, is plastically deformed into the desired shape.
Forging, rolling, extrusion, press working are the examples of deformation processes.
The products include connecting rods, crankshafts, I-section beams, car bodies and springs.

Material Removal or Cutting Processes

In these processes, the material is removed by means of sharp cutting tools.
Turning, milling, drilling, shaping, planning, grinding, shaving and lapping are the examples of material removal processes. The products include transmission shafts, keys, bolts and nuts.
In addition, there are joining processes like bolting, welding and riveting. They are essential for the assembly of the product.
The optimum manufacturing method is selected by considering the following factors:

- Material of the component
- Cost of manufacture
- Geometric shape of the component
- Surface finish and tolerances required
- Volume of production

One of the easiest methods to convert the raw material into the finished component is *casting*.
There are several casting processes such as sand casting, shell-mould casting, permanent mould casting, die casting, centrifugal casting or investment casting.
The advantages of sand casting process as a manufacturing method are as follows:

- The tooling required for casting process is relatively simple and inexpensive. Therefore, Sand casting is one of the cheapest methods of manufacturing.
- Almost any metal such as cast iron, aluminium, brass or bronze can be cast by this method.
- Any component, even with a complex shape, can be cast.
- There is no limit on the size of the component. Even large components can be cast.

The disadvantages of the sand casting process are as follows:

- It is not possible to achieve close tolerances for cast components. Therefore, cast components require additional machining and finishing, which increases cost.
- Cast components have a rough surface finish.
- Long and thin sections or projections are not possible for cast components.

Design Considerations of Castings

In designing a casting, the following points should always be considered.

- Always keep the stressed areas of the part in compression
- Cast iron has more compressive strength than its tensile strength.
- The castings should be placed in such a way that they are subjected to compressive rather than tensile stresses.

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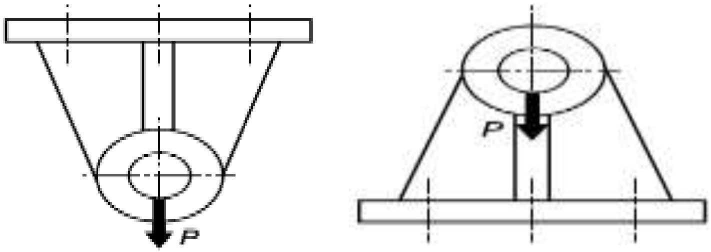
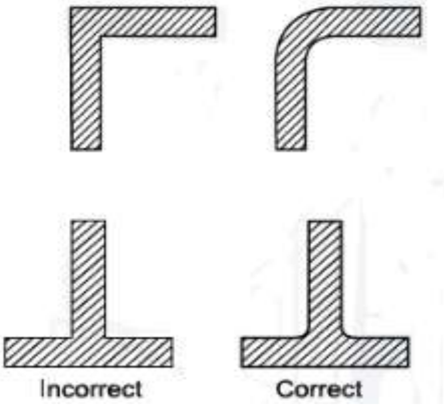
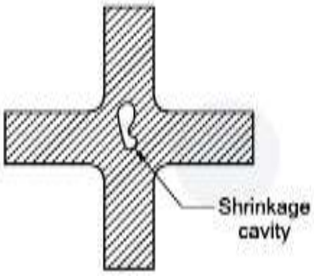


2. Round all external corners
- It has two advantages—it increases the endurance limit of the component and reduces the formation of brittle chilled edges.
- When the metal in the corner cools faster than the metal adjacent to the corner, brittle chilled edges are formed due to iron carbide.

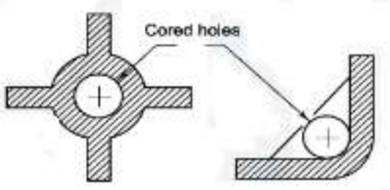


- Abrupt changes in the cross-section result in high stress concentration, so the section thickness throughout should be held as uniform.
- If the thickness is to be varied at all, the change should be gradual

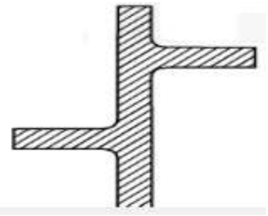
3. Avoid concentration of metal at the junction
- Even after the metal on the surface solidifies, the central portion still remains in the molten stage, with the result that a shrinkage cavity or blowhole may appear at the center.



There are two ways to avoid the concentration of metal.
One is to provide a cored opening in webs and ribs,



One can stagger the ribs and webs.



4. Avoid very thin sections
5. Shot blast the parts wherever possible
   - The shot blasting process improves the endurance limit of the component, particularly in case of thin sections.
6. In designing a casting, the various allowances must be provided in making a pattern.

Forging

- forging is a one of the important deformation processes.
- In forging, the metal in the plastic stage, rather than in the molten stage, is forced to flow into the desired shape.
- There are a number of forging processes such as hand forging, drop forging, press forging or upset forging.
- The **smith** or **hand forging** is done by means of hand tools and it is usually employed for small jobs.
- The **drop forging** is carried out with the help of drop hammers and is particularly suitable for mass production of identical parts
- The drop forging method accounts for more than 80% of the forged components.

The advantages of forging

- The fibre lines of a forged component can be arranged in a predetermined way to suit the direction of external forces that will act on the component when in service.
- Therefore, forged components have inherent strength

and toughness.

- They are ideally suitable for applications like connecting rods and crankshafts.
- In forging, there is relatively good utilization of material compared with machining.
- Forged components can be provided with thin sections, without reducing the strength. This results in lightweight construction.
- The forging process has a rapid production rate and good. The tolerances of forged parts can be held between close limits, which reduce the volume of material removal during the final finishing stages.
- reproducibility.

Disadvantage of forging

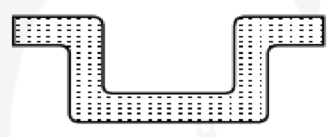
- Forging is a costly manufacturing method. The equipment and tooling required for forging is costly.
- Forging becomes economical only when the parts are manufactured on a large scale.

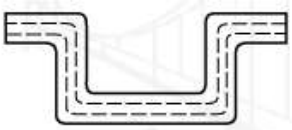
Design considerations of forgings

- Forged components are widely used in automotive and aircraft industries.
- They are usually made of steels and non-ferrous metals.

The general principles for the design of forged components are as follows:

- While designing a forging, the profile is selected in such a way that fibre lines are parallel to tensile forces and perpendicular to shear forces.





- Machining that cuts deep into the forging should be avoided, otherwise the fibre lines are broken and the part becomes weak.
- The forged component should be provided with an adequate draft.
- The parting line should be in one plane as far as possible and it should divide the forging into two equal parts.
- The forging should be provided with adequate fillet and corner radii.
- Thin sections and ribs should be avoided in forged components.

Metal Cutting Processes

Material removal or cutting processes are most common manufacturing methods.
Metal removal processes are broadly divided into three categories—metal cutting processes, grinding processes and unconventional machining processes.

Advantages

- Almost any metal can be machined.
- It is possible to achieve close tolerances for machined components.
- Machined components have a good surface finish.

The disadvantages of machining processes

Machining processes are costly and the rate of production is low compared with casting or forging.
It is not possible to machine thin sections or projections.
There is wastage of material during material removal process.

Design considerations of machined parts

The general principles for the design of machined parts are as follows:

- Avoid Machining
- Specify Liberal Tolerances
- Avoid Sharp Corners
- Use Stock Dimensions
- Design Rigid Parts
- Avoid Shoulders and Undercuts
- Avoid Hard Materials

Hot and cold working of metals

Metal deformation processes that are carried out above the recrystallization temperature (the temperature at which new stress-free grains are formed in the metal is called the recrystallization temperature) are called hot working processes.
Hot rolling, hot forging, hot spinning, hot extrusion, and hot drawing are hot working processes.
Metal deformation processes that are carried out below the recrystallization temperature are called cold working processes.
Cold rolling, cold forging, cold spinning, cold extrusion, and cold drawing are cold working processes.

Advantages of hot working

(i) Hot working reduces strain hardening.
(ii) Hot rolled components have higher toughness and ductility. They have better resistance to shocks and vibrations.

(iii) Hot working increases the strength of metal by refining the grain structure and aligning the grain of the metal with the final counters of the part. This is particularly true of forged parts.
(iv) Hot working reduces residual stresses in the component.

Disadvantages of hot working

(i) Hot working results in rapid oxidation of the surface due to high temperature.
(ii) Hot rolled components have poor surface finish than cold rolled parts.
(iii) Hot working requires expensive tools.

Advantages of cold working

(i) Cold rolled components have higher hardness and strength.
(ii) Cold worked components have better surface finish than hot rolled parts.
(iii) The dimensions of cold rolled parts are very accurate.
(iv) The tooling required for cold working is comparatively inexpensive.

Disadvantages of cold working

(i) Cold working reduces toughness and ductility. Such components have poor resistance to shocks and vibrations.
(ii) Cold working induces residual stresses in the component. Proper heat treatment is required to relieve these stresses.

Design considerations of welded assemblies

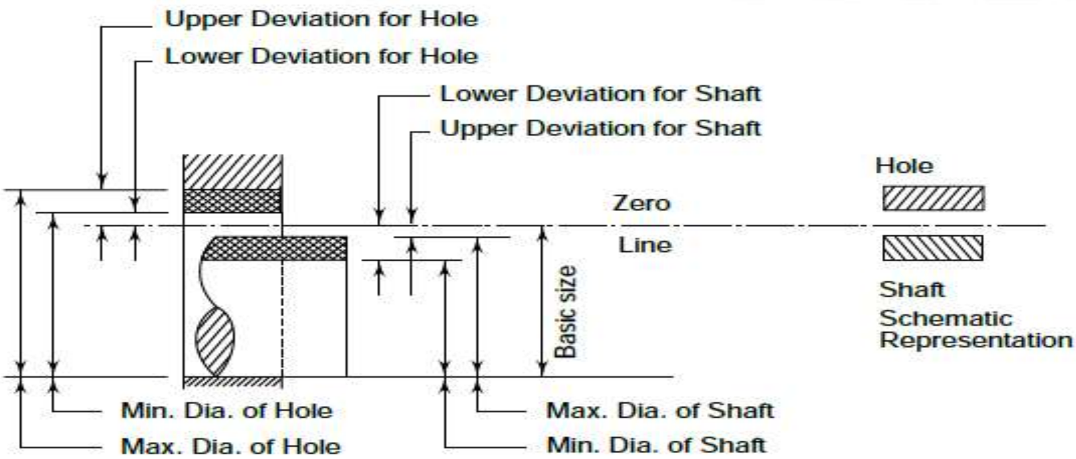
General principles in design of welded assemblies are as

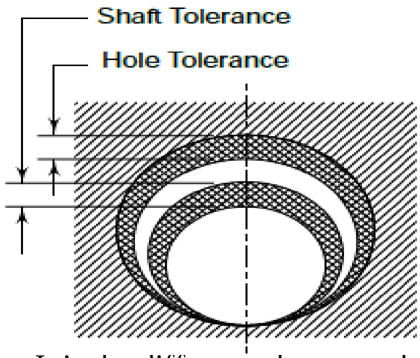
- Select the Material with High Weldability
- Use Minimum Number of Welds
- Do not Shape the Parts Based on Casting or Forging
- Use Standard Components
- Avoid Straps, Laps and Stiffeners
- Select Proper Location for the Weld
- Prescribe Correct Sequence of Welding

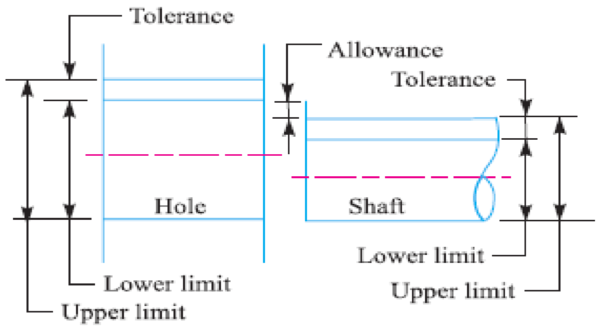
Tolerances

- Due to the inaccuracy of manufacturing methods, it is not possible to machine a component to a given dimension.
- The components are so manufactured that their dimensions lie between two limits—maximum and minimum.
- The basic dimension is called the *normal* or *basic size*, while the difference between the two limits is called *permissible tolerance*.
- *Tolerance is defined as permissible variation in the dimensions of the component*.
- The two limits are sometimes called the *upper* and *lower deviations*.

Nomenclature







Allowance. It is the difference between the basic dimensions of the mating parts.

- The allowance may be ***positive*** or ***negative***.
- When the shaft size is less than the hole size, then the allowance is ***positive***
- when the shaft size is greater than the hole size, then the allowance is ***negative*.**

Example

Shaft has to be manufactured to a diameter of 40 ± 0.02 mm. The shaft has a basic size of 40 mm. It will be acceptable if its diameter lies between the limits of sizes.

Upper limit of 40+0.02 = 40.02 mm

Lower limit of 40-0.02 = 39.98 mm.

Then, permissive tolerance is equal to 40.02 − 39.98 = 0.04 mm.

Classification of Tolerance

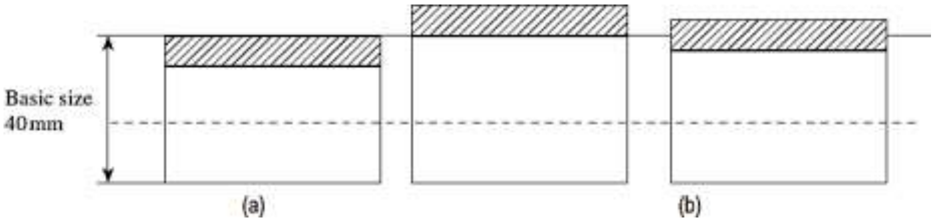
- Unilateral tolerance
- Bilateral tolerance

Unilateral tolerance

- When the tolerance distribution is only on one side of the basic size. Either positive or negative, but not both.

Bilateral tolerance

- When the tolerance distribution lies on either side of the basic size.



- Unilateral b. Bilateral Tolerances

MAXIMUM AND MINIMUM METAL CONDITIONS

Consider a shaft having a dimension of 40 ± 0.05 mm and Hole having a dimension of 45 ± 0.05 mm.

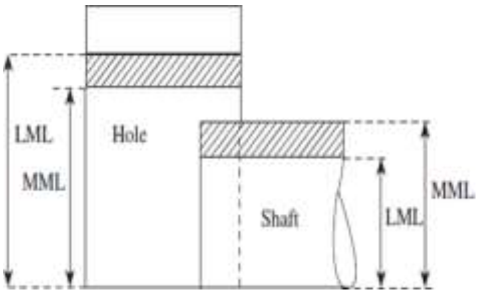
For Shaft

Maximum metal limit (MML) = 40.05 mm

Least metal limit (LML) = 39.95 mm

For Hole

Maximum metal limit (MML) = 44.95 mm Least metal limit (LML) = 45.05 mm

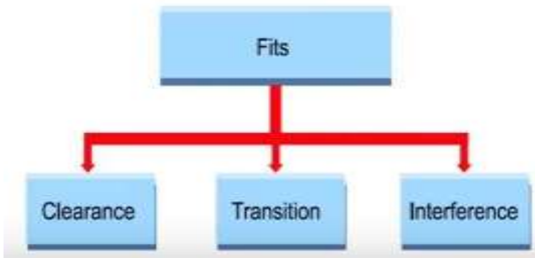


FITS

- The degree of tightness and or looseness between the two mating parts**.**

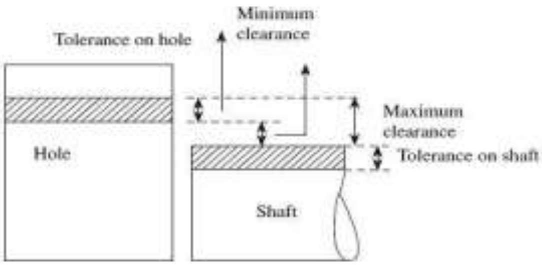
Three basic types of fits can be identified, depending on the actual limits of the hole or shaft.

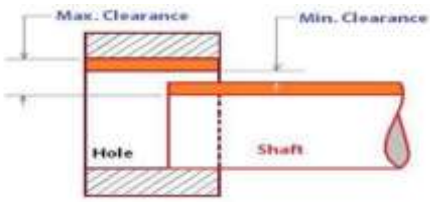
- Clearance fit
- Interference fit
- Transition fit



Clearance fit

The tolerance zone of the hole is entirely above that of the shaft

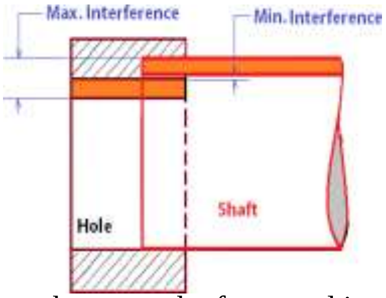




The largest permissible dia. of the shaft is smaller than the dia. of the smallest hole. E.g.: Shaft rotating in a bush

Interference fit

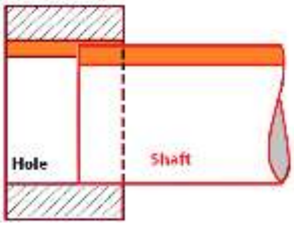
The tolerance zone of the hole is completely below that of the shaft.



- No gap between the faces and intersecting of material will occur.
- Shaft need additional force to fit into the hole.

Transition fit

The tolerance zones of the hole and the shaft overlap



- Neither loose nor tight like clearance fit and interference fit.
- Tolerance zones of the shaft and the hole will be overlapped between the interference and clearance fits.

Hole Basis and Shaft Basis Systems

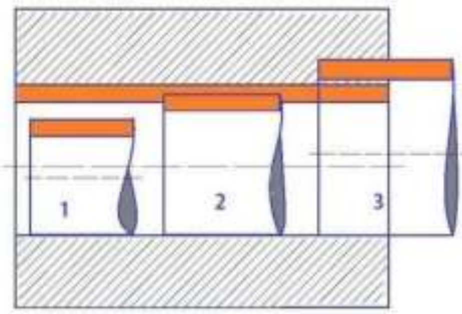
- To obtain the desired class of fits, either the size of the hole or the size of the shaft must vary.

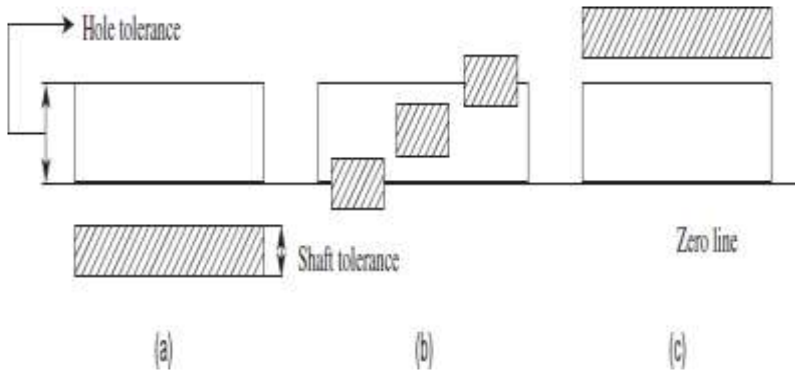
Two types of systems are used to represent three basic types of fits, clearance, interference, and transition fits.

- Hole basis system
- Shaft basis system.

Hole Basis systems

- The size of the hole is kept constant and the shaft size is varied to give various types of fits.
- Lower deviation of the hole is zero, i.e. the lower limit of the hole is same as the basic size.
- Two limits of the shaft and the higher dimension of the hole are varied to obtain the desired type of fit.





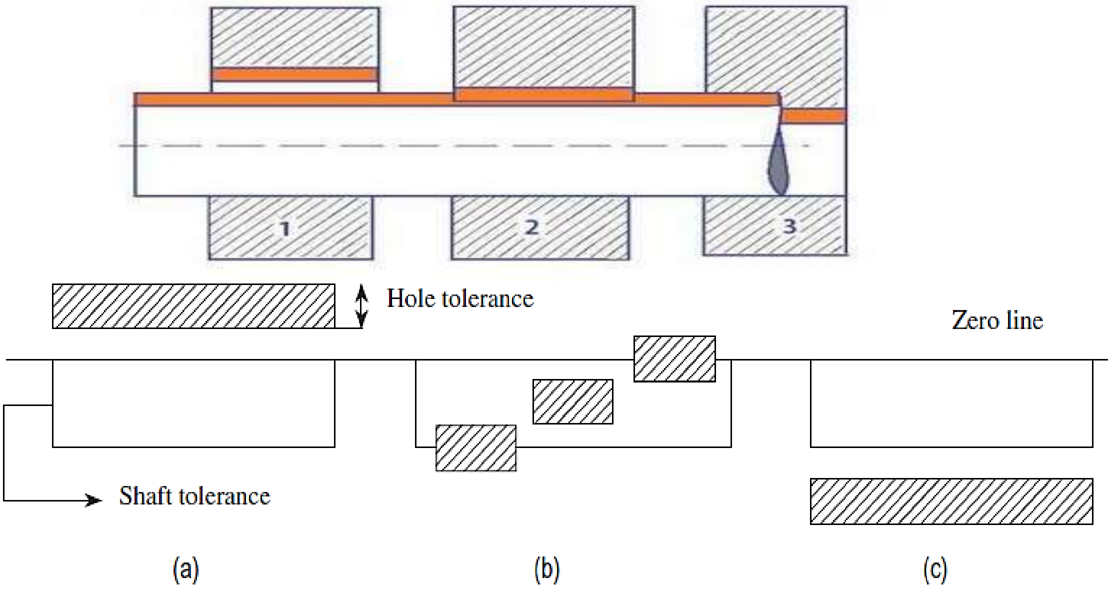
(a) Clearance fit (b) Transition fit (c) Interference fit

This system is widely adopted in industries, easier to manufacture shafts of varying sizes to the required tolerances. Standard-size plug gauges are used to check hole sizes accurately

Shaft Basis systems

- The size of the shaft is kept constant and the hole size is varied to obtain various types of fits.
- Fundamental deviation or the upper deviation of the shaft is zero.

- System is not preferred in industries, as it requires more number of standard- size tools, like reamers, broaches, and gauges, increases manufacturing and inspection costs.



(a) Clearance fit (b) Transition fit (c) Interference fit

In a limit system, the following limits are specified for a hole and shaft assembly:

Hole = $30^{+0.02}_{+0.00}$ mm and shaft = $30^{-0.02}_{-0.05}$ mm
Determine the (a) tolerance and (b) allowance.

(a) Determination of tolerance:

Tolerance on hole $= HLH - LLH$
$= 30.02 - 30.00 = 0.02$ mm

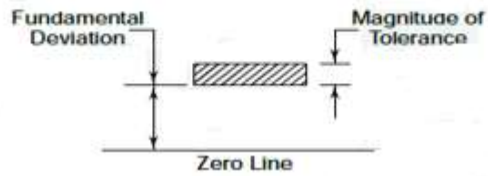
Tolerance on shaft $= HLS - LLS$
$= [(30 - 0.02) - (30 - 0.05)] = 0.03$ mm

(b) Determination of allowance:

Allowance = Maximum metal condition of hole – Maximum metal condition of shaft
$= LLH - HLS$
$= 30.02 - 29.98 = 0.04$ mm

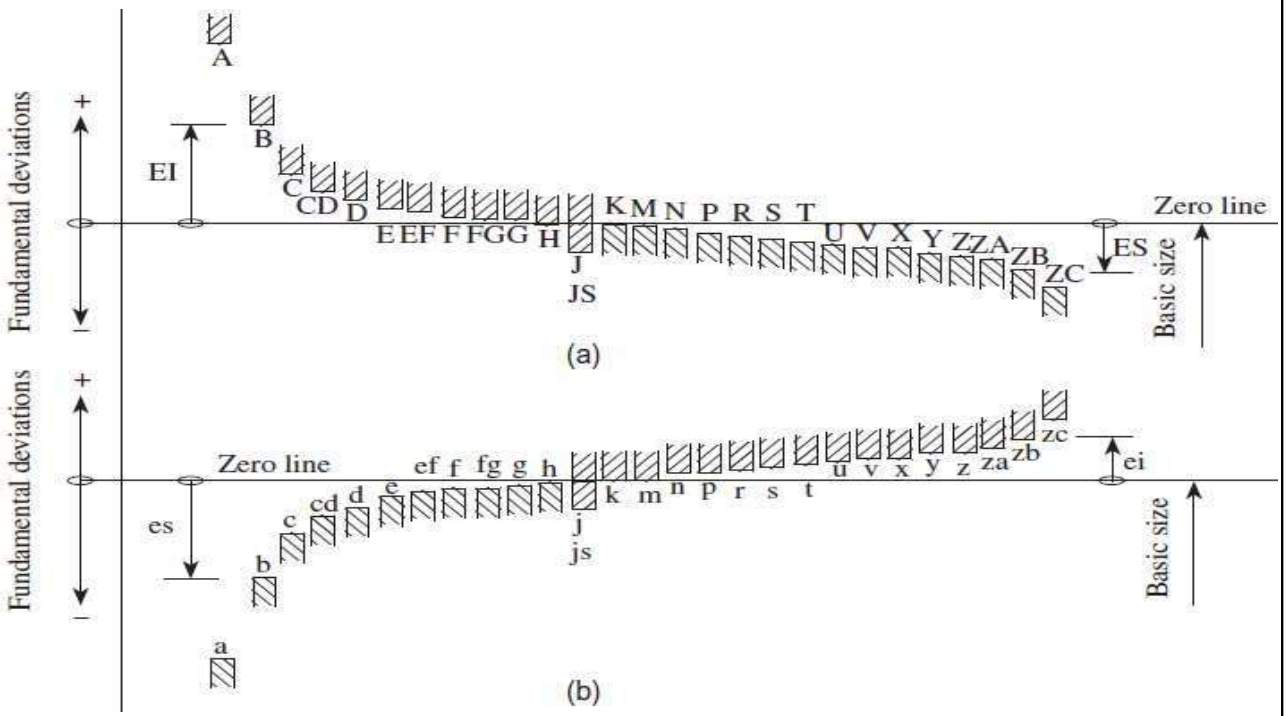
BIS system of fits and tolerances

- According to a system recommended by the Bureau of Indian Standards tolerance is specified by an alphabet, capital or small, followed by a number, e.g., H7 or g6.
- The description of tolerance consists of two parts—*fundamental deviation* and *magnitude of tolerance*, as shown in Fig.



- The fundamental deviation gives the location of the tolerance zone with respect to the zero line.
- It is indicated by an alphabet—capital letters for holes and small letters for shafts.

- There are 25 fundamental deviations (A to ZC for Hole and a to zc for shaft)



Typical representation of different types of fundamental deviations
(a) Holes (internal features) (b) Shafts (external features)

- Fundamental Deviation: Deviation either the upper or lower deviation, nearest to the zero line. (provides the position of the tolerance zone). It may be positive, negative, or zero.
- Upper deviation: Designated as 'ES' for a Hole and as 'es' for a shaft.
- Lower deviation: Designated as 'EI' for a Hole and as 'ei' for a shaft.
- First eight designations from A(a) to H(h) for holes(shafts) are used for clearance fit
- Designations, JS (js) to ZC (zc) for holes (shafts), are used for interference or transition fits

Tolerance Grade

- The magnitude of tolerance is designated by a number, called the grade.
- The grade of tolerance is defined as a group of tolerances, which are considered to have the same level of accuracy for all basic sizes
- BIS: 18 grades of fundamental tolerances are available.
- Designated by the letters IT followed by a number.
- ISO/BIS: IT01, IT0, and IT1 to IT16.
- Tolerance values corresponding to grades IT5 – IT16 are determined using the standard tolerance unit (i, in µm)

Problem: Determine the tolerance on the hole and the shaft for a precision running fit designated by 50 H7g6.

1) 50 mm lies between 30- 50 mm 2) i (microns) = $0.45D^{1/3} + 0.001D$

- Fundamental deviation for 'H' hole = 0
- Fundamental deviation for 'g' shaft = $-2.5D^{0.34}$

5) IT7 = 16i and IT6 = 10i

State the actual maximum and minimum sizes of the hold and shaft and maximum and minimum clearance.

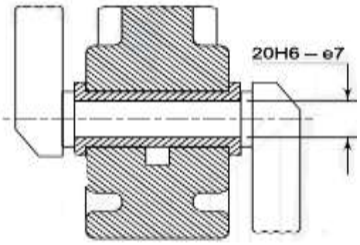
Solution

50 mm diameter lies in the standard diameter step of 30-50 mm
D= 38.7 mm
Fundamental tolerance unit = i (microns) = $0.45D^{1/3} + 0.001D = 1.5597\ \mu$
For H7 hole
Fundamental tolerance (tolerance grade table) = 16i = 24.9μ = 0.025 mm
For ‘H’ Hole, fundamental deviation is 0 (from FD Table)
Hence, hole limits are.
Hole Tolerance = 25.033 – 25 = 0.033 mm
50 mm diameter lies in the standard diameter step of 30-50 mm
D= 38.7 mm
Fundamental tolerance unit = i (microns) = $0.45D^{1/3} + 0.001D = 1.5597\ \mu$
For g6 shaft
Fundamental tolerance (tolerance grade table) = 10i = 16μ = 0.016 mm For ‘g’ shaft, fundamental deviation is $-2.5D^{0.34} = 9\ \mu$
Maximum clearance = 50.025 - 49.975 = 0.05 mm
Minimum clearance = 50.000 - 49.991 = 0.009

Selective assembly

- The selective assembly is a process of sorting the manufactured components into different groups according to their sizes, and then assembling the components of one group to the corresponding components of a matching group.
- In this method, larger shafts are assembled with larger holes and smaller shafts with smaller holes.
- This results in closer clearance or interference without involving costly machining methods.

Problem: The main bearing of an engine is shown in Fig. Calculate (i) the maximum and minimum diameters of the bush and crank pin; and (ii) the maximum and minimum clearances between the crank pin and bush.



Solution

Step I *Maximum and minimum diameters of the bush and crank pin*
The tolerances for the bush and crank pin from Tables 3.2 and 3.3a are as follows:
Bush:

$$20^{+0.013}_{+0.000} \quad \text{or} \quad \frac{20.013}{20.000}\ \text{mm}$$

Crankpin:

$$20^{-0.040}_{-0.061} \quad \text{or} \quad \frac{19.960}{19.939}\ \text{mm} \qquad \text{(i)}$$

Step II *Maximum and minimum clearances*
Maximum clearance = 20.013 – 19.939 = 0.074 mm
Minimum clearance = 20.000 – 19.960 = 0.040 mm
(ii)

Surface roughness

The surface roughness or surface finish plays an important role in the performance of certain machine elements.

Some of the examples are as follows:

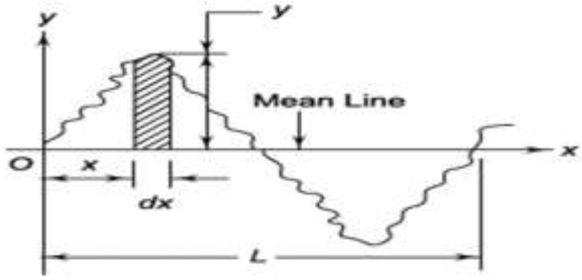
(i) Friction and wear increases with surface roughness, adversely affecting the performance of bearings.

(ii) Rough surfaces have a reduced contact area in interference fits, which reduces the holding capacity of the joints.

(iii) The endurance strength of the component is greatly reduced due to poor surface finish.

(iv) The corrosion resistance is adversely affected by poor surface finish.

There are two methods to specify surface roughness – Centre line average (**cla)** value and (root mean square) **rms** value.



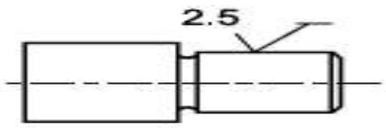
The **cla** value is defined as

$$\text{cla} = \frac{1}{L}\int_0^L y\,dx$$

and the **rms** (root mean square) value as

$$\text{rms} = \left[\frac{1}{L}\int_0^L y^2\,dx\right]^{1/2}$$

A widely used symbol for surface finish is shown in Fig.



Symbol for Surface Roughness

The sides of the symbol are inclined at 60° to the surface and the number indicates the surface roughness (**rms**) in microns.

Unit-3: Design for Strength and Design of Members subjected to direct stress

DESIGN FOR STRENGTH

Introduction:- The fundamental requirements for the machine elements, on which their efficiency depends are: strength, stiffness, wear and corrosion resistance. Wear is most when two surfaces of machine elements have relative motion. Selection of good materials and surface protection will help in preventing the corrosion of the machine surfaces. The other factor that can cause failure of the machine part is its excessive deformation. Deformation means the change of shape due to elastic strain, plastic strain, and fracture.

When a machine part or element is subjected to a load, internal stresses are induced in the part by the applied load. These internal stresses resist the deformation of the part. If the dimensions of the part are based on the elastic deformation so as to prevent excessive elastic deformation, the design is called "rigidity or "stiffness" design. On the other hand, if the design is based on the induced stresses or on plastic deformation, it is said to be the "strength Design".

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Most machine parts are designed on the basis of "strength design" due to the following reasons:

(1) It is easier to base the design on strength than on stiffness

(2) The machine parts designed on the basis of strength usually have sufficient elastic rigidity or stiffness

(3) For parts which are not simple in form, it is very difficult to determine the deflections than to determine the induced stresses.

(4) It is more difficult or impossible to set a limit for permissible elastic deflections than to fix upon the allowable stress.

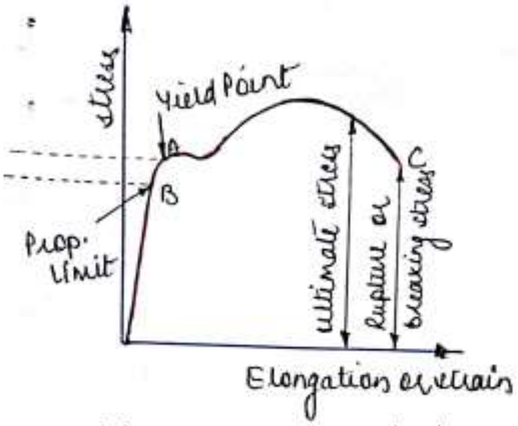
"springs are designed for both strength & stiffness"

★ Failure Criterion :-

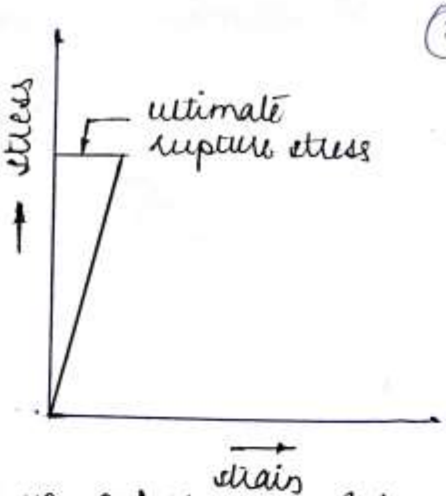
Before finding out the dimensions of the component it is necessary to know the type of failure by which the component will fail when put into the service. The machine component is said to have 'failed' when it is unable to perform its functions satisfactorily. The three basic type of failure are as follows:-

(1) Failure by elastic deflection
(2) Failure by general yielding and
(3) Failure by fracture.

3



(a) Low carbon steel



(b) Brittle material

(1) In applications like transmission shaft supporting gears, the maximum force acting on the shaft is limited by the permissible elastic deflection. When this deflection exceeds a particular value, the meshing between teeth of gears is affected and the shaft cannot perform its function properly and the shaft is said to have 'failed' due to elastic deflection.

(2) Components made of ductile materials like steel lose their engineering usefulness due to large amount of Plastic deformation. This type of failure is called failure by yielding

(3) Components made of brittle materials like cast iron cease to function because of sudden fracture without any Plastic deformation.

* Design of simple Machine Parts:-

(4)

The design of machine Parts is concerned with the determination of dimensions of the parts. In this process a designer should make sure that the stresses developed in a part, are less than the maximum allowable strength of the material. This maximum allowable stress is referred as the design stress and its value depends upon many factors:-
(a) types of load, (b) service conditions (c) Life cycle (d) safety considerations

- Types of load:-

The external forces grouped together constitute what is called the load acting on the body. The nature of load may vary from dead weight to varying load.

- Static Load:- A static or steady load is a stationary force, moment or torque acting on a member. It does not change its magnitude, point of application and direction
- Impact or shock load:- When a component is subjected to a sudden load or when the duration of loading is less than about half the natural period of vibration
- Fatigue load:- The alternating or repeated loading In cyclic loading the load fluctuates between the maximum and minimum levels of loading.

(5)

★ FACTOR OF SAFETY :-

While designing a component, it is necessary to ensure sufficient reserve strength in case of an accident. A designer who designs a component must ensure that the part will not fail during its service period. The failure may be attributed to one or more of the following causes:

(i) uncertainty about the properties of material
(ii) uncertainty about the magnitude of the load, its point of application and direction
(iii) uncertainty about the accuracy of the mathematical model which represents the predicted behaviour of the machine part
(iv) uncertainty about the operating conditions and reliability requirements.

To account for all the uncertainities, we provide a sufficient margin of safety called factor of safety (FOS).

$$\boxed{F.O.S. = \frac{\text{critical stress or failure stress}}{\text{working stress}}}$$

- The ratio of maximum or critical or failure stress to the working stress is called FACTOR OF SAFETY.

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- CRITICAL STRESS:-

It is the highest stress induced in the machine element before failure. Failure means not any actual breaking of material but also a body will not perform its allotted function properly due to plastic deformation.

For ductile material yield point is the critical stress

For Brittle material the critical stress is the ultimate stress

For variable loading the endurance limit is the critical stress.

- WORKING STRESS:- (which is used to determine the dimensions of the component.

It is a common practice to keep the stresses lower than maximum or critical stresses. at which failure (rupture) of the machine element does not take place. These stresses, used in designing a machine element are known as working stress, design stress permissible stress, allowable stress or safe stress practically the same meaning.

For ductile material, yield point is clearly defined so F.O.S. is based on yield point stress, so

$$\boxed{F.O.S. = \frac{\text{Yield Point stress}}{\text{working stress}}}$$

(7)

For brittle material like C.I. the yield point is not clearly defined, hence the factor of safety is based on ultimate stress

$$\boxed{F.O.S. = \frac{\text{ultimate stress}}{\text{working stress}}}$$

For reversible load, factor of safety is based on endurance limit stress

$$\boxed{F.O.S. = \frac{\text{Endurance limit stress}}{\text{working stress}}}$$

- ALLOWABLE STRESS :-

It is the stress allowed to be induced in the machine element keeping the safety in mind

$$\boxed{\text{Allowable stress} = \frac{\text{Critical stress}}{F.O.S.}}$$

$$\Rightarrow \boxed{\sigma_a = \frac{S_{yt} \text{ or } S_{ut}}{F.O.S.}}$$

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- Selection of Factor of Safety :-

The magnitude of factor of safety depends upon following factors:-

(1) Effect of Failure :- Some times the failure of machine element involves only a little inconvenience or loss of time Ex:- Failure of ball bearing in a gear box. On the other hand in some cases, there is substantial financial loss or danger to the human life. e.g. failure of the valve in pressure vessel. The FOS is high in applications where failure of machine part may result in serious accidents.

(2) Type of load :- The FOS is low when the external force acting on the machine element is static and a higher FOS is selected when the machine element is subjected to impact load.

(3) Degree of accuracy in force analysis :- when the forces acting on the machine component are precisely determined, a low FOS can be selected.

(4) Material of component :- when the component is made of homogeneous ductile material like steel yield strength is the criterion of failure. FOS is small in such cases. On the other hand, C.I. component has non-homogeneous structure and a higher FOS. based on ultimate tensile strength is chosen.

(5) Reliability of component :-
FOS increases with increase in reliability.

(6) Cost of component :- As the FOS increases, dimension of component, material requirement and cost increase
FOS is low for cheap machine parts

(7) Testing of machine element :- A low FOS can be choosen when the machine component can be tested under actual conditions of service and operation
A higher FOS is necessary, when it is not possible to test the machine part or where there is deviation between test conditions and actual service conditions.

(8) Service conditions :- when the machine element is likely to operate in corrosive atmosphere or high temperature environment, higher FOS is necessary.

(9) Quality of Manufacture :- when the quality of manufacture is high, variations in dimensions of machine component are less and a low FOS can be selected, A higher FOS is required to compensate for poor manufacturing quality.

• Selection of quantitative values of the FOS :-

(i) For cast iron components :- ultimate tensile strength is considered to be the failure criterion
C.I. components have a non-homogeneous structure.
therefore a large FOS, usually 3 to 5 is used.

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(ii) For components made of ~~ductile~~ ductile material ~~like~~ subjected to external static forces
Yield strength is considered to be the criterion of failure. Ductile materials have a homogeneous structure and the residual stresses can be relieved by proper treatment. The recommended FOS is 1.5 to 2

(iii) For components made of ductile materials and subjected to external fluctuating forces –
Endurance limit is considered to be the criterion of failure. Fatigue failure is sudden and total and to account for this the recommended FOS is usually 1.3 to 1.5

(iv) The design of certain components such as cam and follower, gears, rolling contact bearings. is based on calculation of contact stresses – Failure of such components is usually in the form of small pits on the surface of the component. The recommended FOS for such components is 1.8 to 2.5 based on surface endurance limit

(v) Certain components, such as piston rod, power screws or studs, are designed on the basis of buckling consideration – The recommended FOS. is 3 to 6 based on critical buckling load of such components

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(4)

The maximum stress is the largest algebric value and the minimum stress is the lowest algebric value of a variable stress. The mean stress σ_m is the average of the maximum & minimum stress. The variable stress σ_v is the increase or decrease in stress above or below the mean stress.

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

$$\sigma_v = \frac{\sigma_{max} - \sigma_{min}}{2}$$

☆ Stress Concentration :-

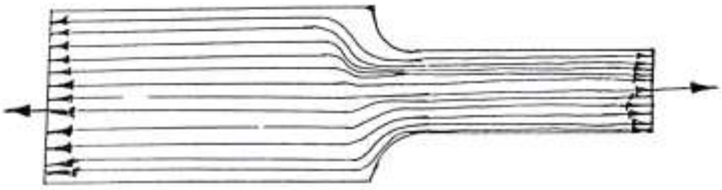
Elementary equations for stresses in the component subjected to tensile force, bending moment and torsional moment are,

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

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

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

These equations are based on the assumption that there are no discontinuities in the cross section of the component. In practice discontinuities are abrupt changes of cross-section are present due to oil holes and grooves, keyways, screw threads etc. The assumption of a uniform cross section cannot be made under these circumstances and the elementry equations do not give correct ~~req~~ results.



stress concentration

* Definition :- stress concentration is defined as the localisation of high stresses due to irregularities or abrupt changes of the cross section. The stresses obtained by the elementry equations are modified to account for the effect of stress concentration

The stress concentration factor K_t is defined as

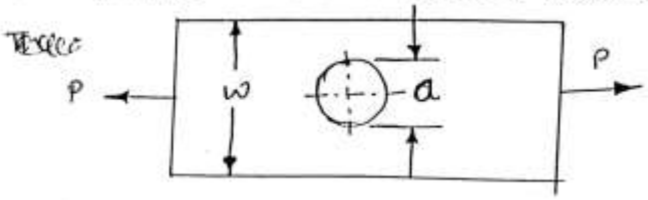
$$K_t = \frac{\text{Highest value of actual stress near discontinuity}}{\text{Nominal stress obtained by elementry equation for minimum c/s}}$$

or $$K_t = \frac{\sigma_{max}}{\sigma_0} = \frac{\tau_{max}}{\tau_0}$$

where σ_0 and τ_0 are stresses calculated by the elementry equations for minimum cross-section. The subscript t denotes the 'theoretical' stress concentration factor and its value depends upon the geometry of the component.

* stress concentration factors for different geometry.

(1) stress concentration factor for a rectangular Plate with a transverse hole loaded in tension or compression



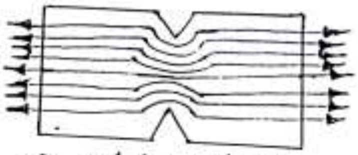
when the stress in the vicinity of the discontinuity reaches the yield Point, there is Plastic deformation resulting in a redistribution of stresses. Therefore, stress concentration factors are not used for ductile materials under static loading.

In case of brittle materials due to their inability to Plastic deformation stress concentration factors are used for components made of brittle materials subjected to static loads.

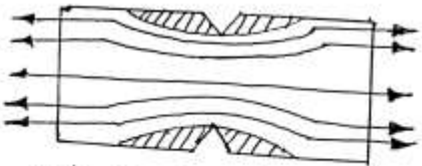
* Reduction of stress concentration effects:-

It is possible for the designer to reduce the severity of stress concentration by selecting the correct geometric shape. A single notch results in a high degree of stress concentration. The severity of stress concentration is reduced by using the ~~principal~~ Principle of minimization of the material

(1) use of multiple notches
(2) removal of undesired material
(3) drilling additional holes



(A) original Notch



(B) Removal of undesired material

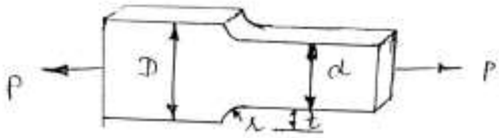
(8)

The nominal stress $\sigma_0 = \frac{P}{(w-a)h}$

h = Plate thickness

K_t = Data Hand Book Page No - 7.10

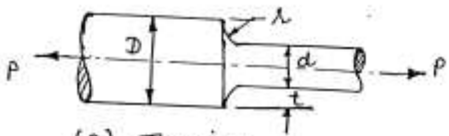
(2) stress concentration factor for a flat Plate with a shoulder fillet subjected to tensile or compressive force



The nominal stress σ_0 for this case is $\sigma_0 = \frac{P}{dt}$

K_t = Data Hand Book Page no - 7-9

(3) stress concentration factor for a round shaft with shoulder fillet subjected to tensile force bending moment and torsional moment

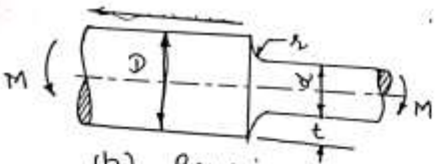


(a) Tension

Tensile force:

$\sigma_0 = \frac{P}{\left(\frac{\pi}{4}d^2\right)}$

K_t = data hand book Page no- 7.11



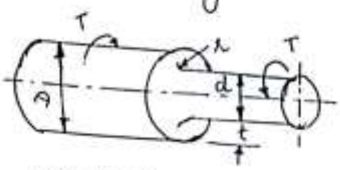
(b) Bending

Bending moment

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

where $I = \frac{\pi}{64}d^4$ and $y = \frac{d}{2}$

K_t = Data Hand Book Page No - 7.14



(c) Torsion

Torsional moment

$\tau_0 = \frac{T}{J} r$ where $J = \frac{\pi}{32}d^4$ & $r = d/2$

K_t = Data Hand Book Page No- 7.18

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★ Causes and Mitigation of stress concentration:-

Causes:-

(1) Variation in properties of materials:-

In design of machine components, it is assumed that the material is homogeneous throughout the component. In practice, there is variation in material properties from one end to another due to following factors:

(a) Internal cracks and flaws like blow holes.
(b) cavities in welds
(c) Air holes in steel components, and
(d) Nonmetallic or foreign inclusions

(2) Load application:- In machine components forces act either at a point or over a small area on the component. The pressure at these points is usually excessive and causes stress concentration.

Examples:-
(a) contact between the meshing teeth of the driving and driven gear.
(b) contact between the cam and follower
(c) contact between the balls and the races of ball bearing.
(d) contact between the crane hook and the chain.

(3) Abrupt changes in section:- In order to mount gears, sprockets, pulleys and ball bearing on transmission shaft, steps are ~~cut~~ cut on the shaft and shoulders are provided from assembly considerations. Although these features are essential, they result in changes in the c/s of the shaft.

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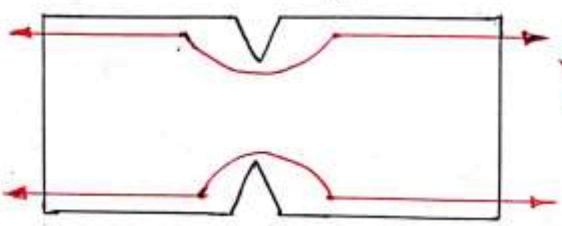
(4) Discontinuities in the component:–
certain features of machine components such as oil holes or grooves, keyways and screw threads result in discontinuities in the c/s of the component results in stress concentration.

(5) Machining scratches:– Machining scratches stamp mark or inspection mark are surface irregularities which cause stress concentration.

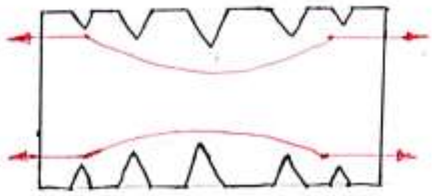
***Mitigation:–**

Although it is not possible to eliminate the effect of stress concentration, The effect of stress concentration is proportional to the bending of the force flow lines Therefore, stressconcentration can be reduced by minimising the bending of the flow lines by following methods:–

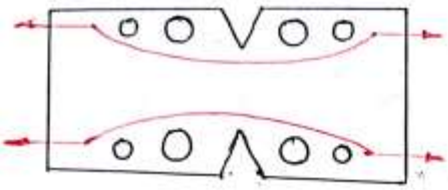
(i) Additional notches and holes in tension member:
A flat Plate with a V-notch subjected to tensile force as shown in fig. A single notch results in a high degree of stress concentration. It can be reduced by (a) use of multiple notches
(b) drilling additional holes
(c) removal of undesired material



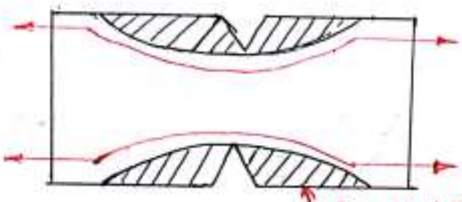
stress concentration due to V-notch



(a) Multiple Notches



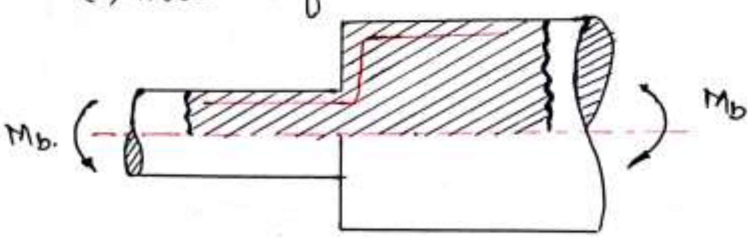
(b) Due Drilling additional holes.



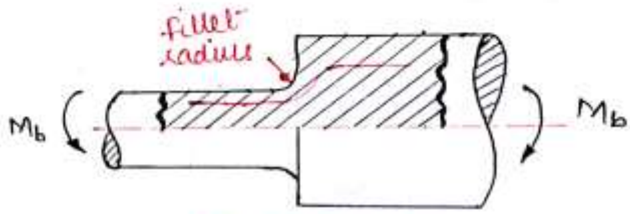
(c) Removal of undesired material

(2) Fillet radius, undercutting and notch for member in tension:-

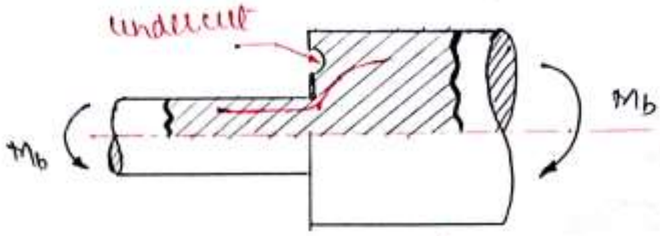
A bar of circular crosssection with a shoulder, which is subjected to bending moment. There are three methods to reduce stress concentration at the base of shoulder (a) Fillet Radius (b) undercutting (c) Addition of notch.



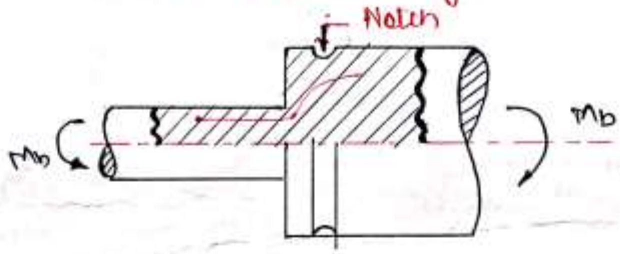
Original Component



(a) Fillet Radius



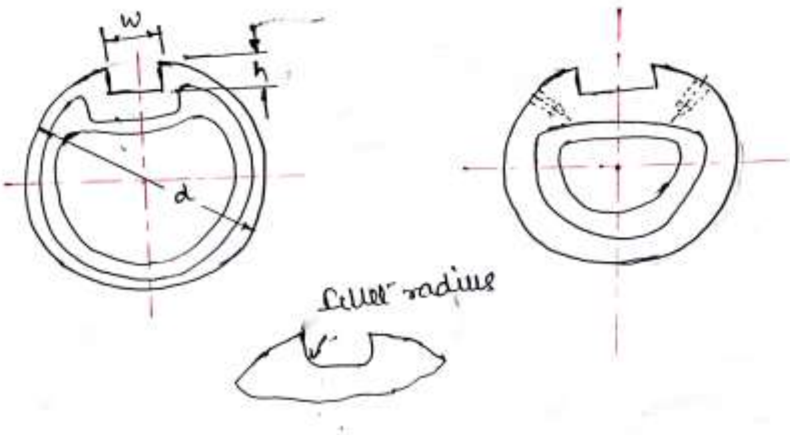
(b) Undercutting



(c) Addition of Notch

(3) Drilling additional holes for shaft :-

A transmission shaft with a keyway is shown in fig. keyway is a discontinuity and results in stress concentration at the corners of the key way and reduces torsional shear strength. The stress concentration can be reduced by rounding corners by providing fillet radius. In addition to giving fillet radius at the inner corners of keyway, there is another method of drilling two symmetrical holes on the sides of keyway. These holes press the ~~for~~ flow lines and minimize their bending in the vicinity of the keyway

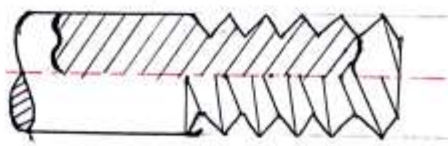


(4) Reduction of stress concentration in threaded members:-

In a threaded component the flow line is bent as it passes from shank portion to threaded portion, this results in stress concentration in the transition plane. Methods to reduce stress concentration:-

(a) A small undercut is taken between the shank and the threaded portion of the component and fillet radius is provided for this undercut.

(b) The shank diameter is reduced and made equal to the core diameter of the thread.



Original component.

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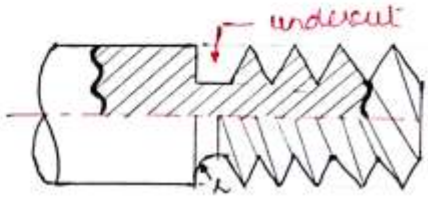
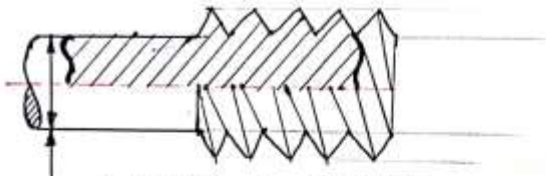


(a) undercutting



(b) Reduction in shank diameter

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Fatigue stress concentration factor.

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

* Notch sensitivity :-

Notch sensitivity is defined as the degree to which the theoritical effect of stress concentration is actually reached. The stress gradient depends mainly on the radius of the notch, hole or fillet and on the grain size of the material. It is denoted by 'q'. When the notch sensitivity factor q is used in cyclic loading, then fatigue stress concentration factor may be obtained from the following relations:-

$$\boxed{q = \frac{K_f - 1}{K_t - 1}}$$

or $K_f = 1 + q(K_t - 1)$ --- (for tensile or bending)

and $K_{fs} = 1 + q(K_{ts} - 1)$ --- (for shear stress)

K_t = Theoritical stress concentration factor for axial or bending loading.

K_{ts} = Theoritical stress concentration factor for torsional or shear loading.

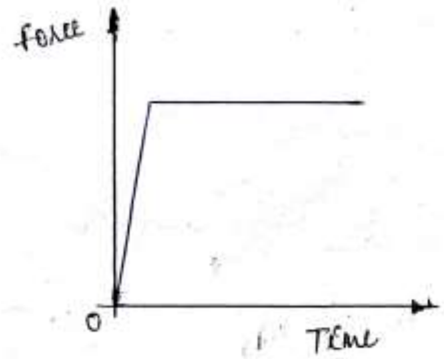
The value of q can be find out by using data hand book.

★ Static and Dynamic Loads:—

(a) **Static load**

(1) Static load is defined as a load which does not vary in magnitude or direction with respect to time, after it has been applied.

(2) Static load is gradually applied to a member and once it is applied, it remains constant.



Static load

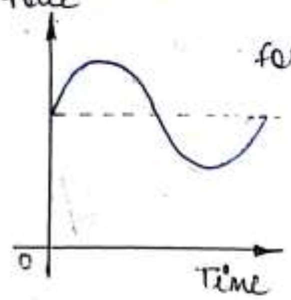
Dynamic load.

(1) Dynamic load is a load which varies in magnitude or in direction or in both with respect to time after it has been applied

(2) There are two types of dynamic loads —
(a) cyclic load (b) Impact load

(a) Cyclic load is a load which when applied, varies in magnitude in a repetitive manner, either completely reversing itself from tension to compression or oscillating about some mean value.

(b) An impact load is a load which is applied suddenly to a member usually at a high velocity and the load in time reaches to its static value.



Cyclic load

Impact load

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(3) Examples of static loads are dead weights of machinery, tightening-up load on bolts and fasteners and load acting on the walls and cover of a pressure vessel subjected to constant pressure.

(4) When a component is subjected to static load, two types of mechanical failures may occur - yielding and fracture.

(5) It is comparatively easy to determine static load and design a component on the basis of static load.

(6)

(3) Examples of cyclic loads are forces induced in gear teeth, loads induced in a rotating shaft subjected to bending moment or forces acting on the compression spring of the valves in IC engines.
~~Apen~~ A punch press or rivet gun are examples of impact loads

(4) In cyclic loading failure may occur when the load cycles are repeated several million times even though the stress is below the elastic limit. This type of failure is called fatigue failure.

(5) The pattern of stress variation due to dynamic loads is irregular and unpredictable. It is difficult to design a component subjected to dynamic loads

(6)

reduced that the remaining portion is no longer in a position to sustain the external force and it is subjected to sudden fracture. Two distinct areas on the fractured surface of the component are:-

(i) A ~~ee.~~ smooth surface area with fine fibrous appearance indicating slow growth of crack.

(ii) A rough surface area with coarse granular appearance indicating sudden fracture.

Fatigue failure occurs without the warning of gross plastic deformation. Fatigue cracks are not visible till they reach the surface of the component and by that time the failure has already occured. The failure is sudden and total. The fatigue failure depends upon a number of factors:-

(1) Number of cycles (2) mean stress

* FATIGUE FAILURE :-

Machine components fail under cyclic stresses at a stress magnitude which is much smaller than the ultimate tensile strength of the material. Sometimes, the magnitude is even lower than the yield strength. The failure begins with a crack, which occurs at a highly stressed point in the component. The crack grows during the cycles and failure occurs suddenly without any indication.

The crack is more likely to occur in the following regions:-

(1) Regions of discontinuity such as oil holes or keyways
(2) Regions of abrupt change in cross-section such as shoulder or steps.
(3) Regions of irregularities in machining operations such as machining scratches, stamp mark or inspection marks.
(4) Internal cracks in materials like blow holes.

These regions are subjected to stress concentration due to presence of crack. The crack propagates with increasing number of stress cycles. A component with a crack, which is subjected to alternate tensile and compressive stresses is shown in fig. During first half of the cycle, the stress is tensile and the crack opens. During the second half of the cycle, there is compressive stress and crack closes. This opening and closing of the crack during each cycle continues and results in crack propagation. Finally the cross section of the component is so

(3) stress amplitude (4) stress concentration
(5) Residual stresses (6) corrosion and creep.

Examples of parts in which fatigue failures are common are transmission shafts, connecting rods, gears, vehicle suspension springs and ball bearings.

★ **Endurance Limit:-**

The fatigue or endurance limit of a material is defined as the maximum amplitude of completely reversed stress that the standard specimen can sustain for an unlimited number of cycles without fatigue failure. Since the fatigue test cannot be conducted for unlimited number of cycles without fatigue failure, 10^6 cycles is considered as a sufficient number of cycles to define the endurance limit, it is denoted by (σ_e).

★ **Fatigue stress concentration factor:-**

When a machine member is subjected to cyclic or fatigue loading, the value of fatigue stress concentration factor shall be applied instead of theoretical stress concentration factor. Since the determination of fatigue stress concentration factor is not an easy task, therefore from experimental tests it is defined as.

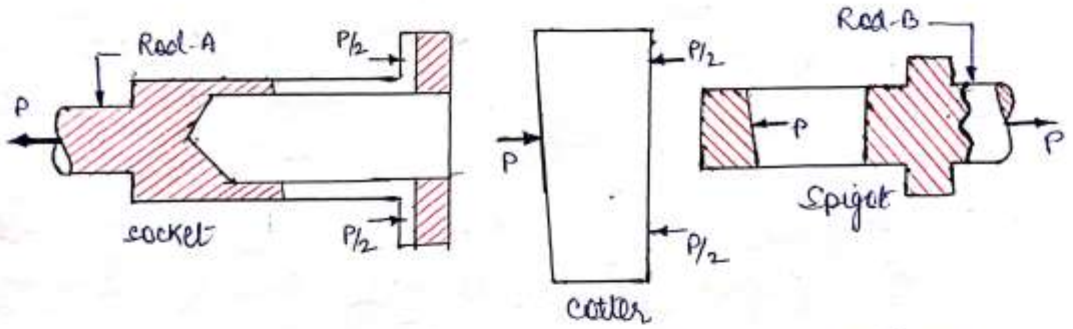
(17)

* Design of simple machine Parts :-

(1) Cotter Joint :-

Introduction :- A cotter joint is a Temporary fastening and used to connect two rigidly coaxial rods, which are subjected to axial tensile or compressive loads. One end of the rod is made with socket type of end and the other end (spigot) is inserted into socket.

* Main Parts of cotter Joint :-



Free Body diagram of Parts with forces

(1) Rod with spigot end :- The end of the rod, which goes into a socket, is called spigot. It is a circular in shape with rectangular slot on enlarged portion and having a collar on the spigot end.

(2) Rod with socket end :- The end of the rod, which is hollow (up to certain extent) with rectangular slot and collar at its end is called socket. The spigot end is inserted into a socket.

(3) Cotter:- A cotter is a flat wedge shaped piece of rectangular cross section. The thickness is uniform but width is tapered through its length. The top and bottom ends of cotter are often round for easy fixing and removing. The cotter is fitted tightly through a rectangular slot in order to mak the temporary connection between two rods. The taper varies from 1 in 48 to 1 in 24.

★ Application of cotter Joint:-

- In reciprocating steam engine to join the piston rod with cross head.
- In cotter foundation bolts
- In Jigs and fixtures
- To join two halves of flywheels.

★ Advantages of cotter Joint:-

- Joint can be done easily and quickly as per the requirement.
- With the help of sleeve structural pipe joint can be made.
- A strong joint to withstand heavy load can be done by cotter joint.

* Disadvantages and Limitations of cotter Joints:-

- Manufacturing time and cost is more
- More time is required to join rods than knuckle joint
- Due to wedge shaped cotter, this type of joint is not for the parts having rotational motion.

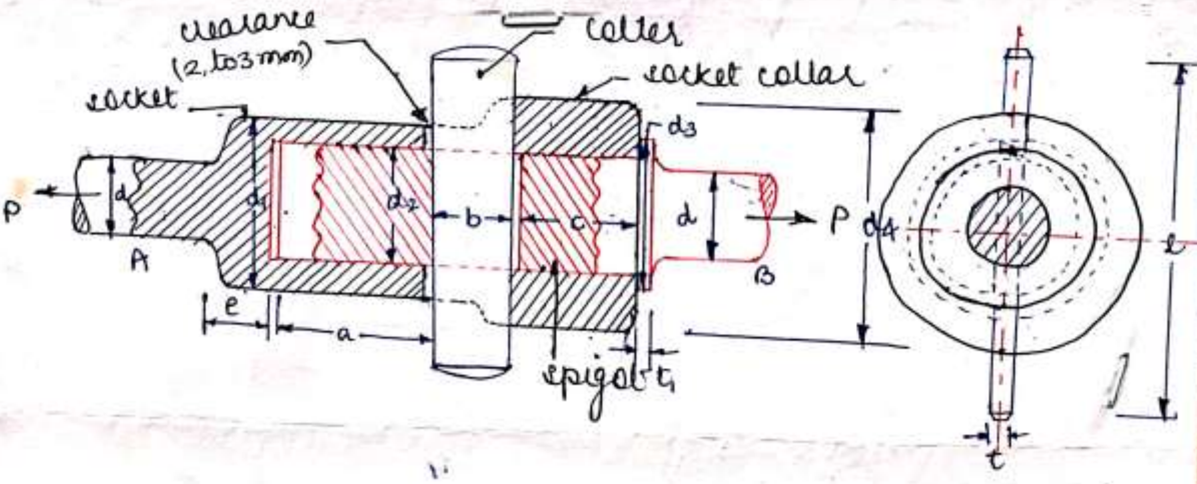
* Material used for cotter Joint:-

- Mild steel
- Alloy steel
- Wrought iron.

* Types of cotter Joint:-

(1) socket and spigot cotter joint
(2) sleeve and cotter joint
(3) Gib and cotter joint.

* Design of socket and spigot cotter Joint:-



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Let P = Load carried by the rods
d = Diameter of the rods
d_1 = outside diameter of socket
d_2 = Diameter of spigot or inside diameter of socket
d_3 = outside diameter of spigot
t_1 = Thickness of spigot collar
d_4 = diameter of socket collar
c = Thickness of socket collar
b = mean width of cotter
t = Thickness of cotter
l = length of cotter
a = Distance from the end of the slot to the end of rod.
σ_t = Permissible tensile stress for the rods material
τ = Permissible shear stress for the cotter material
σ_c = Permissible crushing stress for the cotter material

For the purpose of stress analysis, following assumptions are made:-

(i) The rods are subjected to axial tensile force
(ii) The effect of stress concentration due to slot is neglected

(1) Tensile Failure of Rods:-
The rods may fail in tension due to the tensile load P.

Area resisting tearing $= \frac{\pi}{4} \times d^2$

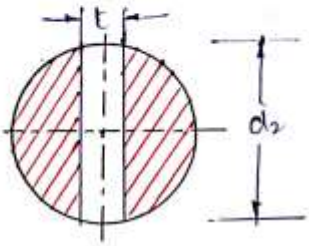
$\therefore$ Tearing strength of the rods $= \frac{\pi}{4} \times d^2 \times \sigma_t$

Equating this to load (P), we have

$$\boxed{P = \frac{\pi}{4} \times d^2 \times \sigma_t}$$

From this equation, diameter of the rods (d) may be determined.

(2) Failure of spigot in tension :-



Since the weakest section of the spigot is that section which has a slot in it for the cotter, therefore area resisting tearing of the spigot across the slot

$$= \frac{\pi}{4} d_2^2 - d_2 \times t$$

and tearing strength of the spigot across the slot

$$= \left[\frac{\pi}{4} d_2^2 - d_2 \times t\right] \sigma_t$$

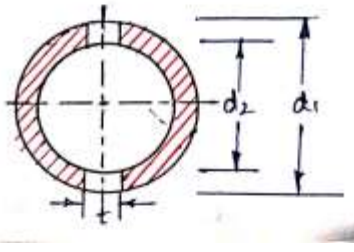
Equating this to load (P), we have

$$\boxed{P = \left[\frac{\pi}{4} d_2^2 - d_2 \times t\right] \sigma_t}$$

From this equation diameter of spigot (d_2) may be determined

In actual practice, the thickness of cotter is $\boxed{t = d_2/4}$

(3) Failure of socket in tension :-



The resisting area of the socket across the slot

$$= \frac{\pi}{4}\left[(d_1)^2-(d_2)^2\right]-(d_1-d_2)t$$

$\therefore$ Tearing strength of the socket across the slot

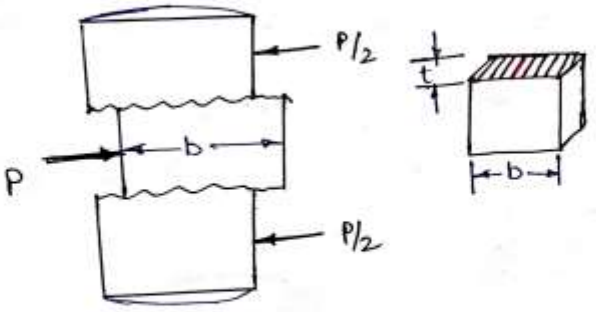
$$= \left\{\frac{\pi}{4}\left[(d_1)^2-(d_2)^2\right]-(d_1-d_2)t\right\}\sigma_t$$

Equating this to load (P),

$$\boxed{P = \left\{\frac{\pi}{4}\left[(d_1)^2-(d_2)^2\right]-(d_1-d_2)\,t\right\}\sigma_t}$$

From this equation, outside diameter of socket (d_1) may be determined.

(4) Failure of cotter in shear:-



Considering the failure of cotter in shear, since the cotter is in double shear, therefore shearing area of the cotter $= 2b \times t$

and shearing strength of the cotter $= 2b \times t \times \tau$

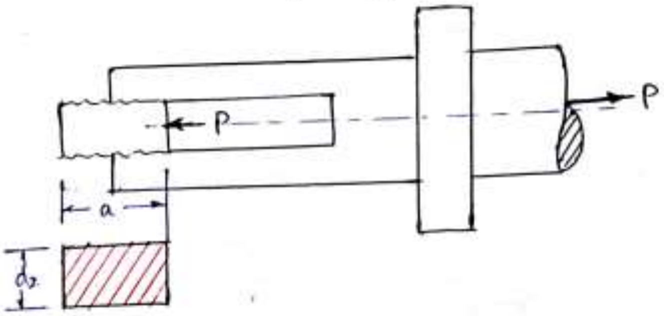
Equating this to load (P),

$$\boxed{P = 2b \times t \times \tau}$$

From this equation, width of cotter (b) is determined

(28)

(5) shear failure of spigot End:-



since the rod or spigot end is in double shear, therefore the area resisting shear of the rod end $= 2\,a \times d_2$

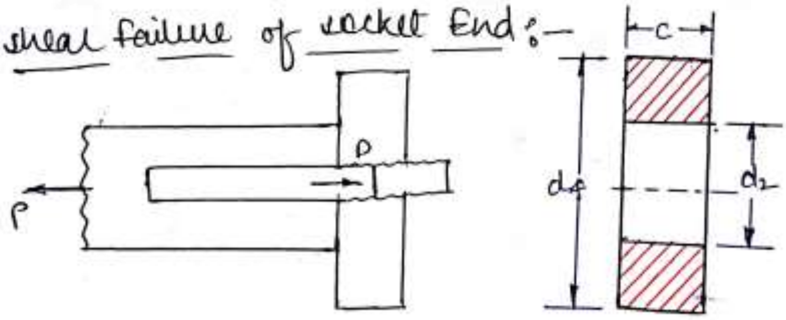
and shear strength of the spigot end $= 2\,a \times d_2 \times \tau$

Equating this to load(P),

$$\boxed{P = 2 \times a \times d_2 \times \tau}$$

From this equation, the distance from the end of the slot to the end of the rod (a) may be obtained.

(6) shear failure of socket End:-



since the socket end is in double shear, therefore area resisting shear of the ~~rod end~~ socket collar

$= 2(d_4 - d_2)c$

and shearing strength of socket collar

$= 2(d_4 - d_2)c \times \tau$

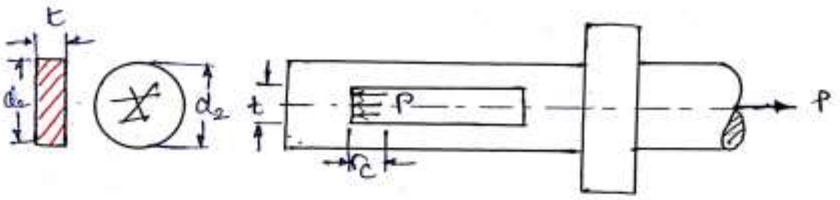
(24)

Equating this to Load (P).

$$\boxed{P = 2(d_4 - d_2)\, c \times \tau}$$

- From this equation the thickness of socket collar (c) may be determined

(7) Crushing failure of spigot end :-



The area that resists crushing of spigot end
$= d_2 \times t$

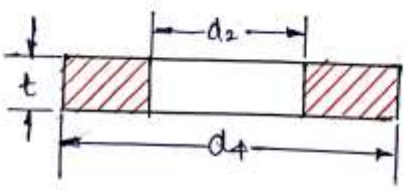
$\therefore$ crushing strength $= d_2 \times t \times \sigma_c$

Equating this to load (P).

$$\boxed{P = d_2 \times t \times \sigma_c}$$

From this equation, the induced crushing stress may be checked.

(8) Crushing failure of socket End :-



The area that resists crushing of socket collar

$= (d_4 - d_2)t$

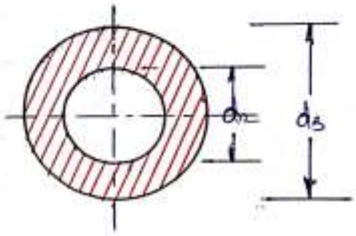
and crushing strength $= (d_4 - d_2)\, t \times \sigma_c$

Equating this to load (P),

$$\boxed{P = (d_4 - d_2)\, t \times \sigma_c}$$

From this equation, the diameter of socket collar (d_4) may be determined.

(9) Crushing failure of spigot collar :-



The area that resists crushing of the collar

$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]$

and crushing strength of collar

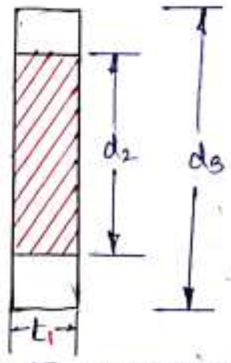
$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] \sigma_c$

Equating this to load (P),

$$\boxed{P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] \sigma_c}$$

From this equation, the diameter of the spigot collar (d_3) may be obtained.

(10) Shear failure of spigot collar :-



The area that resists shearing of the collar

$= \pi d_2 \times t_1$

and shearing strength of the collar

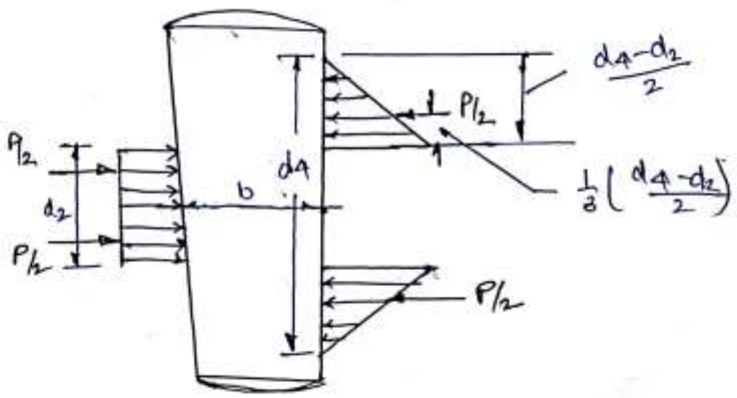
$= \pi d_2 \times t_1 \times \tau$

Equating this to load (P),

$$\boxed{P = \pi d_2 \times t_1 \times \tau}$$

From this equation, the thickness of spigot collar (t_1) may be obtained.

(11) Failure of cotter in bending:-



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In all the above relations, it is assumed that the load is uniformly distributed over the various cross-sections of the joint. But in actual practice, this does not happen and the actual cotter is subjected to bending. In order to find out the bending stress induced, it is assumed that the load on the cotter in the rod end is uniformly distributed while in the socket end it varies from zero at the outer diameter (d_4) and maximum at the inner diameter (d_2). The maximum bending moment occurs at the centre of the cotter and is given by

$$M_{max} = \frac{P}{2}\left(\frac{1}{3} \times \frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2} \times \frac{d_2}{4}$$

$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4}\right) = \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)$$

section modulus of the cotter

$$Z = t \times b^2/6$$

∴ Bending stress induced in the cotter,

$$\boxed{\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t \times b^2/6} = \frac{P(d_4 + 0.5 d_2)}{2t \times b^2}}$$

This bending stress induced in the cotter should be less than the allowable bending stress of the cotter

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★ KNUCKLE JOINT:-

Knuckle joint is used to connect two rods whose axes either coincide or intersect and lie in one plane. The construction of this joint permits limited relative angular movement between rods in this plane about the axis of the pin. The rods connected by knuckle joint are subjected to tensile force.

The construction of a knuckle joint used to connect two rods - A and B subjected to tensile force P as shown in fig. An eye is formed at the end of rod-B, while a fork is formed at the end of rod-A. The eye fits inside the fork and a pin passes through both fork and the eye. This pin is secured in its place by means of a split-pin. Due to this type of construction, knuckle joint is sometimes called "forked-pin" or simply "Pin" joint.

★ Advantages:-

(1) The joint is simple to design and manufacture
(2) There are few parts in knuckle joint which reduces cost and improves reliability
(3) The assembly or disassembly of the parts of the joint is quick and simple.

★ Limitation:-

- Knuckle joint is unsuitable to connect two rotating shafts which transmit torque.

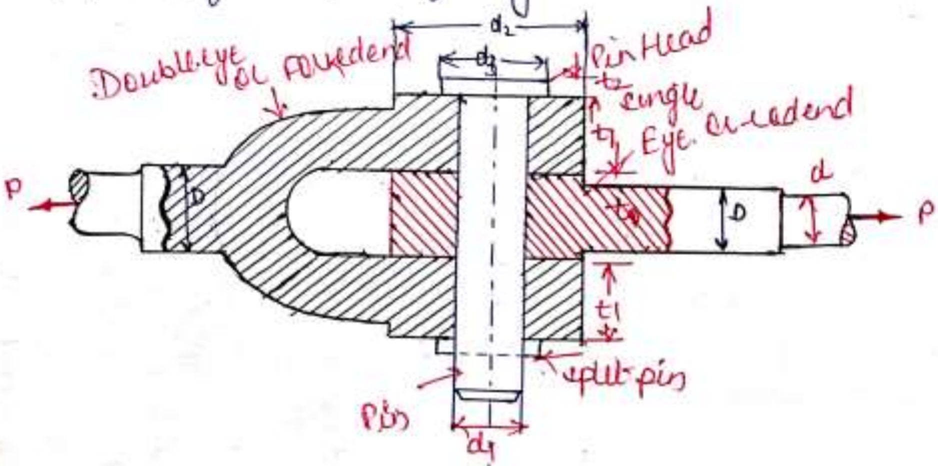
★ Material:-

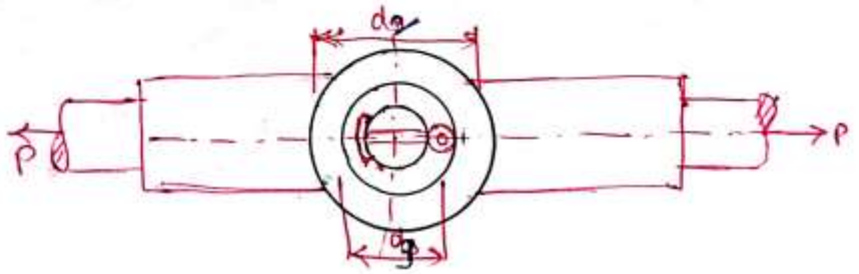
Steel or wrought iron.

* Design of knuckle Joint :-

For the purpose of stress analysis the following assumptions are made.

(i) The rods are subjected to axial tensile force
(ii) The effect of stress concentration due to holes is neglected
(iii) The force is uniformly distributed in various parts





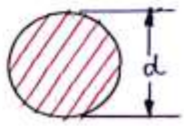
Let

P = Tensile Load acting on the rod
d = Diameter of the rod
d_1 = Diameter of the pin
d_2 = Outer diameter of eye
d_t = Thickness of single eye
t_1 = Thickness of fork
t_2 = Thickness of pin head

σ_t, τ and σ_c = Permissible stresses for the joint material in tension, shear and crushing respectively

(1) Failure of the solid rod in tension :–

Since the rods are subjected to direct tensile load, therefore tensile strength of the rod $= \frac{\pi}{4} d^2 \times \sigma_t$

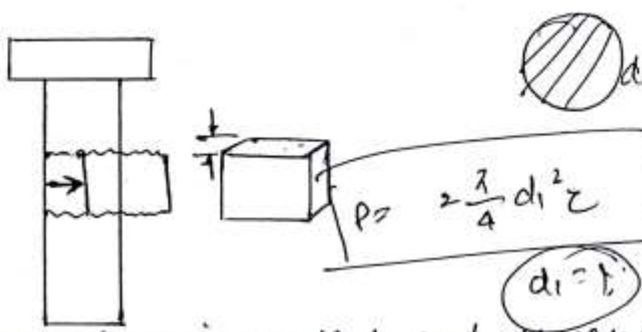


Equating this to the load (P) acting on the rod, we have

$$\boxed{P = \frac{\pi}{4} d^2 \times \sigma_t}$$

From this equation, diameter of the rod (d) is obtained.

(2) Failure of the knuckle pin in shear :–



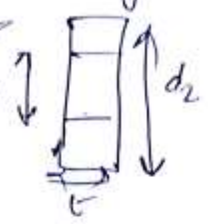
$$P = 2 \frac{\pi}{4} d_1^2 \tau$$

$d_1 = t$

This assumes that there is no slack and clearance between the pin and the fork and hence there is no bending of the pin. But in actual practice, the knuckle pin is loose in forks in order to permit angular movement of one with respect to other

(4) Failure of single eye or rod end in shearing.

$$\boxed{P = 2\frac{(d_2 - d_1)}{2}\, t \times \tau}$$

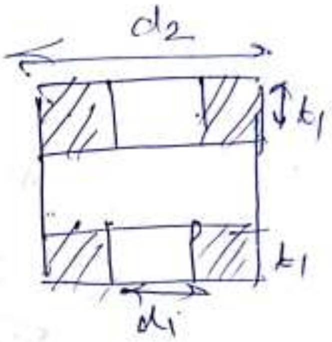


(5) Failure of single eye or rod end in crushing

$$\boxed{P = d_1 \times t \times \sigma_c}$$

(6) Failure of the forked end in tension

$$\boxed{P = (d_2 - d_1) \times 2t_1 \times \sigma_t}$$



(7) Failure of the forked end in shear

$$\boxed{P = \frac{(d_2 - d_1)}{2} \times 4t_1 \times \tau}$$

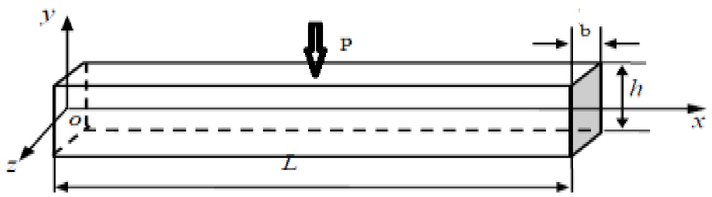
(8) Failure of the forked end crushing.

$$\boxed{P = d_1 \times 2t_1 \times \sigma_c}$$

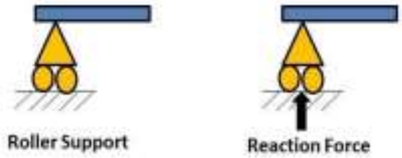
Unit-4: Design of Members in Bending: Beams, levers and laminated springs

Beam

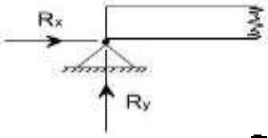
Beam is defined as a structural member which is subjected to transverse loads (i.e. Load acting perpendicular to the beams axis).



Types of Supports



- **Roller Support**
- Vertical Reaction



- **Hinged or Pinned Support**
- Vertical Reaction
- Horizontal Reaction
- **Fixed Support**
- Vertical Reaction
- Horizontal Reaction
- Moment

Types of Beams

According to the condition of equilibrium

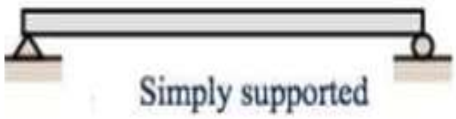
(i) Statically determinate beam

If total number of reactions are equal to numbers of statically equilibrium equations.

Three statically equilibrium equations in case 2-d plane as

$$F_x = 0; F_y = 0; M = 0$$

Example:



(ii) Statically indeterminate beam

If total number of reactions are greater than to numbers of statically equilibrium equations.

According to the geometry

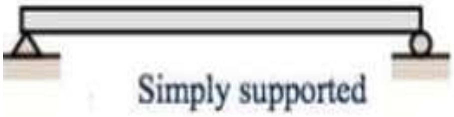
- Straight beam – Beam with straight profile
- Curved beam – Beam with curved profile

According to the cross section

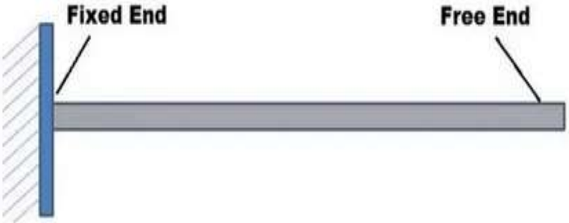
- I-beam – Beam with ‘I’ cross section
- T-beam – Beam with ‘T’ cross section

According to the end supports

Simply supported beams: Beam is supported at both ends



Cantilever beams: A beam that is fixed at one end and free on the other end



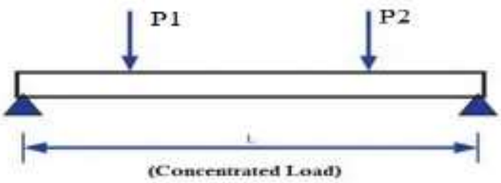
Fixed beams: A beam that is fixed at both ends is called a fixed beam.

Continuous beams: A beam that has more than two supports this kind of beam is called continuous beams.

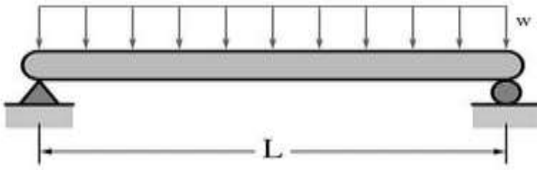
Overhanging beam: A beam that is supported by two points but on the third point is hanging or not support it is called an overhanging beam.

Types of loads on beams

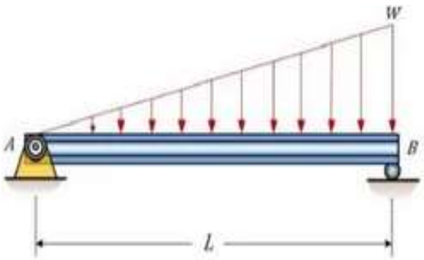
Point load or concentrated load: The point load is defined as a load applied on a single location of the whole span length.



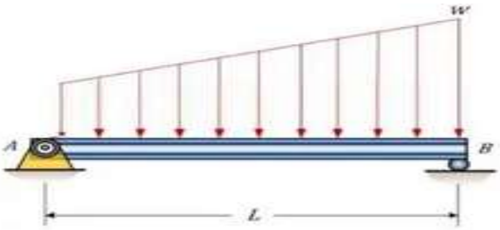
Uniformly distributed load (UDL): The loading magnitude remains the same to the whole span called uniformly distributed load.



Uniformly varying load (UVL): The load whose magnitude is continuously varying throughout the span.



(Triangular Load)

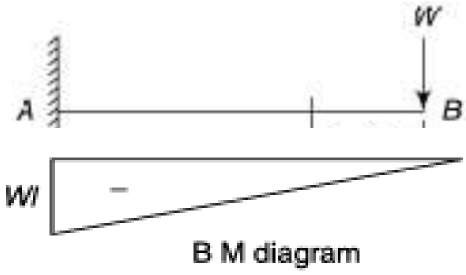


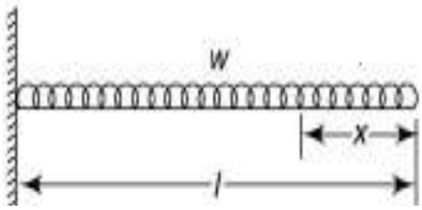
(Trapizoidal Load)

Maximum Bending Moment in Beams:
Taking moment about section

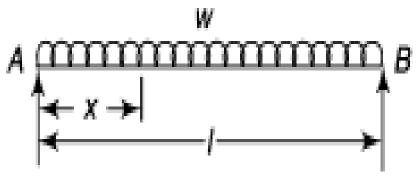
$$M = W \cdot x$$

Maximum Bending Moment at $x = l$
M(max)=Wl

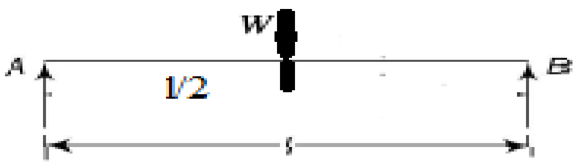




Maximum Bending Moment at $x = l$
M(max)=w$l^2/2$



Maximum Bending Moment at $x = l/2$
M(max)=w$l^2/8$



Maximum Bending Moment at $x = l/2$
M(max)=$wl/4$

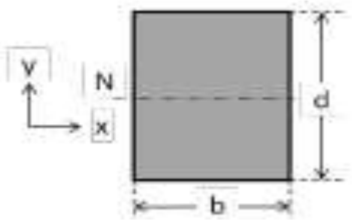
Bending Stress in Beam:
The bending stress at any fibre is given by,

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

The bending stress is maximum in a fibre, which is farthest from the neutral axis.
The distribution of stresses is linear and the stress is proportional to the distance from the neutral axis.

Moment of Inertia of different Sections

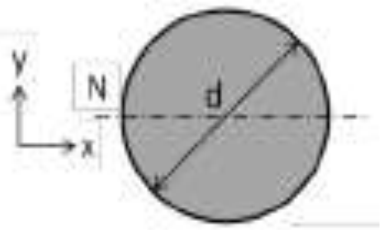
Rectangle



$$I_{xx} = \frac{bd^3}{12}$$

$$I_{yy} = \frac{db^3}{12}$$

Circle



$$I_{xx} = I_{yy} = \frac{\pi d^4}{64}$$

Problem: A beam of rectangular section is to support a load of 20KN uniformly distributed over a span of 3.6 m when beam is simply supported. If the depth is to be twice the breadth, and the stress in beam is not exceed 70 N/mm^2, find the dimensions of the cross section. How could you modify the dimensions with 20 KN of concentrated load is present at center with same breadth and depth ratio.

Solution

Given: w=20KN/m ; l =3.6 m; d=2b; $\sigma_b = 70N/mm2$; P=20 KN

Solution:

Case 1; Beam is subjected to uniformly distributed Load

(i) Maximum Bending Moment:

M=wl^2/8=$20 \times 1000 \times \frac{3.6^2}{8} = 32400\ N - m = 32400 \times 1000$ N-mm

(ii) Cross-section of Beam

Using Bending Equation:

M /I= σ / y

$32400 \times 1000/$(d/2xd^3/12) = 70/(d/2)

d=177.1 mm

b=88.55 mm

Case 2; Beam is subjected to point Load

(i) Maximum Bending Moment:

M=Pl/4=$20 \times 1000 \times \frac{3.6}{4} = 18000\ N - m = 18000 \times 1000$ N-mm

Cross-section of Beam

M /I= σ / y

$18000 \times 1000/$(d/2xd^3/12) = 70/(d/2)

d=145.59 mm

b=72.79 mm

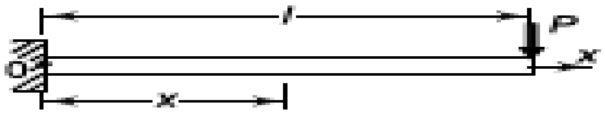
Design of beam based on rigidity:

Design based on rigidity, the induced deflection should be less than the permissible deflection over length. There are some cases of beam are given as

Deflection may be determined from the fundamental equation for the elastic curve of a beam

$$\frac{d^2y}{dx^2} = \frac{M}{EI}$$

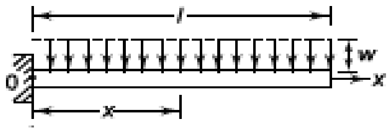
Case 1 Cantilever Beam – End load



$$\delta \text{ at } x = \frac{Px^2}{6EI}(x - 3l)$$

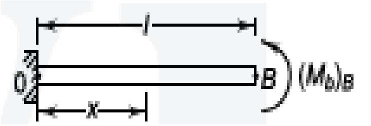
$$\delta_{max} \text{ at } (x = l) = -\frac{Pl^3}{3EI}$$

Case 2 Cantilever Beam—Uniformly distributed load



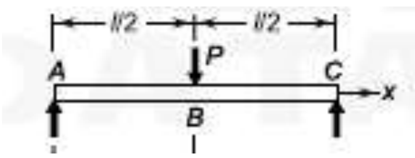
$$\delta_{max} \text{ at } (x = l) = -\frac{wl^4}{8EI}$$

Case 3 Cantilever Beam—Moment load



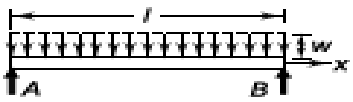
$$\delta_{max} \text{ at } (x = l) = \frac{(M_b)_B l^2}{2EI}$$

Case 4 Simply supported beam—Centre load



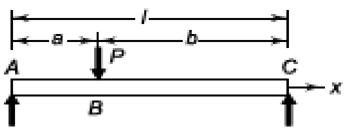
$$\delta_{max} \text{ at } B = -\frac{Pl^3}{48EI}$$

Case 5 Simply supported Beam—Uniformly distributed load



$$\delta_{max} \text{ at } l/2 = -\frac{5wl^4}{384EI}$$

Case 6 Simply supported Beam—Intermediate load



$$\delta \text{ at } x = \frac{Pbx(x^2 + b^2 - l^2)}{6EIl} \quad (0 < x < a)$$

$$\delta \text{ at } x = \frac{Pa(l - x)(x^2 + a^2 - 2lx)}{6EIl} \quad (a < x < l)$$

LEVER

- A lever is a rigid rod or bar capable of turning about a fixed point called fulcrum.
- It is used as a mechanism to lift a load by the application of small effort.
- A lever may be ***straight*** or ***curved***

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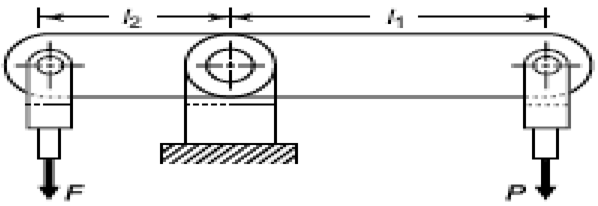
- The forces applied on the lever (or by the lever) may be parallel or inclined to one another.



F is the force produced by the lever
P is the effort required to produce that force.
The perpendicular distance of the line of action of any force from the fulcrum is called the *arm* of the lever. Therefore l_1: effort arm; l_2: load arm

Taking moment of forces about the fulcrum,

$$F \times l_2 = P \times l_1$$

$$\frac{F}{P} = \frac{l_1}{l_2}$$

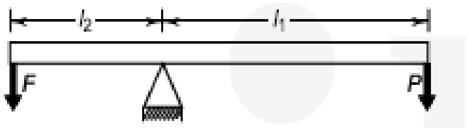
- The ratio of load lifted to effort applied (*F*/*P*) is called as mechanical advantage
- The ratio of effort arm to the load arm $\left(\frac{l_1}{l_2}\right)$ is called as leverage.

Types of levers

There are three types of levers, based on the relative positions of the effort point, the load point and the fulcrum

<u>First Type of Lever:</u>

- The first type of lever, the fulcrum is in between the load and effort.

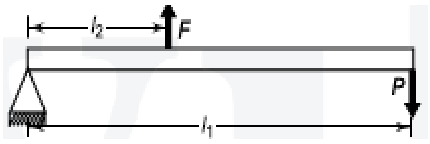


- The effort arm is greater than load arm, therefore mechanical advantage obtained is more than one.

Example: bell crank levers; levers of hand pumps.

<u>Second Type of Lever:</u>

- The second type of lever, the load is in between the fulcrum and effort.



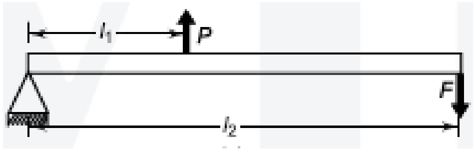
- The effort arm is greater than load arm, therefore mechanical advantage obtained is more than one.

Example: lever-loaded safety valves mounted on the boilers

<u>Third Type of Lever:</u>

- The second type of lever, the effort is in between the fulcrum and load.
- The effort arm is less than load arm, therefore mechanical advantage obtained is less than one.
- This type of lever is not recommended in engineering applications

- Example: A picking fork



Design of Levers

- The length of the lever is decided on the basis of leverage required to exert a given load *F* by means of an effort *P*.
- The cross-section of the lever is designed on the basis of bending stresses.

The design of a lever consists of the following steps:

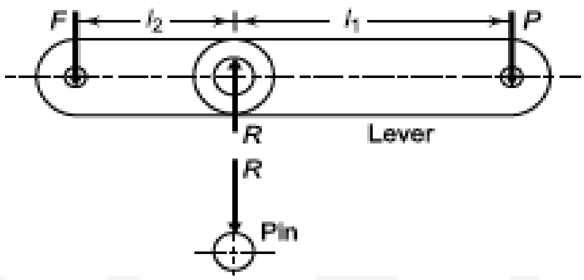
Step I: Force Analysis

- Generally, the load or the force *F* to be exerted by the lever is given.
- The effort required to produce this force is calculated by taking moments about the fulcrum. Therefore,

$$F \times l_2 = P \times l_1$$

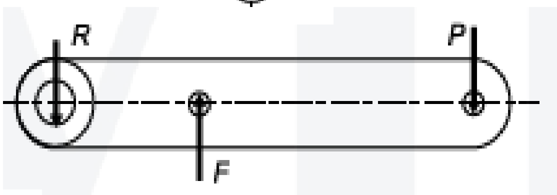
$$P = F\left(\frac{l_2}{l_1}\right)$$

Reactive Forces at Fulcrum Pin



First Type of Lever

$$R = F + P$$



Second Type of Lever:

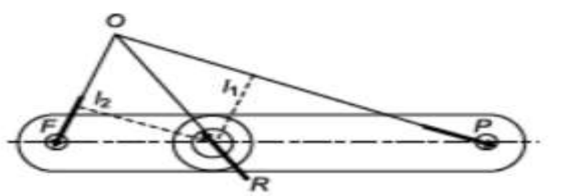
$$F = R + P$$

$$R = F - P$$

Lever with Inclined Forces:

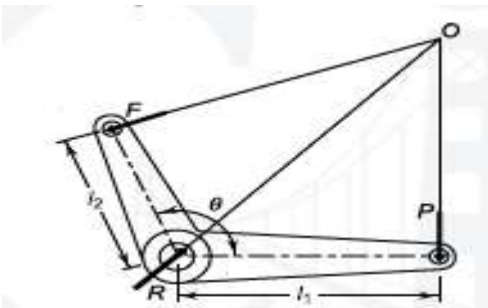
The following rules from statics apply to the reaction *R* at the fulcrum:

- The magnitude of the reaction *R* is equal to the resultant of the load *F* and the effort *P*.
- It is determined by the *parallelogram law* of forces.
- The line of action of the reaction *R* passes through the intersection of *P* and *F*, i.e., the point *O* in Fig. and also through the fulcrum.



Case I:
Force F and P act along lines that are inclined to one another.
Case II (bell-crank lever)
When F and P acts at right angles and the arms are inclined at an angle θ
The reaction R at the fulcrum is given by

$$R = \sqrt{F^2 + P^2 - 2FP\cos\theta}$$



Design of levers

length of lever is decided based on the basis of leverage

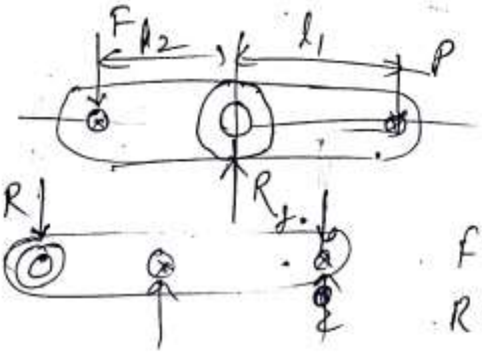
The cross-section of the lever is designed based on the basis of the bending ~~moment~~ stresses.

Step 1: Force Analysis

$R = F + P$ — (1)

$P = F\left(\frac{l_2}{l_1}\right)$

Second type

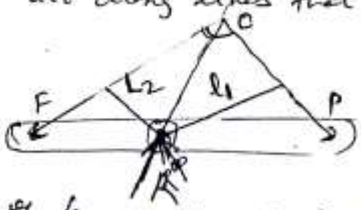


$F = R + P$

$R = F - P$

\

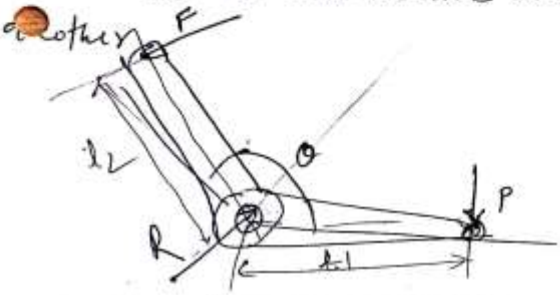
Forces F and P act along lines that are inclined to one-another



(I) Magnitude of the reaction R is equal to the resultant of the load F and effort P. it is determined by the Parallelogram law of forces.

(II) The line of action of reaction R passes through the intersection of P and F.

Bell-crank lever with the arms that are inclined at angle θ with one-another



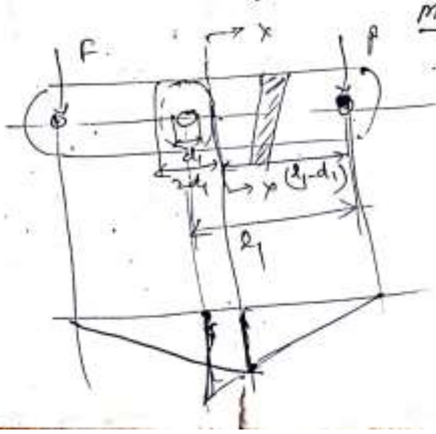
$$R = \sqrt{F^2 + P^2 - 2FP\cos\theta}$$

If $\theta = 90°$ → when the arms of the bell-crank lever are at right angles to one another.

$$R = \sqrt{F^2 + P^2}$$

Design of lever arms

Determine the dimensions of the cross-section of the lever.



maximum bending moment

$M_b = P \times (l_1 - d_1)$

Cross-section of lever can be rectangle elliptical or I-section

for a rectangular cross-section

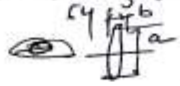
$I = \frac{bd^3}{12}$; $y = \frac{d}{2}$

$\boxed{d = 2b}$

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

for elliptical cross-section

$I = \frac{\pi b a^3}{64}$; $y = \frac{a}{2}$



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Step-3 Design of fulcrum Pin

The dimensions of pin, diameter d_1 and l_1 length are determined by bearing consideration and then checked for shear consideration.

Contact Area $= l_1 d_1$

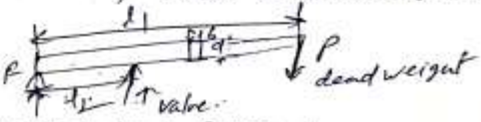
$R = p(l_1 d_1) \rightarrow p$ is the bearing pressure

$\frac{l_1}{d_1} \rightarrow 1$ to $2 \Rightarrow p = \frac{R}{l_1 d_1}$

checked for shearing

$$\tau = \frac{R}{2\left(\frac{\pi}{4} d_1^2\right)}$$

Problem: 3 A lever loaded safety valve is mounted on the boiler to blow-off at a pressure of 1.5 MPa. The effective diameter of the opening of the valve is 50 mm. The distance between the fulcrum and dead weights on the lever is 1000 mm. The distance between the fulcrum and the pin connecting the valve spindle to the lever is 100mm. The lever and the pin are made of plain carbon steel. 30C8 ($S_{yt} = 400\frac{N}{mm^2}$) and factor of safety is 5. The permissible bearing pressure at the pins in the lever is $25\frac{N}{mm^2}$. lever has a rectangular cross-section and ratio of width to thickness is 3. Design a suitable lever for the safety valve.



Given $S_{yt} = 400\ N/mm^2$, $FS = 5$

for valve $\Rightarrow d = 50mm$ $P_v = 1.5 MPa$

lever $l_1 = 1000\ mm$, $l_2 = 100\ mm$; $\frac{d}{b} = 3$

for $p_b = 25\frac{N}{mm^2}$

Permissible stress

$\sigma_t = \frac{S_{yt}}{FS} = \frac{400}{5} = 80\frac{N}{mm^2}$

$\tau = \frac{S_{ys}}{FS} = \frac{0.5 S_{yt}}{5} = 40\frac{N}{mm^2}$

Force Analysis:

$F = \frac{\pi}{4} d^2 \times P_v = \frac{\pi}{4} \times (50)^2 \times (1.5) = 2945.24 N$

Taking moment about fulcrum

$F \times l_2 = P \times l_1$

$P = 294.52 N$

$F = R + P \Rightarrow R = 2945.24 - 294.52 = 2650.72 N$

Diameter and length of pin: $l_1 = d_1$

$F = p(l_1 d_1) \Rightarrow d_1 = 10.85$ or $12mm$

$l_1 = d_1 = 12mm$

$\therefore \tau = \frac{F}{2\left(\frac{\pi}{4} d_1^2\right)} = 13.02\frac{N}{mm^2}$

$M_b = 294.52(1000 - 100) = 265068\ N\text{-}mm$

$d = 36$

$b = 13.01795 mm$

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Problem

A right angled bell-crank lever is to be designed to raise of load 5 kN at the short arm end. The lengths of short and long arms are 100 mm and 450 mm respectively. The lever and the pins are made of steel 30C8 ($S_{yt} = S_{yc} = 400$ MPa) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm². The lever has a rectangular cross-section and ratio of width to thickness is 3 : 1. The length to diameter ratio of the fulcrum pin is 1.25 : 1. Calculate

(i) Diameter and length of the fulcrum pin

(ii) Shear stress in the pin

(iii) Dimensions of the boss of the lever at the fulcrum

(iv) Dimension of cross-section of the lever

Given →

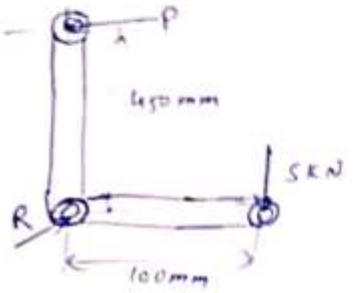
$F = 5 \times 1000 = 5000\ N$

$F_s = 5$

$S_{yt} = 400\ MPa$

$l_1 = 450\ mm;\ l_2 = 100\ mm$

$p = 10\ MPa \quad \frac{l_f}{d_1} = 1.25;\ \frac{d}{b} = 3$



Permissible stress :

$\sigma_t = \frac{400}{5} = 80\ MPa$

$\tau_p = \frac{0.5 \times 400}{5} = 40\ MPa$

Force Analysis →

$P \times 450 = 5000 \times 100$

$\Rightarrow P = \frac{5000 \times 100}{450} = 1111.11\ N$

$R = \sqrt{F^2 + P^2 - 2FP\cos\theta}$, $\theta = 90$ $\Rightarrow R = \sqrt{F^2 + P^2} = \sqrt{(5000)^2 + (1111.11)^2}$

$= 5121.97\ N$

Dimension and length of fulcrum pin :

$R = p_b (l_f \times d_1) \Rightarrow 5121.97 = 10 \times (d_1 \times 1.25 d_1) \Rightarrow d_1 = 20.24\ mm$

$l_p = 1.25 \times 20.24 = 25.30\ mm$

Shear stress in pin :→

$\tau = \frac{R}{2\left(\frac{\pi}{4} d_1^2\right)} = \frac{5121.97}{2 \times \left(\frac{\pi}{4} \times 20.24\right)^2} = 7.96\ \frac{N}{mm^2}$ Ans

Dimension of boss → inner diameter (d_1) = 20.24 mm ≃ 21 mm

outer diameter (d_o) = 21 × 2 = 42 mm

Dimension of cross-section of lever

$d = 3b, \quad M_b = 5000 \times 100$

$\sigma_b = \frac{M_b y}{I} \Rightarrow 80 = \frac{(5000 \times 100) \times (1.5 b)}{\frac{1}{12} \times b \times (3b)^3} \Rightarrow b = 16.09\ mm$

$d = 3b = 3 \times 16.09 = 48.27\ mm$ Ans

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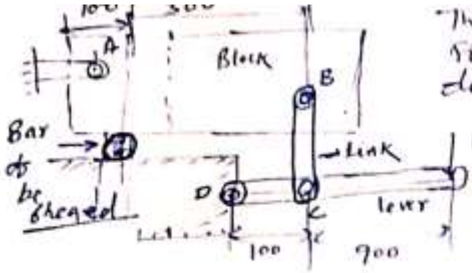


The mechanism of a bench-shearing machine is shown in figure. It is used to shear mild steel bar up to 6.25 mm diameter. The ultimate shear strength of the material is 350 N/mm². The link, lever and pins at B, C and D are made of FeE 250 and the factor of safety is 5. The pins at B, C and D are identical and their length to diameter ratio is 1.25. The permissible bearing pressure at pins is 10 N/mm². The link has circular cross-section. The cross-section of lever is rectangular and the ratio of width to thickness is 2. Calculate the
(I) diameter of pins at B, C, and D
(II) Diameter of link
(III) Dimensions of cross-section of the lever

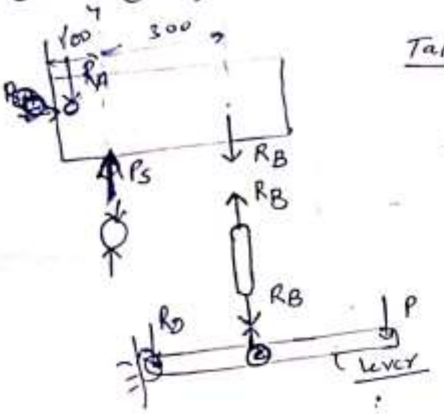
Permissible stress:-

$\sigma_t = \frac{250}{5} = 50 \frac{N}{mm^2}$

$\tau = 0.5 \times 50 = 25 \frac{N}{mm^2}$

Force Analysis:-

$P_s = \frac{\pi}{4}(6.25)^2 \times 350 = 10737.87\ N$



Taking moment about 'A'

$P_s \times 100 = R_B \times 300$

$\Rightarrow R_B = 2684.47\ N$

$R_A + R_B = P_s$

$R_A = 8053.4\ N$

$P \times 1000 = R_B \times 100$

$P = 268.45\ N$

$R_D = R_B - P = 2416.02\ N$

Diameter of Pins:-

$R_B = P_b \times (L_p d_1) \Rightarrow 2684.47 = 10 \times (d_1 \times 1.25 d_1) \Rightarrow \frac{2684.47}{12.5} = d_1^2 \Rightarrow d_1 = 14.65\ mm$
$= 15\ mm$

$L_p = 18.75\ mm$
$\simeq 20\ mm$

$\tau = \frac{R_B}{2(\frac{\pi}{4}d^2)} = \frac{2684.47}{2 \times \{\frac{\pi}{4} \times (15)^2\}} = 7.60 \frac{N}{mm^2}$

Diameter of link

$\sigma_t = \frac{R_B}{\frac{\pi}{4}d^2} \Rightarrow 50 = \frac{2684.47}{\frac{\pi}{4}d^2} \Rightarrow d = 8.27\ mm$ or $10\ mm$

Dimensions of lever:

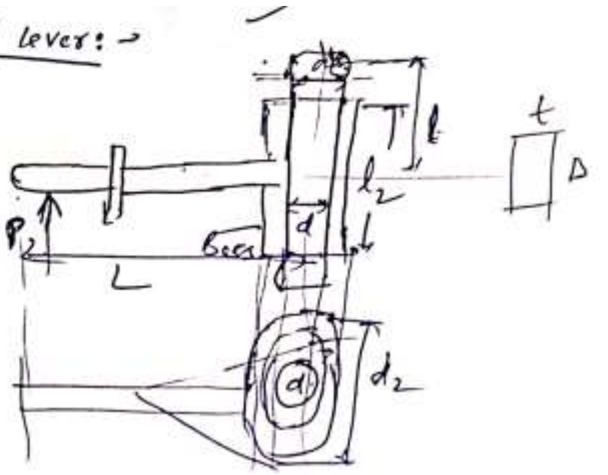
$M_b = P \times 900 = 268.47 \times 900 = 241605\ N\text{-}mm$

$d = 2b \quad I = \frac{bd^3}{12} = \frac{b(2b)^3}{12} = \frac{2}{3}b^4\ mm^4$

$y = \frac{d}{2} = b \quad \sigma_b = \frac{M_b y}{I} = \frac{241605 \times b}{\frac{2b^4}{3}} \Rightarrow b = 19.35\ mm \simeq 20\ mm$

$d = 2b = 2 \times 20 = 40\ mm$

Hand lever: →



Diameter of shaft

$$T = PL \quad \text{———} \quad (1)$$

$$T = \frac{\pi}{16} d^2 \tau \quad - \quad (11)$$

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

Diameter of boss (d_2)

$$d_2 = 1.6d$$

Thickness of boss $t_2 = 0.3$

$$l_2 = 1.5d.$$

Design of lever,

$$M_b = PL$$

Design of key: →

$$w_K = \frac{d}{4}; \; t_K = \frac{d}{4}; \; t_K = \frac{2T}{d \cdot w_K \tau}$$

Diameter

$t = ?$

$B = ?$

d_1 → diameter of

$$M = Pl$$

$$T = PL$$

$$\tau_{max} = \frac{16}{\pi d_1^3}\sqrt{T^2 + m^2}$$

$$d_1^3 = \frac{16}{\pi \tau_{max}}\left(\sqrt{T^2 + m^2}\right.$$

$$I = \frac{tB^3}{12}$$

$$\sigma_b = \frac{My}{I} = \frac{PL \times B/2}{\frac{tB^3}{12}} = \frac{6PL}{tB^2}$$

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Springs:→ A spring is defined as an elastic machine element, which deflects under the action of the load and return to its original shape when the load is removed.

The functions of springs:

(i) springs are used to absorb shocks and vibrations e.g vehicle suspension
(ii) springs are used to store energy e.g. clocks
(iii) springs are used to measure force. e.g (balancer of weight)
(iv) springs are used to apply force and control motion. e.g. cam and follower, valve mechanism

Types of springs

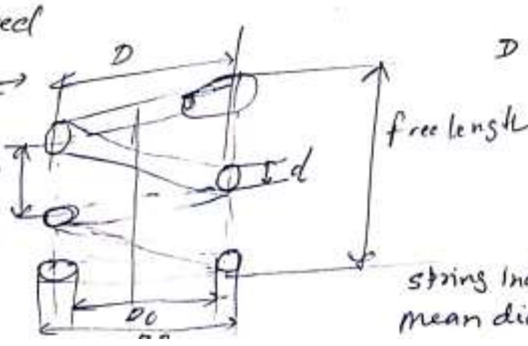
The springs are classified based on shapes
(i) Helical springs
(ii) multi-leaf or laminated spring

Spring materials

The selection of material for the spring wire depends upon the following factors
(i) the load acting on the springs
(ii) The range of stress through which the springs operates
(iii) limitations on the mass and volume of spring
(iv) The expected fatigue life
(v) The environmental conditions
(vi) deformation of the spring.

material used for wire of spring
(i) Patented and cold-drawn steel wires
(ii) oil-hardened and tempered spring steel wires
(iii) oil-hardened and tempered steel (alloyed)
(iv) stainless steel

Helical spring:→



$D = \frac{D_i + D_o}{2}$

d :- wire diameter of spring (mm)
D_i : inside diameter of spring coil
D_o - outside diameter of spring coil
D : mean diameter of coil

spring index (c) is defined as the ratio of mean diameter to wire diameter. $\boxed{C = \frac{D}{d}}$

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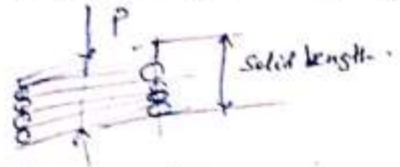
Spring index indicates the relative sharpness of the curvature of the coil.
low spring index means high sharpness of curvature.

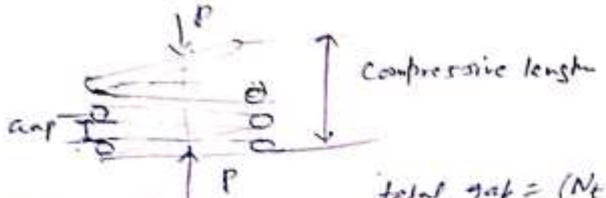
$C \rightarrow 4$ to 12 → generally used for designing of helical spring.

(i) Solid length : Solid length is defined as the axial length of the spring which is so compressed that the adjacent coils touch each other.

Solid length = $N_t d$ → N_t → total Number of coils.



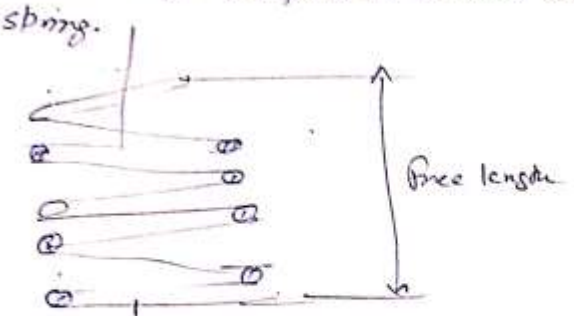
(ii) Compressed length : compressed length is defined as the axial length of the spring is subjected to maximum compressive force.



gap → 1 to 2 mm

total gap = $(N_t - 1) \times$ gap between adjacent coils

(iii) Free length: free length is defined as the axial length of an unloaded helical compression spring.



Free length = compressed length + δ = Solid length + total axial gap

pitch of coil is defined as the axial distance between adjacent coil in uncompressed state of spring. it is denoted by p. It is given by

$$p = \frac{\text{free length}}{N_t - 1}$$

stiffness of spring (k) is defined as the force required to develop unit deflection.

$$K = \frac{P}{\delta} \quad \frac{N}{mm}$$

rate of spring
spring constant
gradient of spring.

Spring coils: → Active coils → support the external force and deflect under the action of force
in-active coils • do not support the external force and do not deflect under the action of an external force.

In-active coil = $N_t - N$ → N: Number of active coil.

Type of end	N (active coil)
Plain ends	N_t
Square ends	$N_t - 2$
Square ends ground	$N_t - 2$

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$\tau = 0.5 S_{ut}$

Wahl factor

$$\tau = K\left(\frac{8PD}{\pi d^3}\right) \quad \text{where } K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

$$\delta = \frac{8PD^3N}{Gd^4} \qquad C = \frac{D}{d}$$

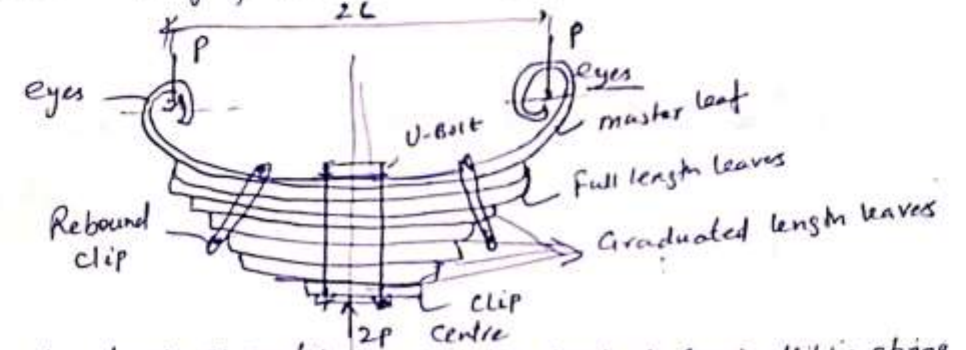
Multi-Leaf or laminated spring:

Multi-leaf springs are widely used for the suspension of heavy vehicles such as (Truck, railway wagons). A multi-leaf spring consists of a series of flat plates, known as leaves of the spring, usually of semi-elliptical shape.

The leaf at the top has maximum length known as master leaf.

The leaves are held together by means of two U-bolts and a centre clip.

Rebound clips are provided to keep the leaves in the alignment and prevent lateral shifting of the leaves during operation.



L : length of cantilever or half the length of semi-elliptic spring

P : Force applied at the end of the spring (N)

P_f = Portion of P taken by the extra full length leaves

P_g = " of P taken by the graduated length leaves (N)

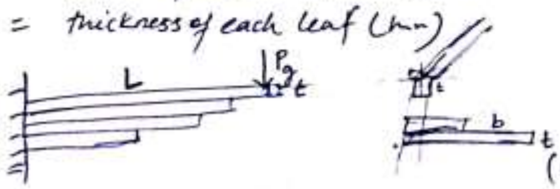
n_f = Number of extra full length leaves

n_g Number of graduated-length leaves

n = total Number of leaf

b = width of each leaf (mm)

t = thickness of each leaf (mm)



$M_b = P_g \times L$

$y = t/2, \quad I = \frac{bt^3}{12} \Rightarrow I_{total} = \frac{n_g b t^3}{12}$

$$(\sigma_b)_g = \frac{6 P_g L}{n_g b t^2}$$

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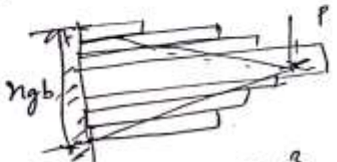
graduated leeves is considered as the triangular plate of thickness t and width of $(n_g b)$



$$\delta_g = \frac{P_g L^3}{2EI} = \frac{P_g L^3}{2E\left(\frac{1}{12}(n_g b)\, t^3\right)} \Rightarrow \delta_g = \frac{6 P_g L^3}{E n_g b t^3}$$

Full length leaves is considered as rectangular plate of thickness t and width $(n_f b)$

$$(\sigma_b)_f = \frac{(P_f L)\cdot t/2}{\frac{1}{12}(n_f b)\, t^3} = \frac{6 P_f L}{n_f b t^2}$$

deflection at point load

$$\delta_f = \frac{PL^3}{3EI} = \frac{P_f L^3}{3E\left\{\frac{1}{12}(n_f b)\, t^3\right\}}$$

$$\boxed{\delta_f = \frac{4 P_f L^3}{E n_f b t^3}}$$

The deflection of full length leaves is equal to the deflection of graduated-length leaves.

$$\delta_g = \delta_f$$

$$\frac{6 P_g L^3}{E n_g \cancel{b t^3}} = \frac{4 P_f L^3}{E n_f \cancel{b t^3}}$$

$$P_g + P_f = P \quad \text{---} \; (II)$$

$$\frac{P_g}{P_f} = \frac{2 n_g}{3 n_f} \quad \text{---} \; (I)$$

$$P_f = \frac{3 n_f P}{(3 n_f + 2 n_g)} \quad \text{---} \; (III) \qquad P_g = \frac{2 n_g P}{(3 n_f + 2 n_g)}$$

$$(\sigma_b)_g = \frac{12 PL}{(3 n_f + 2 n_g)\, b t^2} \; ; \; (\sigma_b)_f = \frac{18 PL}{(3 n_f + 2 n_g)\, b t^2}$$

$$\delta = \frac{12 PL^3}{E b t^3 (3 n_f + 2 n_g)}$$

Bending stresses in full length leaves are 50% more than those in graduated ~~leaves~~ length leaves.

Material of leaves : These are made of steels, $55Si_2Mn90$, 50Cr1 or 50Cr1V23

Nipping of leaf springs

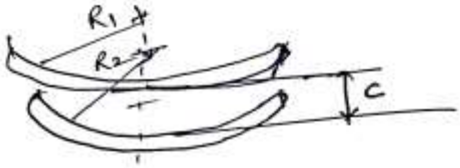
Equalising the stresses in different leaves is the pre-stress the spring.

Method of

~~in this method~~ the ~~Process~~ stressing is achieved by the bending the leaves to different radii of curvature, before they are assembled with the centre clip.

The full length leaf is given a greater radius of curvature than the adjacent leaf

~~The~~ initial gap c between the extra full length leaf and the graduated length leaf before the the assembly, is called a Nip.



Pre-stressing is achieved by a difference in radii of curvature, is known as "nipping"

Stress equalisation due to prestressing

$$(\sigma_b)_g = (\sigma_b)_f$$

$$\Rightarrow \frac{6P_g L}{n_g b t^2} = \frac{6P_f L}{n_f b t^2}$$

$$\Rightarrow \frac{P_g}{P_f} = \frac{n_g}{n_f}$$

$$P_g + P_f = P$$

$$\Rightarrow P_g = \frac{n_g P}{n} \quad \text{—(iv)} \qquad \Rightarrow n = n_f + n_g$$

$$P_f = \frac{n_f P}{n} \quad \text{— (v)}$$

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$$c = \delta_g - \delta_f$$

$$c = \frac{6P_g L^3}{E n_g b t^3} - \frac{4 P_f L^3}{E n_f b t^3}$$

substituting the P_g and P_f

$$\boxed{c = \frac{2PL^3}{Enbt^3}}$$

The initial Preload P_i reauired to close the gap c between the extra-full length leaves and graduated length leaves is determined by considering the initial deflection of leaves.

$$c = (\delta_g)_i + (\delta_f)_i$$

$$\frac{2PL^3}{Enbt^3} = \frac{6P(P_i/2)L^3}{E n_g b t^3} + \frac{4(P_i/2)L^3}{E n_f b t^3}$$

$$\text{or } P_i = \frac{2 n_g n_f P}{n(3n_f + 2n_g)}$$

Resultant stress in Extra-full length leaves

$$(\sigma_b)_f = \frac{6L(P_f - P_i/2)}{n_f b t^2}$$

substituting P_f from eq^n (III) and P_i

$$(\sigma_b)_f = \frac{6PL}{nbt^2}$$

$$\boxed{\sigma_b = \frac{6PL}{nbt^2}}$$ Stresses are eual in all leaves.

Problem A semi-elliptic leaf spring used for automobile suspension consists of three extra full length leaves and 15 graduated length leaves including master leaf. The centre to centre distance between two eyes of spring is 1m. The maximum force that can act on the spring is 75 kN. for each leaf, the ratio of width to thickness is 9:1. The material of elasticity of the leaf material is 207000 N/mm². The leaves are pre-stressed in such way then the free is maximum. The stresses induced in all leaves are same and eaual to 450 N/mm². Determine (I) width and thickness of the leaves (108 mm) (12 mm) (II) the initial gap ($c = 13.48$ mm) (III) initial pre-load reauire to close gap c. ($P_i = 4807.69$ N)

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A Semi-elliptic multi-leaf spring is used for the suspension of the rear axle of a truck. it consists of two extra length leaves and ten graduated length leaves including the master leaf. the centre to centre distance between the spring eyes is 1.2m. the leaves ~~made~~ are made of steel 55 Si 2 Mo 90 ($S_{yt} = 1500 \frac{N}{mm^2}$ and $E = 207000\ N/mm^2$) and Factor of safety is 2.5. The spring is to be designed for a maximum force of 30kN. The leaves are pre-stressed so as to equalize the stresses in all leaves. Determine, Assume $b = 60$mm

(I) The cross-Section of leaves

(II) The deflection at the end of the spring.

$2P = 30\,kN$; $2L = 1.2m$ $N_f = 2$ $N_g = 10$

$E = 207000\ \frac{N}{mm^2}$ $S_{yt} = 1500\ N/mm^2$

$FS = 2.5$

$P = 15{,}000\ N$

$L = 600mm$

$$\sigma_b = \frac{S_{yt}}{FS} = \frac{1500}{2.5} = 600\ \frac{N}{mm^2}$$

$$\sigma_b = \frac{6PL}{nbt^2} \Rightarrow 600 = \frac{6 \times 15000 \times 600}{12 \times bt^2}$$

$$bt^2 = 7500\ mm^3$$

$b = 60mm$

$$t^2 = \frac{7500}{60} \Rightarrow t = 12mm$$

$$\delta = \frac{12PL^3}{Ebt^3(3N_f + 2N_g)} = \underline{69.68mm}$$

Note :-

1. If ~~cent~~ Band is given as then effective length l - Band wide

$2L_1 = 2L - l$

$$\boxed{2L_1 = \frac{2L - l}{2}}$$

2. it is not initially stress.

$\sigma_f = \sigma_g$ $\sigma_f\ max = \frac{18PL}{bt^2(2n_g + 3n_f)}$

$$\delta_f = \delta_g = \frac{12PL^3}{Ebt^3(2N_g + 3N_f)}$$

3. total depth or thickness

total depth or thickness $= nt \Rightarrow$

4. length of each spring:

Length of smallest leaf $= \frac{\text{Effective length}}{n-1} \times 1 + \text{ineffective length}$

length of 2nd leaf $= \frac{\text{Effective length}}{n-1} \times 2 + \text{ineffective length}$

length of 3rd leaf $= \frac{\text{effective length}}{n-1} \times 3 + \text{ineffective length}$

length of 6th leaf $= \frac{\text{effective length}}{n-1} \times 6 + \text{ineffective length}$

length of 7th spring $= 2L_1 + \pi(d+b) = 2L_1 + \pi d + \pi t$ Ans

5. diameter of eye: ⇒

$l_{total} = b + 2 \times 2 = b + 4$ mm

$M_b = \frac{W l_{total}}{4}$

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

$\sigma_b = \frac{M}{Z} = \frac{\cancel{W l_{total}}}{4}$

$\Rightarrow d^3 = ?$ $d = ?$

Check for induced shear stress

$P = 2 \times \frac{\pi}{4} d^2 \times \tau \Rightarrow \boxed{\tau \leq \tau_p}$ Ans

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Unit 5: Design of Shaft, Keys and Couplings

Transmission Shaft :→

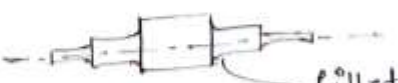
→ A Shaft is a rotating m/c element which is used to transmit power from one place to another place.

→ Shaft are of circular in x s/c.

→ Shaft may be Solid or Hollow depending upon application & requirement.

→ To transfer the power from one shaft to another various mechanical members such as Gears, pulley etc. are mounted on it

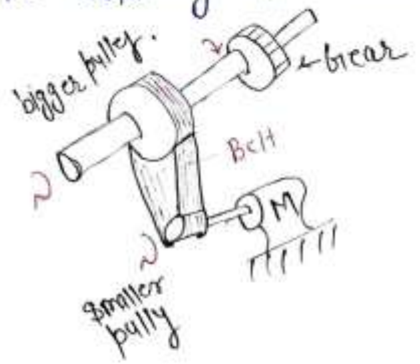
→ The Shaft is stepped with max. diameter in the middle portion and minimum dia at the two ends where bearings are mounted. these steps are provided for positioning transmission elements like gears, pully & bearings.



fillet radius (to reduce Stress Concentration)

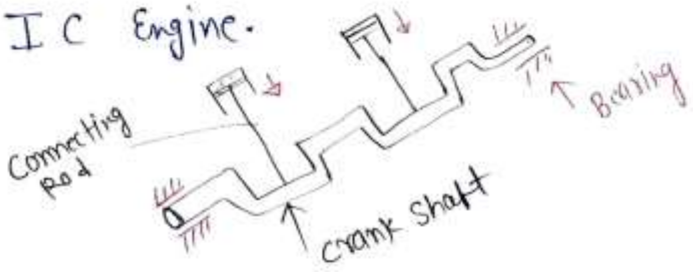
Types of Shafts :-

(i) Transmission Shaft :- They receives power from one end (source) & deliver power at other end (i.e. absorbing m/c element).



eg:- Counter Shaft
line Shaft &
all factory Shaft.

(ii) Machine shaft :→ They are an integral part of a machine it self. eg:- crank shaft in I C Engine.



Matrial :-

Ordinary transmission shafts are made of medium carbon steels with a carbon content from 0.15 to .40 percent such as 30CB or 40CB.

+ where greater strength is required High carbon steel such as 45CB or 50CB or alloy steel are used.

* Design of a shaft :-

(i) Strength basis :-

(a) when shaft is subjected to pure axial tensile force then tensile stress induced in the shaft

$$\sigma_t = \frac{P}{\left(\frac{\pi}{4}d^2\right)}$$

(b) when sub. to pure bending

$$\sigma_b = \frac{M_b \cdot y}{I} \quad \text{where } y = d/2, \ I = \frac{\pi}{64}d^4$$

for solid circular x-s/c.

(c) when sub. to pure torsional moment

$$\tau = \frac{T \cdot r}{J} \quad \text{wher } r = d/2$$

$$J = \frac{\pi}{32}d^4$$

(d) But when sub. to combination of these stresses. then shaft are designed on the basis of Theory of failure

(i) Maximum principal Stress Theory :-

Acc. to M.P.S.T

$$\underbrace{\frac{1}{2}\left[M \pm \sqrt{M^2+T^2}\right]}_{M_e} \leq \frac{\pi}{32} d^3 \left[(\sigma_t)_{per} \text{ or } \frac{S_{yt}}{N}\right]$$

$$M_e \leq (\sigma_t)_{per} \cdot Z_{N.A.}$$

→ Equivalent Bending moment :- is the Bending moment which when acting alone will produce the same bending stresses (tensile & compressive) in the shaft as under the combined action of bending moment (M) and torsional moment (T).

(ii) Maximum Shear Stress Theory :-

Acc. to M.S.S.T.

$$\underbrace{\sqrt{M^2+T^2}}_{T_e} \leq \frac{\pi}{16} d^3 \left[(\tau_{per}) \text{ or } \frac{S_{ys}}{N}\right]$$

$$T_e \leq Z_p \cdot \tau_{per} \quad \left[\because S_{ys} = \frac{1}{2} S_{yt}\right]$$

→ equivalent torsional moment

The equivalent torsional moment is the torsional moment which when acting along will produce the same torsional shear stress in the shaft as under the combined action of B.M. (M) & T.M. (T)

NOTE :→ M.P.S.T. is used for brittle material. and M.S.S.T is used for ductile material. The M.S.S.T is applicable for shaft design because shafts are made of ductile material.

(ii) Shaft design on torsional rigidity basis :-

A transmission shaft is said to be rigid on the basis of torsional rigidity, if it does not twist too much under the action of an external torque. Similarly lateral rigidity when deflection not exceed under the external forces and B.M.

eg:- Spindles are designed on torsional rigidity basis i.e. on the basis of permissible angle of twist per metre length of shaft.

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The angle of twist (θ_r) is given by (in rad.)

$$\theta_r = \frac{T \cdot l}{G \cdot J}$$

Convert, θ_r from radian to degree (θ)

$$\theta = \frac{180}{\pi} \times \frac{Tl}{GJ} \qquad \text{where } J = \frac{\pi}{32} d^4$$

d = dia of shaft

G (modulus of rigidity) for steel is 79300 N/mm² or approximately 80 kN/mm².

ASME code for shaft design :-

According to this code

$$\tau_{max} = 0.30\, S_{yt} \text{ or } 0.18\, S_{ut} \text{ [whichever is min]}$$

↳ without keyways

if keyways are present the above values are to be reduced by 25 percent

i.e. $\tau_{max} = 0.30 \times 0.75\, S_{yt}$ or $0.18 \times 0.75\, S_{ut}$ [which ever is minimum]

- Shafts subjected to fluctuating loads :-

In actual practice, the shaft are subjected to fluctuating torque and Bending moments. In order to design such shafts like line shafts, counter shafts the combined shock factor & fatigue factors must be taken into account, for the computed twisting moment (T) and B.M. (M).

Thus for a shaft subjected to combined bending and torsion the equivalent twisting moment

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

& equivalent B.M.

$$M_e = \frac{1}{2}\left[K_m \cdot M \pm \sqrt{(K_m \cdot M)^2 + (K_t \cdot T)^2}\right]$$

where

K_m = Combined shock and fatigue factor for bending

K_t = Combined shock and fatigue factor for torsion.

Design of Hollow Shaft :-

Solid Shaft
X-S/C.

Hollow Shaft X-S/C.

Solid shaft:

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

$$I = \frac{\pi}{64} d^4$$

$$J = \frac{\pi}{32} d^4$$

$$Z_{NA} = \frac{I}{d/2} = \frac{\pi}{32} d^3$$

$$Z_p = \frac{J}{d/2} = \frac{\pi}{16} d^3$$

Hollow shaft:

$$A = \frac{\pi}{4}(d_o^2 - d_i^2) \text{ or } \frac{\pi}{4} d_o^2 (1-k^2)$$

Where $k = d_i/d_o$

$$I = \frac{\pi}{64} d_o^4 (1-k^4)$$

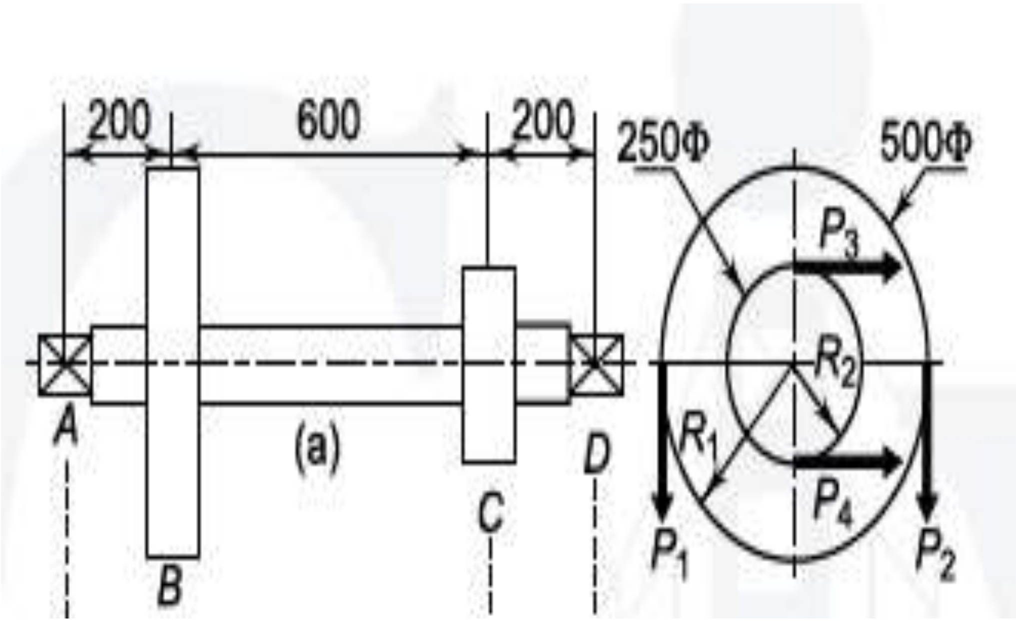
$$J = \frac{\pi}{32} d_o^4 (1-k^4)$$

$$Z_{NA} = \frac{\pi}{32} d_o^3 (1-k^4)$$

$$Z_p = \frac{\pi}{16} d_o^3 (1-k^4)$$

Question :-*The layout of a transmission shaft carrying two pulleys B and C and supported on bearings A and D is shown in Fig. Power is supplied to the shaft by means of a vertical belt on the pulley B, which is then transmitted to the pulley C carrying a horizontal belt. The maximum tension in the belt on the pulley B is 2.5 kN. The angle of wrap for both the pulleys is 180° and the coefficient of friction is 0.24. The shaft is made of plain carbon steel 30C8 (Syt = 400 N/mm2) and the factor of safety is 3.*

Determine the shaft diameter on strength basis transmission shafts



ALL DIMENSIONS ARE IN MM

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Sol^n:-

Given:-

$P_1 = 2.5\,kN$, $FoS = 3$

$\theta = 180^\circ$, $\mu = 0.24$, $S_{yt} = 400\,N/mm^2$

$d = ?$ on strength basis.

Step-I permissible shear stress

$$\tau_{per} = \frac{S_{ys}}{FoS} \text{ or } \frac{.5\,S_{yt}}{FoS} = \frac{.5 \times 400}{3}$$

$$66.67\,N/mm^2$$

Step-II calculation of torsional moment

$$\frac{P_1}{P_2} = e^{\mu\theta} \Rightarrow \frac{2500}{P_2} = e^{.24 \times \pi}$$

$$P_2 = 1176.22\,N$$

so, the total torque supplied to the shaft is (Pully B)

$$T = (P_1 - P_2)R_1$$

$$= (2500 - 1176.22) \times 250 = 330945\,N\text{-}mm$$

and, the same torsional moment is transferred to the pully C i.e. $T = (P_3 - P_4)R_2$

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$$(P_3 - P_4) \times 125 = 330945 \quad -(i)$$

and $\frac{P_3}{P_4} = e^{\mu\theta}$

$$\frac{P_3}{P_4} = e^{.24 \times \pi} \Rightarrow P_3 = P_4 \times e^{.24\pi} \quad -(ii)$$

Put P_3 in eqn (i)

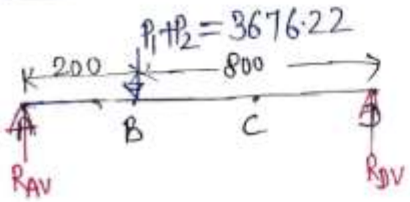
$$P_4 \left(e^{.24\pi} - 1 \right) \times 125 = 330945$$

$$P_4 = 2352.45\ N \quad \text{put it in (ii)}$$

so $P_3 = 5000\ N$

Step III → Calculation of Bending moment

(a) due to vertical forces



$R_{AV} + R_{DV} = 3676.22 \quad (\Sigma V = 0)$

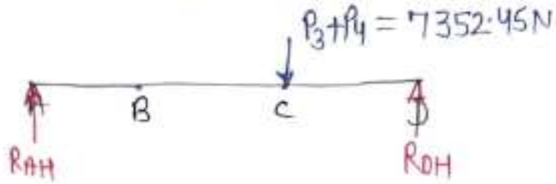
$3676.22 \times 200 = R_{DV} \times 1000$ (taking moment about pt A)

$R_{DV} = 735.24\ N$, $R_{AV} = 2941\ N$

$M_{Bv} = R_{AV} \times 200 = 588200\ N\cdot mm \qquad M_{AV} = 0$

$M_{Cv} = R_{DV} \times 200 = 147048\ N\cdot mm \qquad M_{DV} = 0$

(b) due to horizontal forces



$$R_{AH} + R_{DH} = 7352.45$$

$$7352.45 \times 800 = R_{DH} \times 1000 \Rightarrow R_{DH} = 29409.8$$

$$R_{DH} = 5882\ N$$

$$R_{AH} = 1470.45N$$

$$M_{B_V} = R_{AH} \times 200 = 294090\ N\cdot mm$$

Resultant B.M. $\quad M_{CV} = R_{DH} \times 200 = 1176400\ N\cdot mm$

$$(M)_B = \sqrt{(M_{BH})^2 + (M_{BV})^2}$$

$$= 657624\ N\cdot mm$$

$$M_C = \sqrt{(M_{CH})^2 + (M_{CV})^2}$$

$$= 1185554.75\ N\cdot mm$$

By B.M.D. & T.M.D. , the stresses are max. at pt. C.

Step- IV

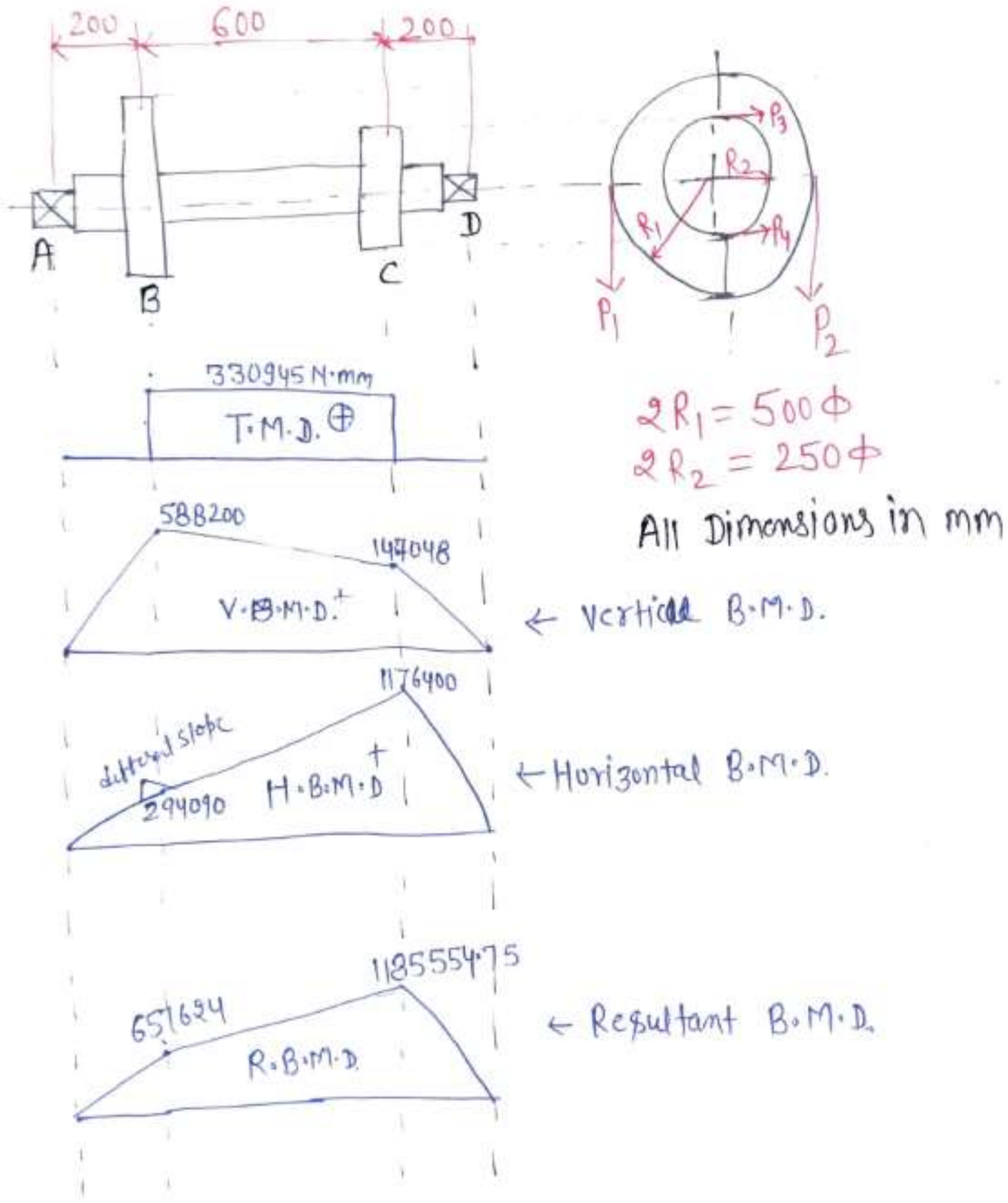
Shaft diameter

Acc. to M.S.S.T.

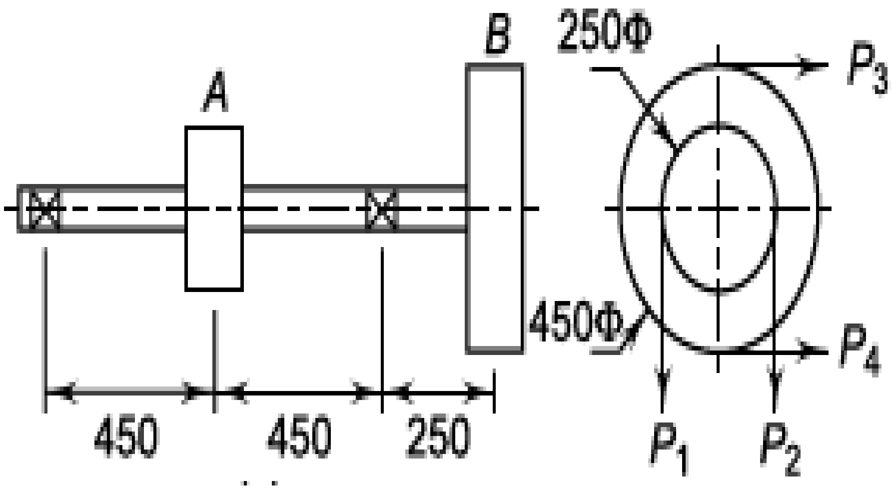
$$\sqrt{M^2 + T^2} \leq \frac{\pi}{16} d^3 \tau_{per}$$

$$\sqrt{(1185554.75)^2 + (330945)^2} \leq \frac{\pi}{16} d^3 \times 66.67$$

$$\boxed{d \geq 45.47 \text{ mm.}}$$


200
600
200
A
B
C
D
P_1
P_2
P_3
P_4
R_1
R_2
330945 N·mm
T.M.D.
$2R_1 = 500\Phi$
$2R_2 = 250\Phi$
All Dimensions in mm
588200
147048
V.B.M.D.
← Vertical B.M.D.
1176400
different slope
294090
H.B.M.D
← Horizontal B.M.D.
1185554.75
657624
R.B.M.D.
← Resultant B.M.D.

Question :-*A line shaft supporting two pulleys A and B is shown in Fig. Power is supplied to the shaft by means of a vertical belt on the pulley A, which is then transmitted to the pulley B carrying a horizontal belt. The ratio of belt tension on tight and loose sides for both the pulleys is 3:1. The limiting value of tension in the belts is 2.7 kN. The shaft is made of plain carbon steel 40C8 (S_{ut} = 650 N/mm^2 and S_{yt} = 380 N/mm^2). The pulleys are keyed to the shaft. Determine the diameter of the shaft according to the ASME code if, k_b= 1.5 and k_t = 1.0.*



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Sol^n

Given $S_{ut} = 650\ N/mm^2$ $\quad S_{yt} = 380\ N/mm^2$

$K_b = 1.5 \quad K_t = 1.0$

for both the belt drive $\frac{P_1}{P_2} = \frac{P_3}{P_4} = 3$

Maximum belt tension $= 2.7\ kN$

Step-I Calculate the permissible shear stress.

$\tau_{max} = .30\ S_{yt}$ or $.18\ S_{ut}$ [whichever is min]

$.30\ S_{yt} = .3 \times 380 = 114\ N/mm^2$
$.18\ S_{ut} = .18 \times 650 = 117\ N/mm^2$

so $114\ N/mm^2$ is selected & and Keyways are there on the shaft

so $\tau_{per} = .75 \times 114$

$$\tau_{per} = 85.5\ N/mm^2$$

Step-II Calculation of torsional moment

at this stage it is not know whether P_1 is max. or P_3 but the torque transmitted by the pulley A is equal to torque received by pully B

Therefore

$$(P_1-P_2)\cdot\left(\frac{250}{2}\right) = (P_3-P_4)\frac{450}{2} = T$$

$$(P_1-P_2) = (P_3-P_4)(1.8) \quad -(i)$$

also $P_2 = P_1/3$ & $P_4 = P_3/3$

put P_2 & P_4 in eqn (i)

$$\frac{2P_1}{3} = 1.8\times\frac{2}{3}P_3 \Rightarrow \boxed{P_1 = 1.8\,P_3}$$

by this $P_1 = 2700$ KN [is max tension in pulley A]

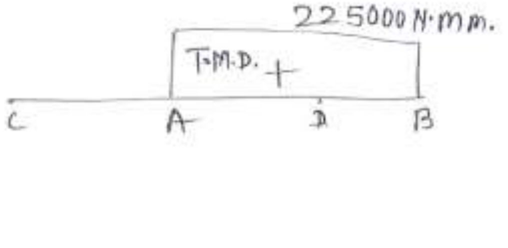
$P_2 = 900$ N , $P_3 = \frac{P_1}{1.8} = \frac{2700}{1.8} = 1500$N

$P_4 = 500$ N.

so, by eqn (i) -

$$T = (1500-500)\times 225 = 225000 \text{ N·mm}$$

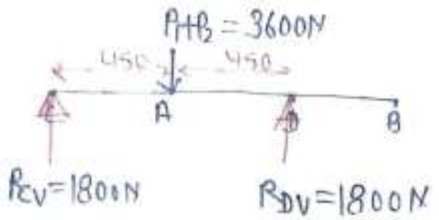
Step-III Calculat

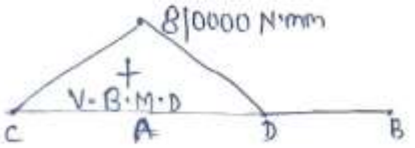


Step-III

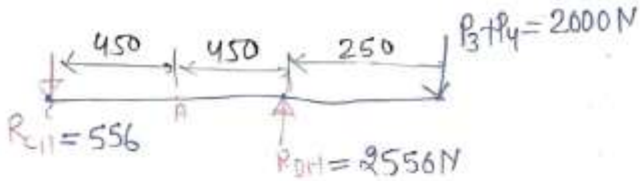
Calculate the Bending moment

(a) due to vertical forces.





(b) due to horizontal forces



~~$R_{CH}+R_{DH} =$~~

$R_{CH} + 2000 = R_{DH}$ ($\Sigma H = 0$)

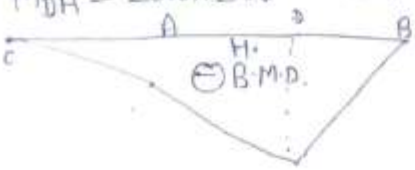
$R_{DH} \times 900 = 2000 \times 1150$ (Moment about pt. C)

$R_{DH} = 2556\ N$

$R_{CH} = 556$

$M_{AH} = 556 \times 450 = 250200\ N\cdot mm$

$M_{DH} = 2000 \times 250 = 50,0000\ N\cdot mm$



Now, Resultant B.M. at H A

$$M_A = \sqrt{(M_{AH})^2 + (M_{AV})^2}$$

$$= \sqrt{(810000)^2 + (250200)^2}$$

$$M_A = 847761.78 \text{ N.mm}$$

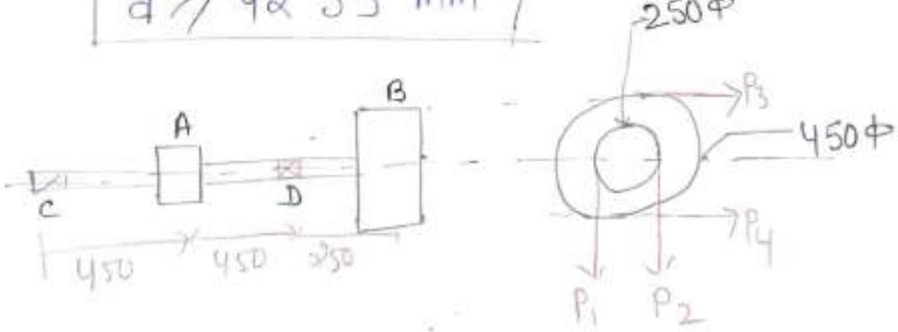
Step - IV diameter of shaft

by M.S.S.T (by ASME code)

$$\sqrt{(K_b M)^2 + (K_t T)^2} \leq \frac{\pi}{16} d^3 \times \tau_{per}$$

$$\sqrt{(1.5 \times 847761.78)^2 + (1 \times 225000)^2} \leq \frac{\pi}{16} d^3 \times 82.5$$

$$\boxed{d \geq 42.53 \text{ mm}}$$



All dimension in mm.

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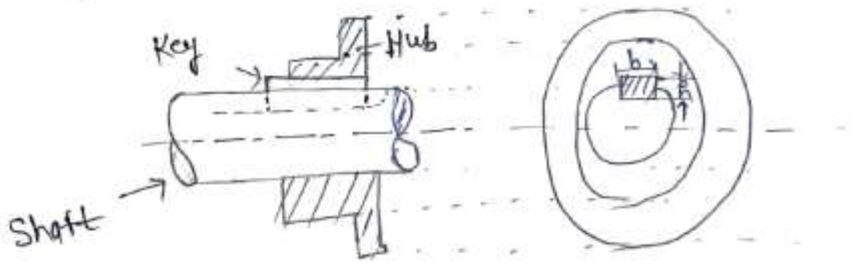
KEY :→

* A Key can be defined as a machine element which is used to connect the transmission shaft to rotating machine element like pulleys, gears sprockets or flywheels.
* A Keyed joint consisting of shaft, hub & key.



Function of Key :-

(i) The primary function of the key is to transmit the torque from the shaft to the hub of the mating element and vice versa.

(ii) The second function of the key is to prevent relative rotational motion between the shaft and the joint machine element like gear or pulley.

it also prevents axial motion betn two elements except in case of feather key or spline connection.

Drawback:-

A recess or slot is machined either on the shaft or in the hub to accommodate the key is called key way. this key way results in stress concentration in the shaft and the part becomes weak.

The key way is usually cut by a vertical or horizontal milling cutter

Material :- plain carbon steel like 45C8 or 50C8 in order to withstand shear and compressive stresses resulting from transmission of torque.

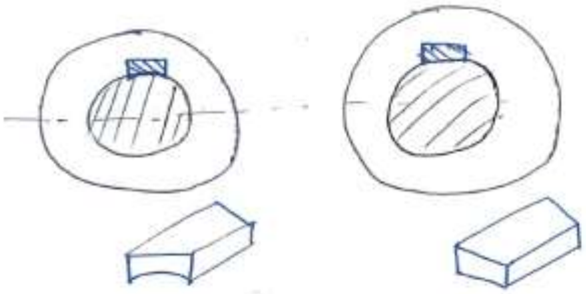
Types of Keys :- The selection of the type of key for a given application depends upon :-

(i) power to be transmitted
(ii) tightness of fit
(iii) stability of connection
(iv) cost.

SADDLE Keys :-

* No key way on the shaft, fits in the key way of the hub only.

Two types of saddle keys
(a) Hollow (b) Flat



concave surface at bottom to match the circular surface of the shaft

flat surface at the bottom & sits on the flat surface machined on the shaft.

→ In both the type of key friction betn the shaft and keys, hub to prevent relative motion betn shaft and the hub.

→ The power is transmitted by means of friction. Therefore, saddle key are suitable for light duty and low power transmission

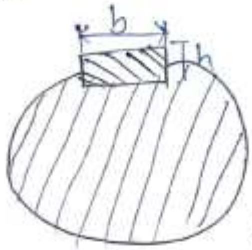
→ cost is low because key way required only in hub.

→ it is liable to slip around the shaft when sub. to heavy torque.

SUNK Keys :→

→ Key way is present in both shaft and hub. i.e. half the thickness of key fits into the key way ~~of~~ on the shaft and the remaining half in the key way on the hub.

→ This is standard form of key and may be either of rectangular or square cross section.



(flat key)
(More stable)

$b = h = d/4$
$l = 1.5d$

used in general industrial machinery

$b > h$,
$b = d/4$, $h = d/6$
$l = 1.5d$

used in machine tool application.

⇓ two groups

(i) parallel → uniform in width as well in height throughout the length of the key.

(ii) taper → uniform in width but taper in height in the top surface (standard taper 1 in 100)

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Advantage of taper:

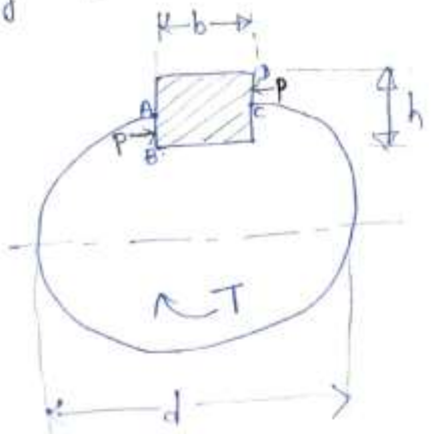
→ tightness increases due to wedge action so that prevents loosening of the parts.

→ easy to remove the key & dismantle the joint.

* Tapered keys are often provided with Gib-head to facilitate removal. the projection of gid-head is hazardous in rotating parts.

Design of SUNK Key (Flat) :-

In Sunk key, power is transmitted by due to shear resistance of the key. the relative motion between the shaft and the hub is also prevented by the shear resistance of the key.

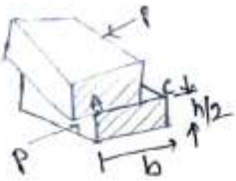


The exact location of force P on the surface AB is unknown. In order to simply the analysis, it is assumed that the force P is tangential to the shaft diameter

Therefore $T = P \times d/2 \Rightarrow P = \frac{2T}{d}$ (a)

The design of Square or flat Key is based on two criteria

(i) failure due to Shear Stress:



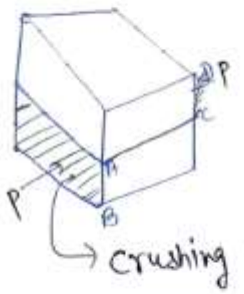
$$\tau = \frac{P}{\text{Shear plane area (AC)}}$$

$$\tau = \frac{P}{b\,l} \quad (b)$$

Combining (a) & (b)

$$\tau = \frac{2T}{d\,b\,l} \quad -(c)$$

(ii) failure due to compressive stress



$$\sigma_c = \frac{P}{\text{Crushing area (AB)}}$$

$$\sigma_c = \frac{P}{\left(\frac{h}{2} \times l\right)}$$

$$\sigma_c = \frac{2P}{h\,l}$$ or Combining

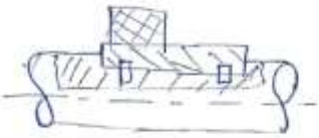
$$\sigma_c = \frac{4T}{d\,h\,l} \quad -(d)$$

eq^n (c) & (d) are for flat keys if we put $h = b$ then obtain eq^n for square keys.

Feather Key :-

→ it is parallel key which is fixed either to the Shaft or to the hub (particular type of Sunk Key)

→ There are no of ways to fix the key to the Shaft or hub.



Fix the key to the shaft by cap screw.

There is a clearance fit between the key and the keyway in the hub. therefore, the hub is free to slide over the key. at the same time, there is no relative rotational motion or movement between the shaft and the hub.

eg:- Clutches or gear Shifting devices.

WOODRUFF Key :- is a SUNK key in the form of an almost semicircular disk of uniform thickness.

Advantage: (i) Can be used on tapered shaft because it can align by slight rotation in the seat

(ii) The extra depth of key in the shaft prevents its tendency to slip over the shaft.

disadvantage:

(i) Stress concentration due to extra depth in the keyway of shaft & reduces its strength

(ii) does not permit axial movement between the shaft and the hub.

Note:- Woodruff keys are used on tapered shafts in machine tools and automobiles.

Question: -The standard cross-section for a flat key, which is fitted on a 50 mm diameter shaft, is 16×10 mm. The key is transmitting 475 N-m torque from the shaft to the hub. The key is made of commercial steel ($S_{yt} = S_{yc} = 230$ N/mm^2).

Determine the length of the key, if the factor of safety is 3

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Sol^n

dia. of Shaft $(d) = 50\,mm$

X-s/c of Key $(b \times h) = 16 \times 10\,mm$

$b = 16\,mm$, $h = 10\,mm$

$T = 475\,N\cdot m$

$S_{yt} = S_{yc} = 230\,N/mm^2$

To determine l, if FOS = 3.

Step-I calculate the permissible stress

$$\sigma_c = \frac{S_{yc}}{FOS} = \frac{230}{3} = 76.67\,N/mm^2$$

$$\tau_{per} = \frac{.5\,S_{yt}}{FOS} = \frac{.5 \times 230}{3} = 38.33\,N/mm^2$$

Step-II Key length

by Shear consideration

$$\tau_{ind} \leq \tau_{per}$$

$$\frac{2T}{dbl} \leq 38.33$$

$$l \geq \frac{2 \times 475 \times 1000}{50 \times 16 \times 38.33}$$

$$\boxed{l \geq 30.98\,mm}$$

by Crushing Consideration :-

$$(\sigma_c)_{ind} \leq (\sigma_c)_{per}$$

$$\frac{4T}{dhl} \leq 76.67$$

$$l \geq \frac{4 \times 475 \times 1000}{50 \times 10 \times 76.67}$$

$$\boxed{l \geq 49.56}$$

by both Consideration length of the key should be choosen 50 mm.

SHAFT COUPLING :-

shafts are usually available up to 7 meters length due to inconvenience in transport. In order to have a greater strength, it becomes necessary to join two or more pieces of the shaft by means of a coupling.

shaft couplings are used in machinery for several purposes. the most common of which are the following :

(1) To provide for the connection of shafts of units that are manufactured separately such as a motor and generator and to provide for disconnection for repairs or alterations.

(2) To provide for misalignment of the shafts or to introduce mechanical flexibility

(3) To reduce the transmission of shock loads from one shaft to another.

(4) To introduce protection against overloads.

(5) To alter the vibration characteristics of rotating units.

Requirements of a Good shaft coupling :-

(1) It should be easy to connect or disconnect
(2) It should transmit the full power from one shaft to the other shaft without losses.
(3) It should hold the shafts in perfect alignment
(4) It should reduce the transmission of shock loads from one shaft to another shaft.
(5) It should have no projecting parts.

* Types of shaft couplings:-

(1) Rigid coupling:- It is used to connect two shafts which are perfectly aligned. Following types of rigid couplings are important
(a) sleeve or muff coupling
(b) clamp or split-muff or compression coupling
(c) flange coupling PNo 209 DHB

(2) Flexible coupling:- It is used to connect two shafts having both lateral and angular misalignmen
(a) Bushed pin type coupling
(b) Universal coupling.
(c) Oldham coupling

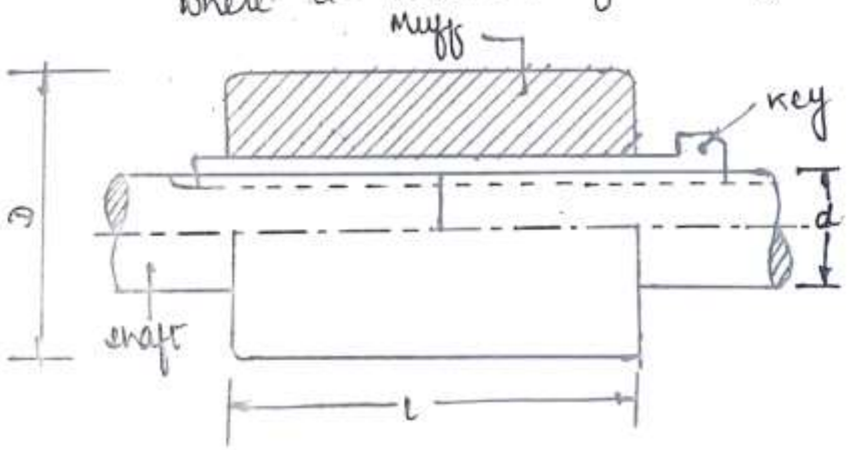
Sleeve or Muff-coupling :- 9 (5)

It is the simplest type of rigid coupling, made of cast iron. It consists of a hollow cylinder whose inner diameter is the same as that of the shaft. It is fitted over the ends of the two shafts by means of a gib head key. The power is transmitted from one shaft to the other shaft by means of a key and a sleeve. It is therefore, necessary that all the elements must be strong enough to transmit the torque. The usual proportions of a cast iron sleeve coupling are as follows:-

Outer diameter of the sleeve ; $D = 2d + 13$ mm

and length of the sleeve : $L = 3.5d$

where d = diameter of the shaft



Design Procedure:-

(1) Design for sleeve:-

The sleeve is designed by considering it as a hollow shaft.

Let T = Torque to be transmitted by the coupling

τ_c = Permissible shear stress for the material of the sleeve which is cast iron. The safe value of shear stress for cast iron may be taken as 14 MPa.

The torque transmitted by a hollow section

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

from this expression, the induced shear stress in the sleeve may be checked.

(2) Design for key:-

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

$$\boxed{l = \frac{L}{2} = \frac{3.5d}{2}}$$

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

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

* Design Procedure for Muff coupling:

(1) calculate the diameter of each shaft by the following equations;

$$M_t = \frac{60 \times 10^6 P(kW)}{2\pi n} \quad \text{and} \quad \tau = \frac{16 M_t}{\pi d^3}$$

(2) calculate the dimensions of the sleeve by the following empirical equations,

$$D = (2d+13)\,mm \quad \text{and} \quad L = 3.5d$$

Also, check the torsional shear stress induced in the sleeve by the following equations:

$$\tau = \frac{M_t r}{J} \quad J = \frac{\pi(D^4 - d^4)}{32} \quad r = \frac{D}{2}$$

(3) Determine the standard cross-section of flat sunk key from data hand Book. The length of the key in each shaft is one half of the length of the sleeve. Therefore,

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

with these dimensions of the key, check the shear and compressive stresses in the key

$$\tau = \frac{2M_t}{dbl} \quad \text{and} \quad \sigma_c = \frac{4M_t}{dhl}$$

The shafts and key are made of plain carbon steel. The sleeve is usually made of grey cast iron of Grade FG 200.

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Clamp or Compression Coupling :-

It is also known as split muff coupling. In this case the muff or sleeve is made into two halves and are bolted together. The halves of the muff are made of cast iron. The shaft ends are made to abutt each other and a single key is fitted directly in the keyways of both the shafts. One half of the muff is fixed from below and the other half is placed from above. Both the halves are held together by means of mild steel studs or bolts and nuts. The number of bolts may be two, four and six. The nuts are recessed into the bodies of the muff casting. This coupling may be used for heavy duty and moderate speeds. The advantage of this coupling is that the position of the shafts need not be changed for assembling or disassembling of the coupling.

The usual proportions are:-

Diameter of the muff or sleeve, $D = 2d + 13$ mm

length of the muff or sleeve, $L = 3.5d$

d = Diameter of shaft.

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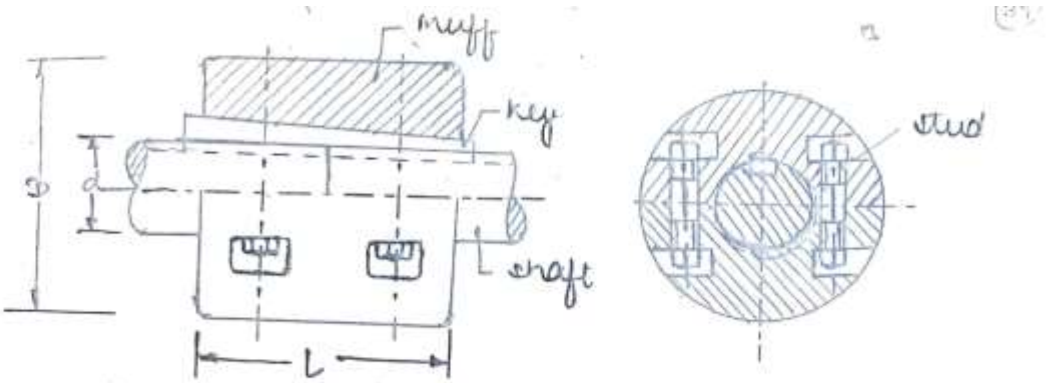

In the clamp or compression coupling the power is transmitted from one shaft to the other by means of key and the friction between the muff and shaft

* Design Procedure:-

(1) Design of muff and key. - same as in muff coupling

(2) Design of clamping bolts :-

Let T = Torque transmitted by the shaft.
d = Diameter of shaft
d_b = Root or effective diameter of bolt
n = Number of bolts
σ_t = Permissible tensile stress for bolt material
μ = coefficient of friction between the muff and shaft.
L = length of muff.

The force exerted by each bolt $= \frac{\pi}{4}(d_b)^2\sigma_t$

∴ Force exerted by the bolts on each side of the shaft
$= \frac{\pi}{4}(d_b)^2\sigma_t \times \frac{n}{2}$

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Let p be the pressure on the shaft and the muff surface due to the force. then for uniform pressure distribution over the surface.

$$p = \frac{\text{Force}}{\text{Projected area}} = \frac{\frac{\pi}{4}(d_b)^2 \times \sigma_t \times \frac{n}{2}}{\frac{1}{2} L \times d}$$

∴ Frictional force between each shaft and muff

$$F = \mu \times \text{pressure} \times \text{area}$$
$$= \mu \times p \times \frac{1}{2} \times \pi d \times L$$
$$= \mu \times \frac{\frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2}}{\frac{1}{2} L \times d} \times \frac{1}{2} \pi d \times L$$
$$= \mu \times \frac{\pi}{4}(d_b)^2 \sigma_t \times \frac{n}{2} \times \pi = \mu \times \frac{\pi^2}{8}(d_b)^2 \sigma_t \times n$$

and the torque that can be transmitted by the coupling

$$T = F \times \frac{d}{2} = \mu \times \frac{\pi^2}{8}(d_b)^2 \sigma_t \times n \times \frac{d}{2}.$$
$$= \frac{\pi^2}{16} \times \mu (d_b)^2 \times \sigma_t \times n \times d$$

from this equation, the root diameter of the bolt (d_b) may be evaluated.

The value of μ may be taken as 0.3

Design of Flange coupling :-

Let d = Diameter of shaft or inner diameter of hub
D_2 = outer diameter of hub
d_1 = Nominal or outside diameter of bolt
D_1 = Diameter of bolt circle
n = Number of bolts
t_f = Thickness of flange
τ_s, τ_b and τ_k = Allowable shear stress for shaft; bolt and key material respectively.
τ_c = Allowable shear stress for the flange material i.e. cast iron
σ_{cb} and σ_{ck} = Allowable crushing stress for the ~~flange~~ bolt material i.e. key material

(i) Design for hub :-

The hub is designed by considering it as a hollow shaft transmitting the same torque (T) as that of a solid shaft.

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

The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore, from the above relation, the induced shearing stress in the hub may be checked.

The length of hub L is taken as $1.5d$.

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Design for key – same.

3) Design for flange:

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the torque transmitted,

$T =$ circumference of hub × Thickness of flange × shear stress of flange × Radius of hub

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

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

as $t_f = \frac{1}{2} d$. i.e. induced shearing stress may be checked.

(4) Design for bolts:-

The bolts are subjected to shear stress due to the torque transmitted. The number of bolts (n) depends upon the diameter of shaft and the pitch circle diameter of bolts (D_1) is taken as 3d. We know that

Load on each bolt $= \frac{\pi}{4}(d_1)^2 \tau_b$

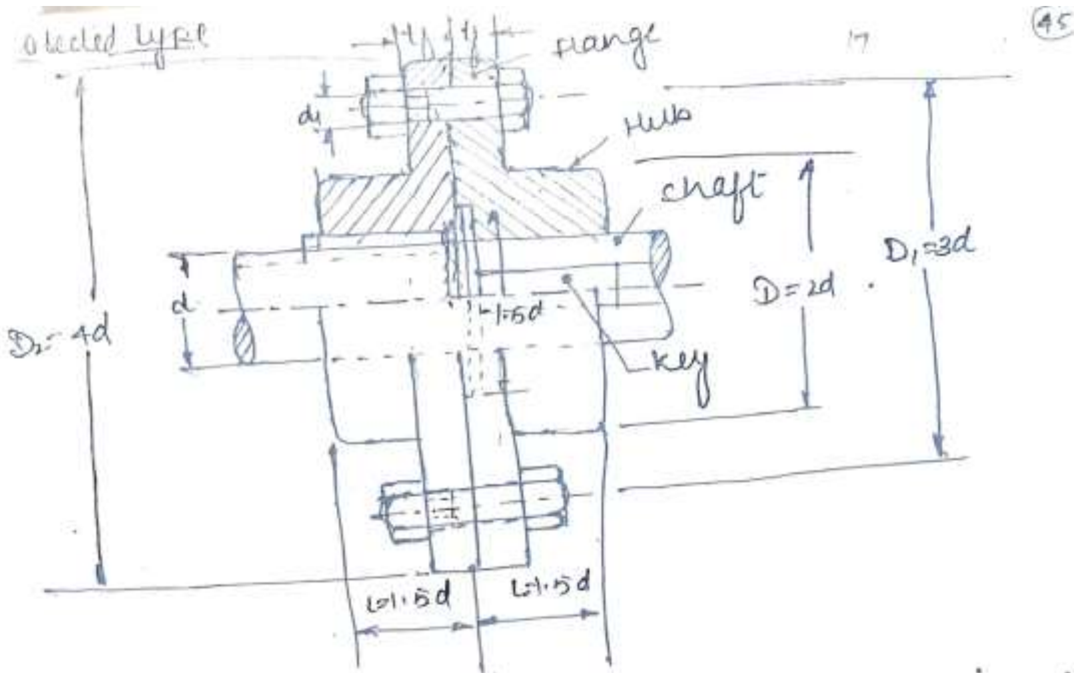
$\therefore$ Total Load on all bolts $= \frac{\pi}{4}(d_1)^2 \times \tau_b \times n$

and torque transmitted $T = \frac{\pi}{4}(d_1)^2 \times \tau_b \times n \times \frac{D_1}{2}$ (check)

Area resisting crushing of all the bolts $= n \times d_1 \times t_f$

$\therefore$ crushing strength $= (n \times d_1 \times t_f)\, \sigma_{cb}$ (check)

Torque $\Rightarrow T = (n \times d_1 \times t_f \times \sigma_{cb})\, D_1/2$



if d is the diameter of the shaft or inner diameter of the hub, then

outside diameter of hub
$D = 2d$.

Length of hub $L = 1.5d$

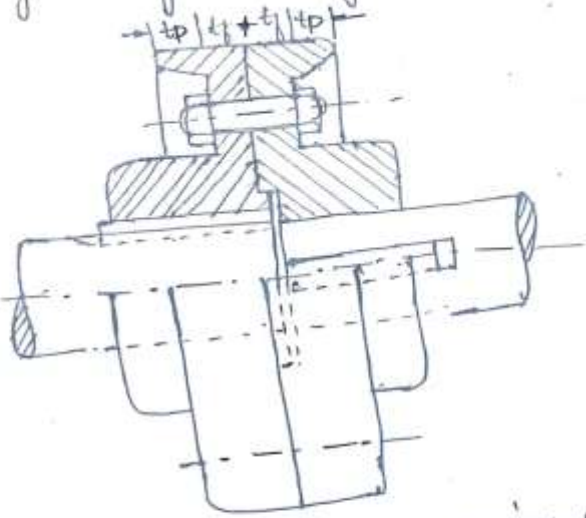
Pitch circle diameter of bolts
$D_1 = 3d$

~~outs~~ outside diameter of flange
$D_2 = D_1 + (D_1 - D) = 2D_1 - D = 4d$

Thickness of flange ~~$D_2 = D_1 + (D_1 - D) = 2D_1 - D =$~~
$t_f = 0.5d$.

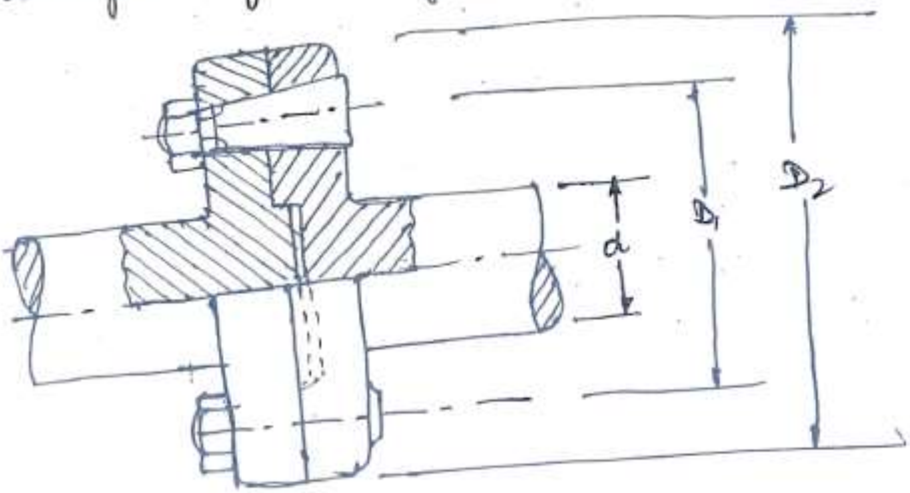
Number of bolts = 3, for d upto 40mm
= 4, for d upto 100mm
= 6, for d upto 180mm

Protected type Flange coupling :-



The Thickness of the protecting circumferential flange (tp) is taken as 0.25d

(3) Marine Type Flange coupling :-



Unit 6: Design of Threaded fasteners, Power screws and Design of curved members

Introduction:-

A screw thread is formed by cutting a continuous helical groove on a cylindrical surface. A screw made by cutting a single helical groove on the cylinder is known as single threaded (or single start) screw and if a second thread is cut in the space between the grooves of the first a double threaded (or double start) screw is formed. The helical grooves may be cut either right hand or left hand.

A screwed joint is mainly composed of two elements i.e. a bolt and nut. The screwed joints are widely used where the machine parts are required to be readily connected or disconnected without damage to the machine or the fastening. This may be for the purpose of holding or adjustment in assembly or service inspection, repair or replacement or it may be for the manufacturing or assembly reasons. The parts may be rigidly connected or provisions may be made for predetermined relative motion.

* Advantages and Disadvantages of screwed joints :- 2

Advantages:-

(1) screwed joints are highly reliable in operation
(2) screwed joints are convenient to assemble and disassemble
(3) A wide range of screwed joints may be adopted to various operating conditions
(4) screws are relatively cheap to produce due to standardisation and highly efficient manufacturing processes.

Disadvantage :-

The main disadvantage of the screwed joints is the stress concentration in the threaded portions which are vulnerable points under variable load conditions

* Important Terms used in screw Threads:-

(1) Major diameter :-
It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter It is also known as outside or nominal diameter.

(2) Minor diameter :- It is the smallest diameter of an external or internal screw thread. It is also known as core or root diameter.

★ stresses in screwed fastening due to static loading:- 3

(1) Internal stresses due to screwing up forces
(2) stresses due to external forces.
(3) stress due to combination of stresses at (1) and (2)

(1) Initial stresses due to screwing up Forces:-
The following stresses are induced in a bolt, screw or stud when it is screwed by tightly

(a) Tensile stress due to streching of bolt:-
Bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the intermined stresses. The initial tension in a bolt, based on experiments, may be found by the relation

$$\boxed{P_i = 2840\,d\ N}$$

where P_i = Initial tension in a bolt
d = Nominal diameter of bolt, in mm

The above relation is used for making a joint fluid tight like steam engine cylinder cover joints etc. When the joint is not required as tight as fluid-tight joint then the initial tension in a bolt may be reduced to half of the above value. In such cases.

$$\boxed{P_i = 1420\,d\ N}$$

The small diameter bolts may fail during tightening therefore bolts of smaller diameter are not permitted in making fluid tight joints.

If the bolt is not initially stressed, then the maximum safe axial load which may be applied to it, is given by 4

P = Permissible stress × Cross-sectional area at bottom of the thread (i.e. stress area)

The stress area may be obtained from Data Hand Book Page No. 5.42 or using the relation

$$\boxed{\text{stress area} = \frac{\pi}{4}\left(\frac{d_2+d_3}{2}\right)^2}$$ Data Hand Book P.No. 5.42

d_p or d_2 = Pitch diameter
d_c or d_3 = core or minor diameter

(b) Torsional shear stress caused by the frictional resistance of the threads during its tightening:-

The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation, we know that

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

$$\Rightarrow \boxed{\tau = \frac{T}{J} r = \frac{T}{\frac{\pi}{32}(d_c \text{ or } d_3)^4} \times \frac{d_c \text{ or } d_3}{2} = \frac{16T}{\pi (d_c \text{ or } d_3)^3}}$$

where τ = Torsional shear stress

T = Torque applied

d_3 or d_c = minor or core diameter of the thread.

(c) Shear stress across the threads -

The average thread shearing stress for the screw (τ_s) is obtained by using the relation:

$$\boxed{\tau_s = \frac{P}{\pi d_c \times b \times n}}$$ 5

where b = width of the thread section at the root

The average thread shearing stress for the nut is

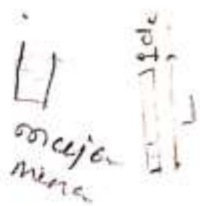
$$\boxed{\tau_n = \frac{P}{\pi d \times b \times n}}$$

where d = major diameter

(d) compression or crushing stress on the threads:-

The compression or crushing stress between the threads (σ_c) may be obtained by using the relation:

$$\boxed{\sigma_c = \frac{P}{\pi\left[d^2 - (d_c)^2\right]n}}$$



where d = Major diameter
d_c = minor diameter
n = Number of threads in engagement

(e) Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis —

when the outside surfaces of the parts to be connected are not parallel to each other, then the bolt will be subjected to bending action. The bending stress (σ_b) induced in the shank of the bolt is given by

$$\boxed{\sigma_b = \frac{x \cdot E}{2L}}$$

where x = Difference in height between the extreme corners of the nut or head.

l = length of the shank of the bolt 6

E = Young's modulus for the material of the bolt

[2] stresses due to External forces:-

(a) Tensile stress - The bolts, studs and screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt

let d_c = Root or core diameter of the thread.

σ_t = Permissible tensile stress for the bolt material

we know that external load applied

$P_i = 4890\,d$

$$P = \frac{\pi}{4}(d_c)^2\sigma_t$$

or $$\boxed{d_c = \sqrt{\frac{4P}{\pi\sigma_t}}}$$

Now from Data Hand Book Page No. 5.42 the value of nominal diameter of bolt corresponding to the value of d_c may be determined or stress area $[\frac{\pi}{4}d_c^2]$ may be fined.

Note:- If the external load is taken up by a number of bolts, then

$$\boxed{P = \frac{\pi}{4}d_c^2\,\sigma_t \times n}$$

(B) shear stress - sometimes the bolts are used to prevent the relative movement of two or more parts as in case of flange coupling, then the shear stress is induced in the bolts. the shear stresses should be

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avoided as far as possible. It should be noted that when the bolts are subjected to direct shearing loads. they should be located in such a way that the shearing load comes upon the body (i.e. shank) of the bolt and not upon the threaded portion.

Let d = major diameter of the bolt
n = Number of bolts.

∴ shearing load carried by the bolts.

$$P_s = \frac{\pi}{4} d^2 \times \tau \times n$$

or $\boxed{d = \sqrt{\frac{4P_s}{\pi \tau n}}}$

$\sigma_t = \frac{32 M_b}{\pi d^3}$

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

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

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

(c) Combined tension and shear stress –

When the bolt is subjected to both tension and shear loads, then the diameter of the shank of the bolt is obtained from the shear load and that of threaded part from the tensile load. A diameter slightly larger than that required for either shear or tension may be assumed and stresses due to combined load should be checked for the following principal stresses

maximum principal shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_t^2 + 4\tau^2}$$

and maximum principle tensile stress

$$\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{\sigma_t^2 + 4\tau^2}$$

These stresses should not exceed the safe permissible values of stresses.

8

(5) Stresses due to combined forces:-

The resultant axial load on a bolt depends upon the following factors:-

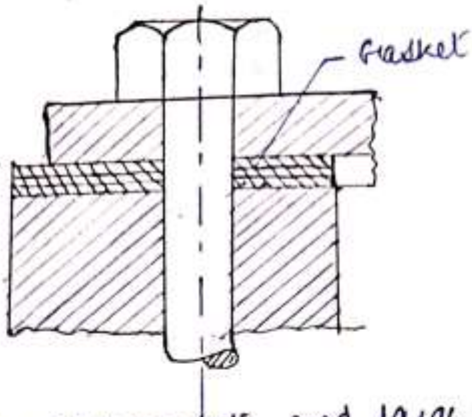
(1) The initial tension due to tightening of the bolt
(2) The external load
(3) The relative elastic yielding of the bolt and the connected members.

When the connected members are very yielding as compared to with the bolt. which is a soft gasket as shown in fig. then the resultant load on the bolt is approximately equal to the sum of the initial and the external load. On the other hand if the bolt is very yielding as compared with the connected members as shown in fig (b). then the resultant load will be either the initial tension or the external load. whichever is greater. The actual conditions usually lie between the two extremes conditions. In order to determine the resultant axial load (P) on the bolt, the following equation may be used.

$$P = P_1 + \frac{a}{1+a} \times P_2 = P_1 + K \cdot P_2$$

where P_1 = Initial tension due to tightening of the bolt
P_2 = External load on the bolt
a = Ratio of elasticity of connected parts to the elasticity of bolt

9



For soft gaskets and large bolts, the value of a is high and the value of $\frac{a}{1+a}$ is approximately equal to unity so that the resultant load is equal to the sum of the initial tension and the external load.

For hard gaskets or metal to metal contact surfaces. and with small bolts, the value of a is small and the resultant load is mainly due to the initial tension

The value of.

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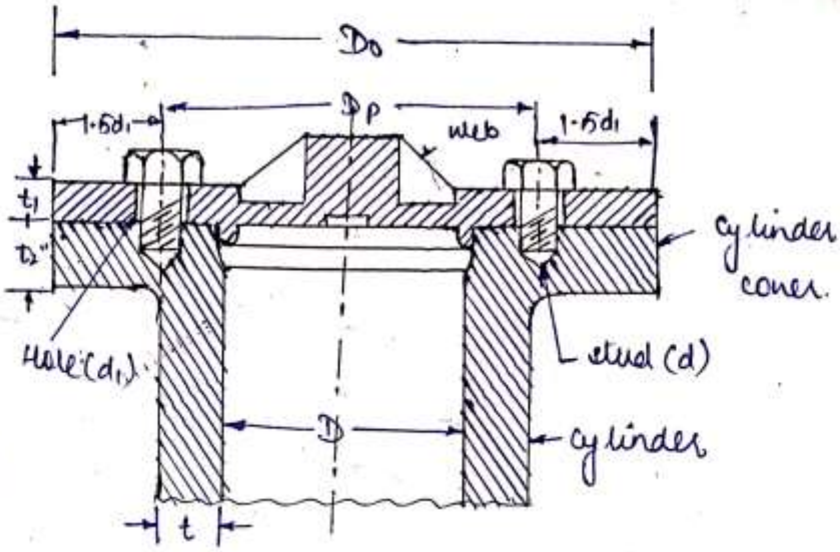

11

★ Design of cylinder covers:-

The cylinder covers may be secured by means of bolts or studs, but studs are Prefered. The possible arrangement of securing the cover with bolts and studs is shown in fig. (a) & (b).

(1) Design of bolts or studs:-

In order to find the size and number of bolts or studs the following procedure may be also adopted

Let D = Diameter of the cylinder
p = Pressure in the cylinder
d_c = core diameter of the bolts or studs
n = Number of bolts or studs, and
σ_{tb} = Permissible tensile stress for the bolt or stud material.

The upward force acting on the cylinder cover

$$\boxed{P = \frac{\pi}{4} D^2 p} \quad ----(1)$$

This force is resisted by n number of bolts or studs provided on the cover

∴ Resisting force offered by n number of bolts or studs

$$\boxed{P = \frac{\pi}{4} d_c^2 \sigma_{tb} \times n} \quad ---(2)$$

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From equations (i) and (ii), we have.

$$\boxed{\frac{\pi}{4} D^2 p = \frac{\pi}{4} d_c^2 \times \sigma_{tb} \times n}$$

If the size of the bolt or stud is known then the number of bolts or studs may be obtained and vice-versa. Usually the size of the bolt is assumed. If the value of n as obtained from the above relation is odd or a fraction, then next higher even number is adopted.

The bolts or studs are screwed up tightly along with metal gasket or asbestos packing, in order to provide a leak proof joint. Due to tightening of bolts, sufficient tensile stress is produced in the bolts or studs. This may break the bolts or studs, even before any load due to internal pressure acts upon them. Therefore a bolt or a stud less than 16 mm diameter should never be used.

The tightness of the joint also depends upon the circumferential pitch of the bolts or studs. The circumferential pitch should be between $20\sqrt{d_1}$ and $30\sqrt{d_1}$, where d_1 is the diameter of the hole in mm for bolt or stud.

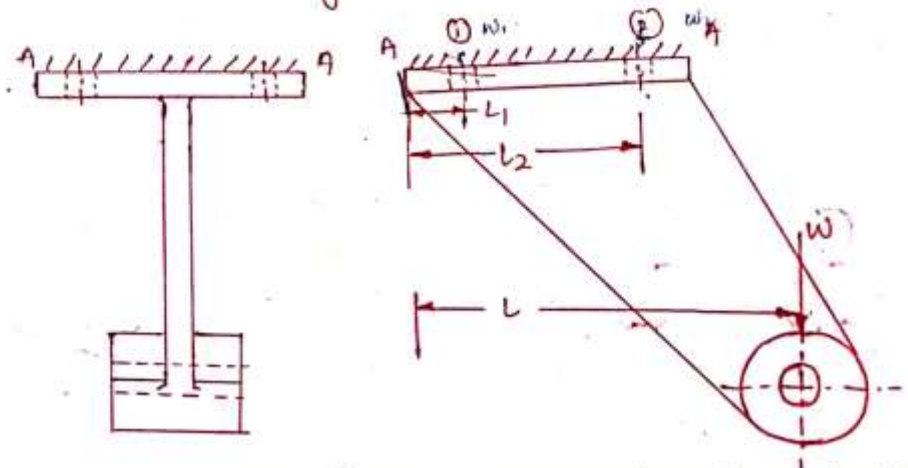
The pitch circle diameter (D_p) is usually taken as $\boxed{D+2t+3d_1}$ and outside diameter of the cover is kept as $\boxed{D_o = D_p + 3d_1 = D+2t+6d_1}$

where t = Thickness of the cylinder wall

Bolted Joints under Eccentric Loading:-

(1) Parallel to the axis of the bolts
(2) Perpendicular to the axis of the bolts
(3) In the plane containing the bolts

(1) Eccentric load acting Parallel to the axis of bolts



consider a bracket having a rectangular base bolted to a wall by means of four bolts. Each bolt is subjected to a direct tensile load of $W_{t1} = \frac{W}{n}$, where n is the number of bolts.

The load W tends to rotate the bracket about the edge A-A. Due to this, each bolt is stretched by an amount that depends upon its distance from the tilting edge. since the stress is a function of elongation, therefore each bolt will experience a different load which also depends upon the distance from the tilting edge. For convenience, all the bolts are made of same size. In case the flange is heavy, it may be considered as a rigid body.

Let w be the load in a bolt per unit distance due to the turning effect of the bracket and let W_1 and W_2 be the loads on each of the bolts at distances L_1 and L_2 from the tilting edge.

$\therefore$ Load on each bolt at distance L_1

$$W_1 = w L_1$$

and moment of this load about the tilting edge

$$= w_1 L_1 \times L_1 = w(L_1)^2$$

Similarly, load on each bolt at distance L_2

$$W_2 = w \cdot L_2$$

and moment of this load about the tilting edge

$$= 2w(L_1)^2 + 2w(L_2)^2 \quad \text{----} (1)$$

(as there are 2 bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge $= W \cdot L$ ---- (2)

From equations (1) and (2), we have

$$W \cdot L = 2w(L_1)^2 + 2w(L_2)^2$$

$$\text{or} \quad w = \frac{WL}{2[(L_1)^2 + (L_2)^2]} \quad \text{---} (3)$$

It may be noted that most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. Therefore the bolts at distance L_2 are heavily loaded.

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∴ Tensile Load on each bolt at distance L_2.

$$W_{t2} = W_2 = w \cdot L_2 = \frac{W \cdot L \cdot L_2}{2[(L_1)^2 + (L_2)^2]}$$

and the total tensile load on the most heavily loaded bolt.

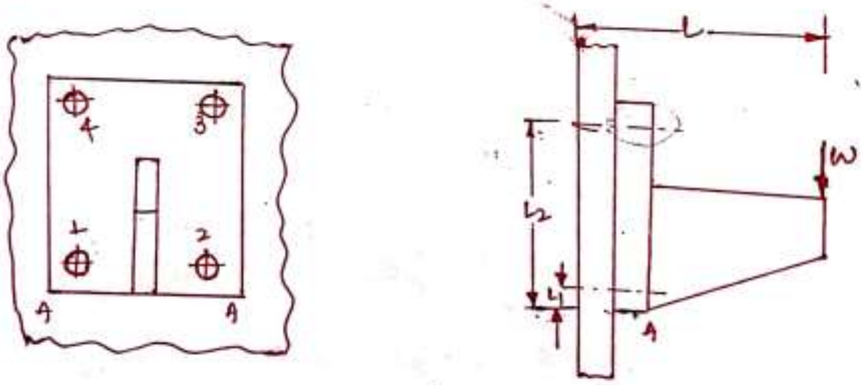
$$W_t = W_{t1} + W_{t2} \quad \text{----} \quad (4)$$

If (d_c) is the core diameter of the bolt and σ_t is the tensile stress for the bolt material, then total tensile load.

$$W_t = \frac{\pi}{4}(d_c)^2 \sigma_t \quad \text{---} \quad (5)$$

from equations (4) and (5), the value of d_c may be obtained.

(2) Eccentric Load Acting Perpendicular to the Axis of Bolts :-



In this case, the bolts are subjected to direct shearing load which is equally shared by all the bolts. Therefore, direct shear load on each bolts.

$$W_s = W/n \text{, where } n \text{ is number of bolts.}$$

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The eccentric load W will try to tilt the bracket in the clockwise direction about the edge A-A. The bolts will be subjected to tensile stress due to the turning moment. The maximum tensile load on a heavily loaded bolt (W_t) may be obtained in the similar manner. In this case, bolts 3 and 4 are heavily loaded

∴ Maximum tensile load on bolt 3 or 4

$$W_{t2} = W_t = \frac{W \cdot L \cdot l_2}{2[(l_1)^2 + (l_2)^2]}$$

When the bolts are subjected to shear as well as tensile loads, then the equivalent loads may be determined by the following relations:

Equivalent tensile load

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

and equivalent shear load

$$W_{se} = \frac{1}{2}\left[\sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

Knowing the value of equivalent loads, the size of the bolt may be determined for the given allowable stresses.

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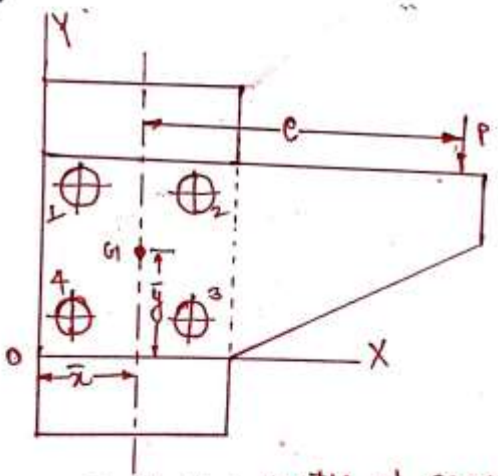
(3) Eccentric load acting in the Plane containing the Bolts :-

When the line of action of the load does not pass through the centroid of the bolt system and thus all bolts are not equally loaded, then the joint is said to be an eccentric loaded bolted joint. The eccentric loading results in secondary shear caused by the tendency of force to twist the joint about the centre of gravity in addition to direct shear or primary shear.



(1) First of all. find the centre of gravity G of the bolt system

Let A = cross-sectional area of each bolt

x_1, x_2, x_3 etc = Distances of bolts from OY and

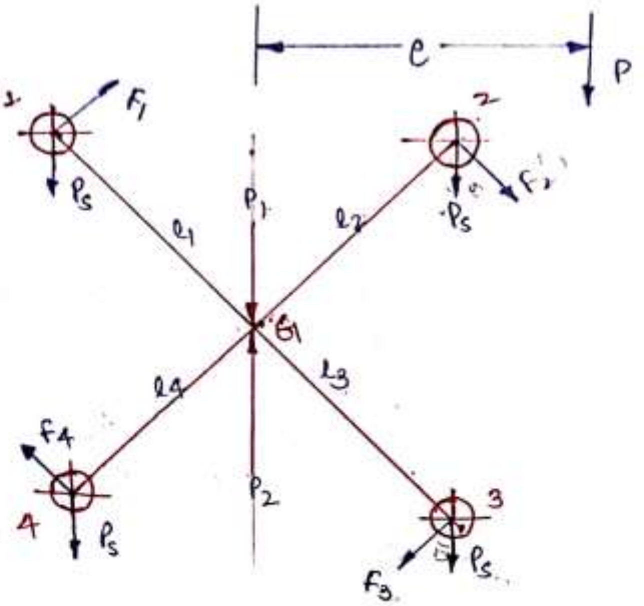
y_1, y_2, y_3 etc = Distances of bolts from OX.

We know that $\bar{x} = \dfrac{A_1x_1 + A_2x_2 + A_3x_3 + \cdots}{A_1 + A_2 + A_3 + \cdots} = \dfrac{Ax_1 + Ax_2 + Ax_3 + \cdots}{n \cdot A}$

$= \dfrac{x_1 + x_2 + x_3 + \cdots}{n}$ n = no. of bolts

Similarly $\bar{y} = \dfrac{y_1 + y_2 + y_3 + \cdots}{n}$

(2) Introduce two forces P_1 and P_2 at the centre of gravity 'G' of the bolt system. These forces are equal and opposite to P.



(3) Assuming that all the bolts are of same size.
The effect of $P_1 = P$ is to produce direct shear load on each ~~shaft~~ bolt of equal magnitude. Therefore. direct shear load on each bolt.

$$P_s = \frac{P}{n}, \text{ acting Parallel to the load } P$$

(4) The effect of $P_2 = P$ is to produce a turning moment of magnitude $P \times e$ which tends to rotate the joint about the centre of gravity 'G' of the bolt system in a clock wise direction.
Due to the turning moment. secondary shear load on each bolt is produced. In order to find the secondary shear load the following two assumptions are made made:-

(a) The secondary shear load is proportional to the radial distance of the ~~rivet~~ bolt under consideration from the centre of gravity of the bolt system.

(b) The direction of secondary shear load is perpendicular to the line joining the centre of the bolt to the C.G. of the bolt system.

Let $F_1, F_2, F_3 \ldots$ = secondary shear loads on the ~~rivets~~ bolts 1, 2, 3 ... etc.

$l_1, l_2, l_3 \ldots$ = Radial distance of the ~~rivets~~ bolts 1, 2, 3 from the centre of gravity 'G' of the bolt system.

$\therefore$ From assumption (a)

$F_1 \propto l_1$; $F_2 \propto l_2$ and so on.

$$\frac{F_1}{l_1} = \frac{F_2}{l_2} = \frac{F_3}{l_3} = \ldots$$

$$\therefore \quad F_2 = F_1 \times \frac{l_2}{l_1} \quad \text{and} \quad F_3 = F_1 \times \frac{l_3}{l_1}$$

The sum of the external turning moment due to the eccentric load and of internal resisting moment of the bolts must be equal to zero.

$$P.e = F_1.l_1 + F_2.l_2 + F_3 l_3 + \ldots$$

$$= F_1 l_1 + F_1 \cdot \frac{l_2}{l_1} \times l_2 + F_1 \times \frac{l_3}{l_1} \times l_3 + \ldots$$

$$= \frac{F_1}{l_1}\left[(l_1)^2 + (l_2)^2 + (l_3)^2 + \ldots\right]$$

The direction of these forces are at right angles to the lines joining the centre of bolt to the centre of gravity of bolt system and should produce the moment in the same direction about the C.G. as the turning moment (P×e)

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(5) The primary and secondary shear load may be added vectorially to determine the resultant shear load (R) on each rivet. It may also be obtained by using

$$R = \sqrt{(P_s)^2 + F^2 + 2P_s \times F \times \cos\theta}$$

where θ = Angle between the primary or direct shear load (P_s) and secondary shear load (F)

When the secondary shear load on each each ~~rivet~~ bolt is equal, then the heavily loaded ~~rivet~~ bolt will be one in which the included angle between the direct shear load and secondary shear load is minimum. The maximum loaded bolt becomes the critical one for determining the strength of the bolted joint knowing the permissible shear stress (τ). the diameter of the bolt hole may be obtained by using the relation.

Maximum resultant shear load $(R) = \frac{\pi}{4} \times d^2 \times \tau$

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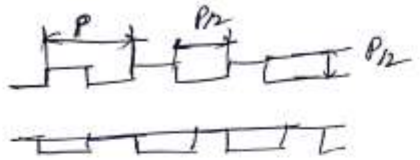
Power screw:

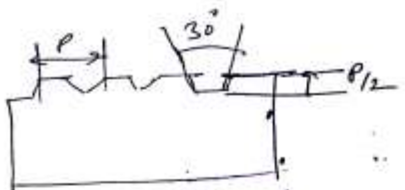
A Power screw is a mechanical device used for converting rotary motion into linear motion and transmitting Power.

(*) The main application of power screws are as follows
(i) to raise the load : Screw Jack
(ii) to obtain accurate motion in machining operations → lead screw.

Form of threads:→

(1) Square Thread





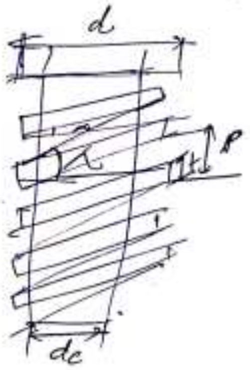
Trapozoidal Thread

Sq 30×6 – Pitch
↑
Nominal diameter

Tr 40×7 → Pitch
Nominal diameter

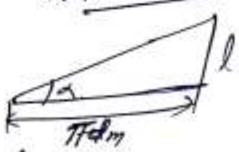
Tr 40×14 (P7)

Lead = np ⇒ n, number of starts



$\boxed{\text{lead} = np}$

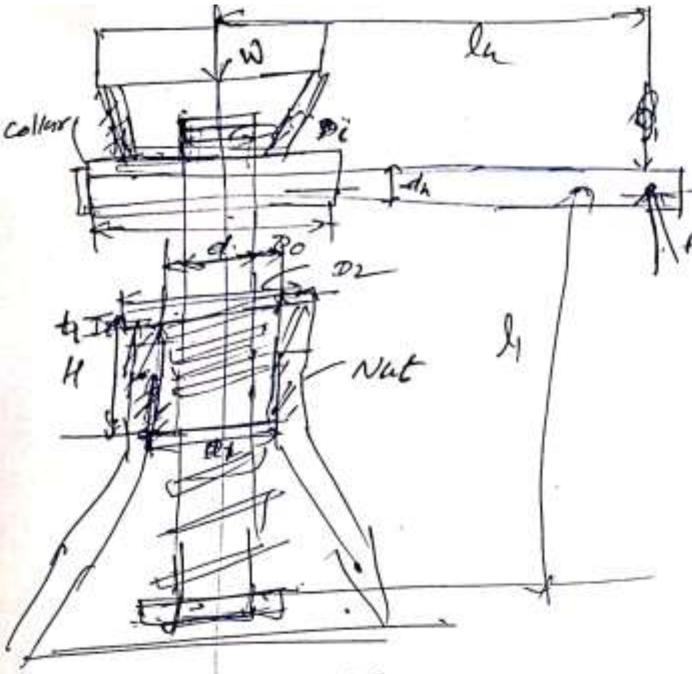
α - helix Angle



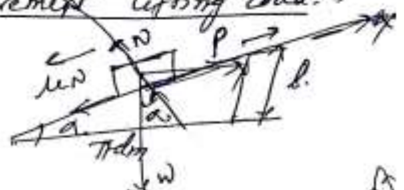
$$\tan\alpha = \frac{l}{\pi d_m}$$

$d_c = d - p$ $\qquad d_m = \frac{d + d_c}{2}$

$d_m = d - \frac{p}{2}$ $\qquad t = \frac{p}{2}$ (thickness or width of thread)



Torque requirement lifting load:



Horizont force:

$\Sigma F_x = 0$; $\Sigma F_y = 0$

$P = \mu N\cos\alpha + N\sin\alpha$; $W = N\cos\alpha - \mu N\sin\alpha$

$P = \frac{W(\mu\cos\alpha + \sin\alpha)}{(\cos\alpha - \mu\sin\alpha)}$

$P = W\left(\frac{\mu + \tan\alpha}{1 - \mu\tan\alpha}\right)$

$\mu = \tan\phi$

$P = \frac{W(\tan\phi + \tan\alpha)}{1 - \tan\phi\tan\alpha}$

$P = W\tan(\phi + \alpha)$

$T = P \times \frac{d_m}{2} = W\tan(\phi+\alpha)\frac{d_m}{2}$

~~Force at bar~~ effort at handle will be

$P \times \frac{d_m}{2} = P_{effort} \times l_h \Rightarrow \boxed{P_{effor} = \frac{P \times d_m}{2 l_h}}$

Collar friction Torque

According to uniform Pressure Theory

$T_1 = \mu W R_{ee}$

$R_{ee} = \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$

According to uniform wear Condition

$T_1 = \mu W \times R_{ee}$

$\boxed{R_{ee} = \frac{R_o + R_i}{2}}$

$T_{total} = T + T_1$

$= W\tan(\phi+\alpha)\frac{d_m}{2} + \mu W R_e$

Torque required to lower the load

$\Sigma F_x = 0$

$P = N\sin\alpha - \mu N\cos\alpha$

$\Sigma F_y = 0$

$W = N\cos\alpha + \mu N\sin\alpha$

$P = \frac{W(\cos\alpha\mu - \sin\alpha)}{(\cos\alpha + \mu\sin\alpha)}$

$\mu = \tan\phi$

$P = W\tan(\phi - \alpha)$

$T = P \times \frac{d_m}{2}$

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Total Torque = $T + \mu W R_{eq}$.

$$\boxed{P_{effort} = \frac{P\, d_m}{2\, l_h}}$$

Efficiency of screw jack

$$\eta = \frac{P_{ideal}}{P_{actual}} = \frac{W \tan\alpha}{W(\tan(\phi+\alpha))} = \frac{\tan\alpha}{\tan(\phi+\alpha)}$$

$$\eta = \frac{W \tan\alpha \frac{d_m}{2}}{\left(W \tan(\alpha+\phi)\frac{d_m}{2} + \mu W R_{eq}\right)}$$

$T = W\frac{d_m}{2}.\tan(\phi-\alpha)$

When $\phi < \alpha$;

T = Negative

no force is required to lower the load. This condition is know as overhauling.

$\phi \geq \alpha$

$T = +ve$

The load will not turn the screw and will not descend itself unless an effort P is applied. This condition is known as Self-locking.

$$\boxed{\eta < \frac{1}{2} \text{ or } 50\%}$$

Maximum efficiency of screw jack

$$\eta = \frac{\tan\alpha}{\tan(\phi+\alpha)}$$

$$\frac{d\eta}{d\alpha} = 0 = \frac{d}{d\alpha}\left\{\frac{\tan\alpha}{\tan(\phi+\alpha)}\right\}$$

$$\alpha = \frac{\pi}{4} - \frac{\phi}{2}$$

$$\eta = \frac{\tan\left(\frac{\pi}{4} - \frac{\phi}{2}\right)}{\tan\left(\frac{\pi}{4} + \frac{\phi}{2}\right)}$$

$$\boxed{\eta_{max} = \frac{1 - \sin\phi}{1 + \sin\phi}}$$

$$\text{Efficiency} = \frac{\text{Work output}}{\text{work input}} = \frac{W l}{2\pi(\text{Torque})}$$

ACME or Trapezoidal Thread

$$\mu = \frac{\mu_a}{\cos\theta}$$

$$\boxed{\mu = \mu_a \sec\theta}$$

$P = W\tan(\alpha + \phi)$ $\mu = \tan\phi$

$\Rightarrow \mu_a \sec\theta = \tan\phi$

Design of Screw Jack

Design of Screw:

(i) compressive stress

$$\sigma_c = \frac{W}{\frac{\pi}{4} d_c^2} \leq \sigma_{cp}$$

(ii) Torsional shear stress

$$T = P\frac{d_m}{2} + \mu W R_{eq}$$

$$\tau = \frac{16T}{\pi (d_c)^3} < \tau_p$$

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Maximum shear stress theory

$$\tau_{max} = \sqrt{\left(\frac{\sigma_c}{2}\right)^2 + \tau^2} < \tau_p$$

Bending stress in screw

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

$$\sigma_n = \sigma_b + \sigma_c$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma_n}{2}\right)^2 + \tau^2} < \tau_{permissible}$$

Design of Nut

Shear stress in ~~Nut~~ ...

$$\tau = \frac{W}{n t \pi d}$$

Crushing stress

$$p_b = \frac{W}{\frac{\pi}{4}(d^2 - d_c^2) \times n} = \frac{4W}{\pi(d^2 - d_c^2) \times n} \le p_b \text{ permissible}$$

$n = ?$ $\boxed{H = np}$ height of Nut

$D = 2d = 2 \times 60 = 120\,mm$

Check for shear stress.

check for

Design of cup

$D_o = 1.6d$

$D_i = 0.8d$

$$T_{total} = W\tan(\phi + \alpha)\frac{d_m}{2} + \mu W R_{eq}$$

$$\frac{T_{total}}{300} = P_{effort} \, l_h$$

$$\Rightarrow \boxed{P_{effort} = \frac{T_{total}}{300 \times l_h}}$$

Design of handle

$$P_e \times \frac{d_m}{2} = P_{effort} \times l_h$$

$$\Rightarrow l_h = \frac{P \frac{d_m}{2}}{P_{effort}}$$

buckling load in screw

L = height of screw $+ \frac{1}{2} \times$ height of Nut

Johnson's Equation:

$$P_{cr} = A\, S_{yt}\left[1 - \frac{S_{yt}}{4 \times C \pi^2 E} \times \left(\frac{L}{k}\right)^2\right]$$

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

$c = 0.25$ end fixity coefficient

$$k = \sqrt{\frac{I}{A}}$$

$I = \frac{\pi}{64} d_c^4$

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

$$\boxed{k = \frac{d_c}{4}}$$

$\frac{l}{k}$ = slenderness Ratio

Design of Collar of Nut

$$\sigma_t = \frac{4W}{\pi((D_1)^2 - d^2)} < \sigma_{t_p}$$

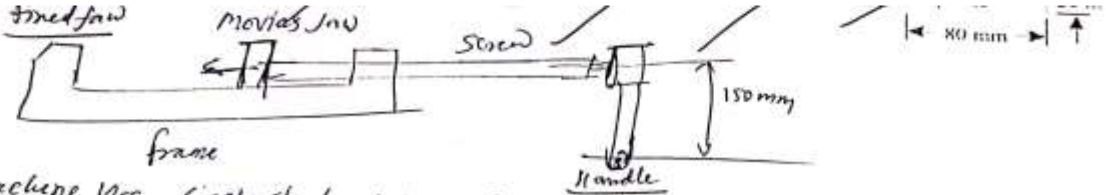
$$\sigma_c = \frac{4W}{\pi(D_2^2 - D_1^2)}$$

$$\tau = \frac{W}{\pi D_1 t_1} < \tau_p$$

$$\Rightarrow \sigma_b = \frac{32M}{\pi (d_h)^3}$$

$M = P_{effort} \times l_h$

$d_h = ?$



A machine vice, single start, square thread with 22 mm nominal diameter and 5 mm pitch. The outer and inner diameters of the friction collar are 55 mm and 45 mm resp. coefficients of friction for thread and collar are 0.15 and 0.17 respectively. The machinist apply a force of 125 N on the handle at a mean radius of 150 mm. Assuming unif wear for the collar calculate:
(i) the clamping force developed between the jaws
(ii) Overall efficiency of clamp.

Soln Given:-

$d = 22\,mm$, $p = 5\,mm$, $n = 1$ start

$D_o = 55\,mm$, $D_i = 45\,mm$, $\mu_t = 0.15$, $\mu_c = 0.17$, $P = 125\,N$, $R_h = 150\,mm$

(1) $d_m = d - \frac{p}{2} = 22 - 5/2 = 19.5\,mm$

$\tan\alpha = \frac{l}{\pi d_m} = \frac{1 \times p}{\pi \times 19.5} = \frac{5}{\pi \times 19.5} \Rightarrow \alpha = 4.67^\circ$

$\tan\phi = \mu_t \Rightarrow \phi = 8.531^\circ$

$T = \frac{W d_m}{2} \tan(\phi + \alpha) = W \times \frac{19.5}{2} \tan(8.531 + 4.67^\circ)$

$T = 2.286W$ N-mm

$T_c = \mu_c W \left(\frac{R_o + R_i}{2}\right) = \frac{\mu_c W}{4}(D_o + D_i)$

$T_c = \frac{0.17 \times W (55 + 45)}{4} \Rightarrow T_c = 4.25W$

Total Torque $\Rightarrow (T)_{total} = T + T_c$

$= 2.286W + 4.25W$

$125 \times 150 = 2.286W + 4.25W$

$\Rightarrow W = 2868.73\,N$

Overall efficiency $= \frac{W \times l}{2\pi T_{total}} = \frac{2868.73 \times 5}{2\pi \times (125 \times 150)}$

$\eta_o = 12.18\%$ Ans

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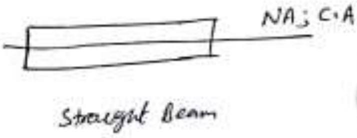
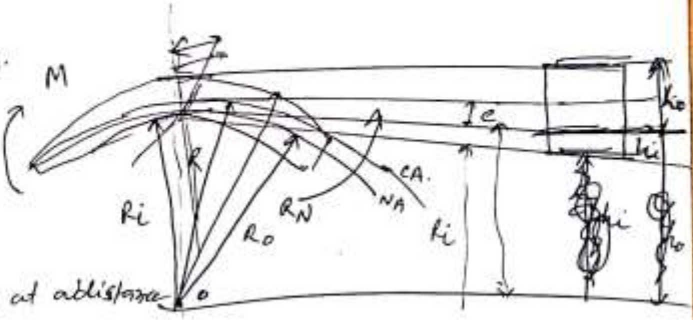
Curved Beam: →





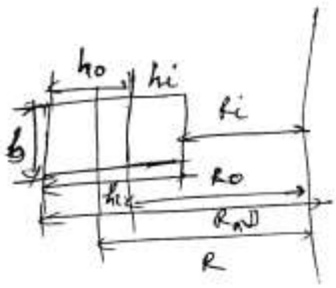
$e = R - R_N$

bending stress at fibre at which is at a distance y from neutral axis

$$\sigma_b = \frac{M_b\, y}{Ae(R_N - y)} \qquad \frac{\sigma_b}{y} = \frac{M}{I}$$

hyperbolic distribution

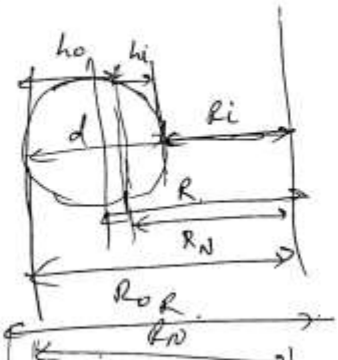
$$\sigma_{bi} = \frac{M h_i}{Ae R_i} \; ; \; \sigma_{bo} = \frac{M h_o}{Ae R_o}$$



Rectangular cross-section

$$R_N = \frac{h}{\log_e\left(\frac{R_o}{R_i}\right)} \qquad A = bh$$

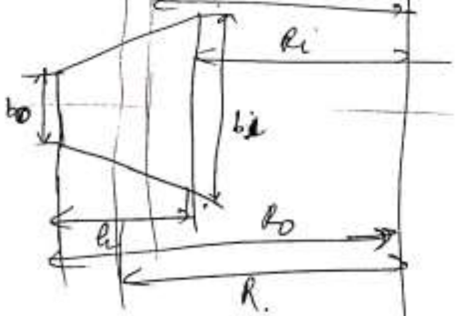
$$R = R_i + \frac{h}{2}$$



Circular cross-section

$$R_N = \frac{\left(\sqrt{R_o} + \sqrt{R_i}\right)^2}{4} \qquad A = \frac{\pi}{4} d^2$$

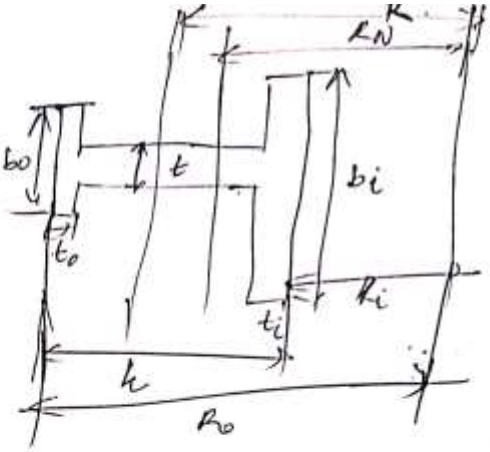
$$R = R_i + \frac{d}{2}$$



Trapezoidal cross-section

$$R_N = \frac{\left(\frac{b_i + b_o}{2}\right) h}{\left(\frac{b_i R_o - b_o R_i}{h}\right) \log_e\left(\frac{R_o}{R_i}\right) - (b_i - b_o)}$$

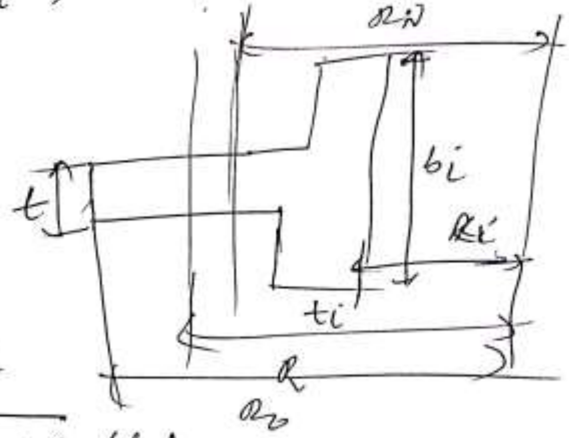
$$R = R_i + \frac{h(b_i + 2b_o)}{3(b_i + b_o)} \qquad A = \frac{1}{2}\left[h(b_i + b_o)\right.$$



$$R_N = \frac{t_i(b_i - t) + t_o(b_o - t) + th}{b_i \log_e\left(\frac{R_i + t_i}{R_i}\right) + t \log_e\left(\frac{R_o - t_o}{R_i + t_i}\right) + b_o \log_e\left(\frac{R_o}{R_o - t_o}\right)}$$

$$R = R_i + \frac{\frac{1}{2} th^2 + \frac{1}{2} t_i^2 (b_i - t) + t_o(b_o - t)\left(h - t_o/2\right)}{t_i(b_i - t) + t_o(b_o - t) + th}$$

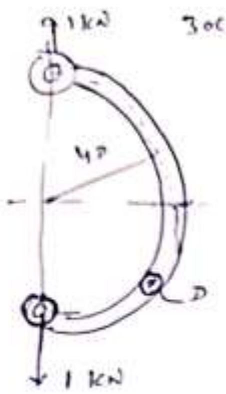
for <u>T Section</u>



$$R_N = \frac{t_i(b_i - t) + th}{(b_i - t) \log_e\left(\frac{R_i + t_i}{R_i}\right) + t \log\left(\frac{R_o}{R_i}\right)}$$

$$R = R_i + \frac{\frac{1}{2} th^2 + \frac{1}{2} t_i^2 (b_i - t)}{th + t_i(b_i - t)}$$

Problem



30C8 $S_{yt} = 400 \frac{N}{mm^2}$

$fs = 3.5$

$M = 1000 \times 40$

$P =$

$R_o = 40$

$R_i = 4D - 0.5D = 3.5D$

$R_o = 4D + 0.5D = 4.5D$

$R_N = \left(\frac{\sqrt{R_o} + \sqrt{R_i}}{4}\right)^2 = 3.984D$

$e = R - R_N =$

$\sigma_{bi} = \frac{44886.51}{D^2} \frac{N}{mm^2}$

$\sigma_t = \frac{P}{\frac{\pi}{4}D^2} = \frac{1000}{0.785D^2} = (1273.24) \frac{N}{mm^2}$

$h_i = R_N - R_i = 0.484D$

$\sigma_{bi} + \sigma_t = 114.29$

$\Rightarrow D = 20.10\ mm$

$\sigma_b = \frac{M h_i}{A e R_i}$

Service factor
Application factor

Starting Torque	1 to 1.2	1.3–1.5	1.6–1.8	2 or 3.0
	Generally Rare	light	moderate	heavy