

COURSE FILE

Academic year : 2022-23

Department : ME

Course Name : B.Tech

Student's Batch : 2020-24

Regulation : R20

Year and Semester : D B.Tech Semester

Name of the Subject : DME-2

Subject Code : Raom = 3201

Faculty In charge : D. Rasharka

Signature of Faculty

Head of the Department



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INSTITUTE VISION AND MISSION



INSTITUTE VISION AND MISSION

VISION:

To emerge as a Centre of excellence in technical education with a blend of effective student centric teaching learning practices as well as research for the transformation of lives and community.

MISSION:

- 1. Provide the best class infrastructure to explore the field of engineering and research.
- 2. Build a passionate and a determined team of faculty with student centric teaching, imbibing experiential and innovative skills.
- 3. Imbibe lifelong learning skills, entrepreneurial skills and ethical values in students for addressing societal problems.

PRINCIPAL



DEPARTMENT VISION AND MISSION



DEPARTMENT VISION AND MISSION

VISION:

To strive for making competent Mechanical Engineering Professionals to cater the real time needs of Industry and Research Organizations of high repute with Entrepreneurial Skills and Ethical Values.

MISSION:

- M1. To train the students with State of Art Infrastructure to make them industry ready professionals and to promote them for higher studies and research.
- M2. To employ committed faculty for developing competent mechanical engineering graduates to deal with complex problems.
- M3. To support the students in developing professionalism and make them socially committed mechanical engineers with morals and ethical values.

HOD-ME



PROGRAM EDUCATIONAL OBJECTIVES (PEOs) AND PROGRAM SPECIFIC OUTCOMES (PSOs)



PROGRAM EDUCATIONAL OBJECTIVES (PEOs)

- PEO 1: Excel in profession with sound knowledge in mathematics and applied sciences
- PEO 2: Demonstrate leadership qualities and team spirit in achieving goals
- PEO 3: Pursue higher studies to ace in research and develop as entrepreneurs.

PROGRAM SPECIFIC OUTCOMES (PSOs)

- **PSO1.** The students will be able to apply knowledge of modern tools in manufacturing enabling to conquer the challenges of Modern Industry.
- **PSO2.** The students will be able to design various thermal engineering systems by applying the principles of thermal sciences.
- **PSO3.** The students will be able to design different mechanisms and machine components of transmission of power and automation in modern industry.

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PROGRAM OUTCOMES (POs)



PROGRAM OUTCOMES (POs):

Engineering Graduates will be able to:

- 1. Engineering knowledge: Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to the solution of complex engineering problems.
- 2. Problem analysis: Identify, formulate, review research literature, and analyse complex engineering problems reaching substantiated conclusions using first principles of mathematics, natural sciences, and engineering sciences.
- 3. 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.
- **4. Conduct investigations of complex problems**: Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and synthesis of the information to provide valid conclusions.
- **5. Modern tool usage:** Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools-including prediction and modelling to complex engineering activities with an understanding of the limitations.
- **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.
- 7. Environment and sustainability: Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the knowledge of, and need for sustainable development.
- **8.** Ethics: Apply ethical principles and commit to professional ethics and responsibilities and norms of the engineering practice.
- **9. Individual and team work:** Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.
- 10. Communication: Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation, make effective presentations, and give and receive clear instructions.
- 11. Project management and finance: Demonstrate knowledge and understanding of the engineering and management principles and apply these to one's own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.
- 12. Life-long learning: Recognize the need for, and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.





BLOOM'S TAXONOMY LEVELS

REVISED Bloom's Taxonomy Action Verbs

		· .	· 5		,	
Definitions	I. Remembering	II. Understanding	III. Applying	'IV: Analyzing	V. Evaluating	VI. Creating
Bloom's Definition	Exhibit memory of previously learned material by recalling facts, terms, basic concepts, and answers.	Demonstrate understanding of facts and ideas by organizing, comparing, translating, interpreting, giving descriptions, and stating main ideas.	Solve problems to new situations by applying acquired knowledge, facts, techniques and rules in a different way.	Examine and break information into parts by identifying motives or causes. Make inferences and find evidence to support generalizations.	Present and defend opinions by making judgments about information, validity of ideas, or quality of work based on a set of criteria.	Compile information together in a different way by combining elements in a new pattern or proposing alternative solutions.
Verbs	 Choose Define Find How Label List Match Name Omit Recall Relate Select Show Spell Tell What When Where Which Who Why 	 Classify Compare Contrast Demonstrate Explain Extend Illustrate Infer Interpret Outline Relate Rephrase Show Summarize Translate 	 Apply Build Choose Construct Develop Experiment with Identify Interview Make use of Model Organize Plan Select Solve Utilize 		 Agree Appraise Assess Award Choose Compare Conclude Criteria Criticize Decide Deduct Defend Determine Disprove Estimate Evaluate Explain Importance Influence Interpret Judge Justify Mark Measure Opinion Perceive Prioritize Prove Rate Recommend Rule on Select Support Value 	 Adapt Build Change Choose Combine Compile Compose Construct Create Delete Design Develop Discuss Elaborate Estimate Formulate Happen Imagine Improve Invent Make up Maximize Minimize Modify Original Originate Plan Predict Propose Solve Suppose Test Theory

derson, L. W., & Krathwohl, D. R. (2001). A taxonomy for learning, teaching, and assessing, Abridged Edition. Boston, MA: Allyn and Bacon.

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COURSE OUTCOMES (COs)

DESIGN OF MACHINE ELEMENTS -II COURSE OUTCOMES

Co	Course Name: DESIGN OF MACHINE ELEMENTS-II Course Code: C321			
.CO _	After successful completion of this course, the students will be able to:			
C321.1	Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings.			
C321.2	Identify the loads and stresses while designing the connecting rod and crank shaft.			
C321.3	Analyze stresses on the piston and cylinder depending upon Design and proportions.			
C321.4	Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design.			
C321.5	Compare capacities of power transmission of Belt, Rope and Chain Drives.			



COURSE INFORMATION SHEET



Narasaraopeta Engineering College (Autonomous)

Yalamanda (Post), Narasaraopet- 522601 Department of Mechanical Engineering

COURSE INFORMATION SHEET

PROGRAMME: B.TECH MECHANICAL ENGINEERING		
COURSE: DESIGN OF MACHINE	Semester: VI CREDITS: 3	
ELEMENTS -2		
(Design Data Hand Book Allowed)		
COURSE CODE:	COURSE TYPE (CORE /ELECTIVE /	
REGULATION: R20	BREADTH/ S&H): CORE	
COURSE AREA/DOMAIN: DESIGN	PERIODS: 6 Per Week.	

COURSE PRE-REQUISITES:

C.CODE	E COURSE DESCRIPTION		SEM
R20ME3103 Design of Machine Elements -1 R20ME2105 Mechanics of Solids		Knowledge of design considerations in machine members and applying engineering principles in designing simple machine members (keys, cotters, couplings, springs)	v
		Knowledge of stress, strain, shear force and bending moment, center of gravity, slope and deflection, torsion.	ш
R20CC1107	Engineering Mechanics	Knowledge of free body diagram, kinematics and dynamics, impulse and momentum.	I
R20CC1102	Linear Algebra & Calculus	An ability to solve ordinary and differential equations	I

COURSE OUTCOMES:

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rank shaft.
nd proportions.
c and dynamic

	stresses to ensure safe design. [K3]
CO5	Compare capacities of power transmission of Belt, Rope and Chain Drives. [K2]

SYLLABUS:

UNIT	DETAILS
I	DESIGN OF BEARINGS: Types of Journal bearings, Lubrication, Bearing
	Modulus, Full and partial bearings, Clearance ratio, Heat dissipation of bearings,
	bearing materials, Journal bearing design, Ball and roller bearings, Static loading of
	ball & roller bearings, Bearing life.
	DESIGN OF I.C ENGINE COMPONENTS: Connecting Rod, Thrust in
77	connecting rod, Stress due to whipping action on connecting rod ends, Cranks and
II	Crank shafts, strength and proportions of over hung and center cranks, Crank pins,
ľ	and Crank shafts.
TTT	Pistons, Forces acting on piston, Construction, Design and proportions of piston,
III	Cylinder. Cylinder liners.
	DESIGN OF CURVED BEAMS: introduction, stresses in curved beams,
IV	Expression for radius of neutral axis for rectangular, circular, trapezoidal and T-
	Section, C – clamps. Design of crane hook.
	POWER TRANSMISSION SYSTEMS: Transmission of power by Belt and Rope
V	drives, Transmission Efficiencies, Belts – Flat and V types – Ropes - pulleys for belt
	and rope drives, Materials, Chain drives.

TEXT BOOKS			
T	BOOK TITLE/AUTHORS/PUBLISHER		
T1	Machine Design, V.B. Bandari, Tata McGraw Hill Publishers.		
T2	Machine Design, Pandya & Shaw, Charotar publishers.		
REFE	REFERENCE BOOKS		
R	BOOK TITLE/AUTHORS/PUBLISHER		
R1	Machine Design, R.N. Norton, Pearson Publishers.		
R2	Design Data hand book. Mahadevan, CBS Publishers.		
R3	Mechanical Engineering Design, JE Shingly, Tata McGraw Hill Publishers.		

TOPICS BEYOND SYLLABUS/ADVANCED TOPICS:

S.NO	DESCRIPTION	Associated PO & PSO
1	Introduction to Design of Machine Tool Elements.	PO1, PO2, PO3, PSO3

WEB SOURCE REFERENCES:

1	Student Resources provided in NEC website student login: http://www.nrtec.ac.in		
2	2 https://lecturenotes.in/subject/426/design-of-machine-members-dmm		
3	https://nptel.ac.in/courses/112106137/		
4	http://www.jntumaterials.co.in/2015/03/jntu-mechanical-engineering-design-of.html		

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DELIVERY/INSTRUCTIONAL METHODOLOGIES:

□1PPT	□1Active Learning
☐ Students Seminars	☐Case Study
□1Quiz	□ Tutorials → -
□1NPTEL/MOOCS	☐ Simulation
☐Industrial Visit	☐ Model Demonstration
□Role Play	□Virtual Labs
	□JQuiz □JNPTEL/MOOCS □Industrial Visit

MAPPING CO'S WITH PO'S:

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	PO	PO	PO	PO	PO	PO	PO	ro	10				100		1 3
CO	1	2	3	4	5	6	7	8	9	10	11	12	1 _	2_	3
		-		-			 	- -	_	-	-	-	-	-	3
CO1	2	3	3					<u> </u>							
CO2	2	3	3	2	-	-	-		-	-		-			3
CO3	2	3	3	2	-	-	-	-	-	-	-	-	-	-	3
COS		3 1	<u> </u>			<u> </u>	 				+	-		-	3
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Average	2	3	3	2		<u> </u>			<u> </u>				<u> </u>		

MAPPING COURSE WITH POs & PSOs:

	Cours	PO 1		PO 3		PO 5	PO 6	PO 7	PO 8	PO 9	PO1 0	PO1 1	PO1 2	PSO 1	PSO 2	PSO 3
				!					├──			-				3
ſ	C311	2	3	3	2	-	-	-	-	-	-	-	-			

Course	Outcome Assessm	Weighta	iges		
Direct	Cumulative	Descriptive Test		1	
Assessment	Internal	Objective Test	30%		Final
	Examinations	Assignment Test	5070	90%	Course
	(CIE)				Outcome
	Semester End Ex	Semester End Examinations (SEE)			(100%)
Indirect	Course End Surv	/ey		10%	
Assessment					

Rubrics for o	overall attainment of o	course outcomes:

If 50% of the students crossed 50% of the marks: Attainment Level 1

If 60% of the students crossed 50% of the marks: Attainment Level 2

If 70% of the students crossed 50% of the marks: Attainment Level 3

Course Instructor

Course Coordinator

Module Coordinator

Head of the Department

ANNEXURE I:

(A) PROGRAM OUTCOMES (POs) Engineering Graduates will be able to:

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

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

mathematics, natural sciences, and engineering sciences.

3. 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.

4. Conduct investigations of complex problems: Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and

synthesis of the information to provide valid conclusions.

5. Modern tool usage: Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modeling to complex engineering activities with an understanding of the limitations.

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.

7. Environment and sustainability: Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the knowledge of, and need for sustainable development.

8. Ethics: Apply ethical principles and commit to professional ethics and responsibilities and

norms of the engineering practice.

9. Individual and team work: Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.

10. Communication: Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation, make effective presentations, and give and receive clear instructions.

11. Project management and finance: Demonstrate knowledge and understanding of the engineering and management principles and apply these to one's own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.

12. Life-long learning: Recognize the need for, and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.

(B) PROGRAM SPECIFIC OUTCOMES (PSOs):

1. The students will be able to understand the modern tools of machining which gives them good expertise on advanced manufacturing methods.

2. The students are able to design different heat transfer devices with emphasis on

combustion and power production.

3. The students are able to design different mechanisms and machine components suitable to automation industry.

Cognitive levels as per Revised Blooms Taxonomy:

Cognitive Domain	Level	Key words
Remembering	K1	Choose, Define, Find, How, Label, List, Match, Name, Omit, Recall, Relate, Select, Show, Spell, Tell, What, When, Where, Which, Who, Why.
Understanding	K2	Classify, Compare, Contrast, Demonstrate, Explain, Extend, Illustrate, Infer, Interpret, Outline, Relate, Rephrase, Show, Summarize, Translate.
Applying	К3	Apply, Build, Choose, Construct, Develop, Experiment With, Identify, Interview, Make Use of, Model, Organize, Plan, Select, Solve, Utilize.
Analyzing	K4	Analyze, Assume, Categorize, Classify, Compare, Conclusion, Contrast, Discover, Dissect, Distinguish, Divide, Examine, Function, Inference, Inspect, List, Motive, Relationships, Simplify, Survey, Take part in, Test for, Theme
Evaluating	K5	Agree, Appraise, Assess, Award, Choose, Compare, Conclude, Criteria, Criticize, Decide, Deduct, Defend, Determine, Disprove, Estimate, Evaluate, Explain, Importance, Influence, Interpret, Judge, Justify, Mark, Measure, Opinion, Perceive, Prioritize, Prove, Rate, Recommend, Rule on, Select, Support, Value
Creating	K6	Adapt, Build, Change, Choose, Combine, Compile, Compose, Construct, Create, Delete, Design, Develop, Discuss, Elaborate, Estimate, Formulate, Happen, Imagine, Improve, Invent, Make up, Maximize, Minimize, Modify, Original, Originate, Plan, Predict, Propose, Solution, Solve, Suppose, Test, Theory

Unit wise Sample assessment questions

COURSE OUTCOMES: Students are able to

CO1: Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings. [K3]

CO2: Identify the loads and stresses while designing the connecting rod and crank shaft. [K3]

CO3: Analyze stresses on the piston and cylinder depending upon Design and proportions. [K4]

CO4: Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design. [K3]

CO5: Compare capacities of power transmission of Belt, Rope and Chain Drives. [K2]

S.NO	QUESTION	KNOWLEDGE LEVEL	СО
	UNIT 1		
1	a) What are rolling contact bearings? Discuss their advantages over sliding contact bearings. b) A ball bearing subjected to a radial load of 5KN is expected to have a life of 8000 hours at 1450 rpm with a reliability of 99 per cent. Calculate dynamic load capacity of the bearing so that it can be selected from manufactures catalogue based on a reliability of 90 per cent.	K1 & K3	CO1
2	 a) List the important physical characteristics of a good bearing material. b) The main bearing of a steam engine is 100 mm and 175 mm long. The bearing supports a load of 28 KN at 250 rpm. If the ratio of diametral clearance to the diameter is 0.001 and absolute viscosity of the lubricating oil is 0.015 kg/m-s. Find: i) the coefficient of friction and ii) heat generated at the bearing due to friction. 	K1 & K3	CO1
3	a) A 100 mm long and 60 mm diameter journal bearing supports a load of 2500 N at 600 rpm. If the room temperature is 20°C, what should be the viscosity of oil to limit the bearing surface temperature to 60°C? Diametral clearance = 0.06 mm. Energy dissipation coefficient based on the projected area of the bearing = 210 W/m ² °C b) A 250×250 mm bearing carries a load of 108 KN. The bearing rotates at 1500 rpm. The clearance ratio is 670. For full journal bearing the power lost in friction is 14.36 KW. Find the viscosity of the oil.	К3	CO1
4	What are the various terms used the journal bearings analysis and design. Give their definitions in brief.	K1	CO1
	UNIT 2		
1	The following data is given for the cap and bolts of the big end of the connecting rod. Engine speed=1500rpm length of the connecting rod =320mm length of stroke=140mm mass of reciprocating parts=1.75kg length of crank pin=54mm diameter of the crank pin=38mm permissible tensile stress for the bolts=120MPa permissible bending stress for cap=120MPa Calculate the nominal diameter of the bolts and thickness of the cap for the big end.	К3	CO2
2	 a) Explain the design of centre crankshaft when crank is at dead center. b) Explain the design of centre crankshaft when crank is at an angle with the line of dead centre positions and subjected to maximum torsional moment. 	К2	CO2

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3	A connecting rod is to be designed for a high speed, four stroke IC engine. The following data are available. Piston diameter=88mm; Mass of reciprocating parts=1.6kg; Length of the connecting rod (centre to centre)=300mm; Stroke=125mm; RPM=2200 (when developing 50kw); Possible over speed=3000rpm; compression ration = 6.8:1; Probable explosion pressure=3.5MPa. Design and draw fully dimensioned drawing of the connecting rod.	К3	CO2
	UNIT 3	<u>-</u>	
1	The following data is given for the piston of a four stroke diesel engine. Cylinder bore=100 mm Material of piston rings=grey cast iron Allowable tensile stress=90 MPa Allowable radial pressure on the cylinder wall=0.035MPa Thickness of the piston head=16 mm Number of piston rings=4 Calculate (i) Radial width of piston rings (ii) Axial thickness of the piston rings (iii) Gap between the free ends of the piston ring after assembly (iv) Gap between the free ends of the piston ring after assembly (v) Width of the top land (vi) Width of ring grooves (vii) Thickness of piston barrel and (viii) Thickness of barrel at top end	К3	CO3
2	Design a cast iron piston for a four stroke IC engine for the following specifications Cylinder bore=120 mm Stroke length=150 mm Maximum gas pressure=5 MPa Brake mean effective pressure=0.7 MPa Fuel consumption=0.25 Kg/KW/hr Speed=2400 rpm Assume any other data necessary for the design	К3	CO3
3	a) Explain the various types of cylinder liners.b) Discuss the design of piston for an internal combustion engine.	K2 & K6	CO3
4	The following data is given for the piston of a four stroke diesel engine. Cylinder bore = 120 mm Maximum gas pressure = 6 MPa Allowable bearing pressure for skirt = 0.45 MPa Ratio of side thrust on liner to maximum gas load on piston=0.1 Width of top land =20 mm Width of ring grooves = 3 mm Total number of piston rings = 4 Axial thickness of piston rings = 3.5 mm	К3	CO3

	Calculate (i)length of skirt (ii) length of piston		
	UNIT 4		<u> </u>
1	A punch press, used for stamping sheet metal, has a punching capacity of 50 kN. The section of the frame is as shown in Fig. Find the resultant stress at the inner and outer fibre of the section.(All dimensions are in mm) 800. 800. Section at X-X	К3	CO4
2	Determine (i) location of neutral axis, (ii) maximum and minimum stresses when a curved beam of circular section of diameter 60 mm is subjected to pure bending moment of + 1350 Nm. The bottom width is towards the centre of curvature. The radius of curvature is 75 mm and beam is curved in a plane parallel to depth. Also plot the variation of stresses across the section.	К3	CO4
3	A central horizontal section of a hook is symmetrical trapezium 90 mm deep. The inner width being 90 mm and outer being 45 mm. The hook carries a load of 67.5 KN, the load line passes at a distance of 40 mm from the inside edge of the section. The centre of curvature is in the load line. Calculate the extreme intensities of stress. Also plot the stress distribution across the section.	К3	CO4
4	a) Write and derive the expression for Winkler-Bach formula.b) What are the assumptions made in the derivation of stresses in a curved bar.	K1	CO4
	UNIT 5		
1	A V belt drive is required for a 15 KW, 1440 rpm electric motor, which drives a centrifugal pump running at 360 rpm for a service of 24 hours per day. From space considerations, the center distance should be approximately 1m. Determine (i) Belt specifications (ii) Number of belts (iii) Correct center distance and (iv) Pulley diameters	К3	COS
2	It is required to select a v-belt drive to connect a 20 KW, 1440 rpm motor to a compressor running at 480 rpm for 15 hours per day. Space is available for a center distance	К3	COS

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l	approximately 1.2m. Determine		T
	i) The specifications of the belt		
	ii) Diameter of the motor and compressor pulleys		
	iii) The correct center distance and the number of belts		
3	A100 mm wide and 10 mm thick belt transmits 5 KW between two parallel shafts. The distance between the shaft centers is 1.5 m and the diameter of the smaller pulley is 440 mm. The driving and driven shafts rotate at 60 rpm and 150 rpm respectively. Find the stress in the belt, if the two pulleys are connected by i) open belt ii) a cross belt. The coefficient of friction is 0.22	К3	CO5



(AUTONOMOUS)

III B.Tech II Semester Regular Examinations, Month/Year: Sub Code: R20ME3201

SUBJECT NAME: DESIGN OF MACHINE ELEMENTS-II

(ME)

MODEL PAPER-I

Time: 3 hours Max. Marks: 70

Note: Answer All FIVE Questions. All Questions Carry Equal Marks (5 X 14 = 70M)

Q. NO		QUESTION	KL	co	MARKS		
110		Unit - I					
1	а	Solve the following problem. Design a journal bearing for a centrifugal pump running at 1440 r.p.m. The diameter of the journal is 100 mm and load on each bearing is 20 kN. The factor <i>ZN/p</i> may be taken as 28 for centrifugal pump bearings. The bearing is running at 75°C temperature and the atmosphere temperature is 30°C. The energy dissipation coefficient is 875 W/m²/°C. Take diametral clearance as 0.1 mm.	К3	CO1	[14M]		
		OR					
	b	Solve the following problem. Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 r.p.m. for an average life of 5 years at 10 hours per day. Assume uniform and steady load.	К3	CO1	[14M]		
		Unit - II		1	<u></u>		
2	а	Solve the following problem. Design a suitable connecting rod for a car with the following data: Piston diameter = 68mm. Stroke length = 80 mm. Length of connecting rod = 160mm. Maximum explosion pressure = 3.5 Mpa. Weight of the reciprocating parts = 2.5 kg Speed =4000 rpm. Compression ratio 8:1	К3	CO2	[14M]		
	_	Compression ratio 6.1		1	·		
	b	Explain the design of Center Crankshaft when the crank shaft is at dead center.	K2	CO2	[14M]		
3		Unit - III					

	\neg	Calve the fallering 11									
	а	Brake power = 5 kW., Speed = 1200 r.p.m Imep = 0.35 N/mm ² , Mechanical Efficiency = 80 %.	K3	CO3	[14M]						
		Calculate the dimensions of the cylinder and the cylinder head.									
			1								
}	-	Solve the following problem.	T								
	b	Design a plain carbon steel center crank shaft when the crank is at dead center for a single acting 4 stroke single cylinder engine for the following data:	K3	CO3	[14M]						
	1	Unit - IV	<u> </u>								
4	a	Solve the following problem. A curved beam of rectangular section of 30 mm width and 40 mm depth and mean radius of curvature of 60 mm is initially unstressed. If a bending moment of 400 N-m is applied to the beam which tends to straighten it, determine the stresses at the inner and outer surface and sketch a diagram to show the variation of stresses across the section. Also find the position of Neutral axis.	K3	CO4	[14M]						
	<u> </u>	OR									
	b	What are the assumptions made in Winkler Bach theory? Derive an expression for radius of neutral axis for T section.	K2	CO4	[14M]						
	<u> </u>	Unit - V									
	a	Derive the expression for length of on Open belt drive & Cross belt drive.	K5	CO5	[14M]						
		OR									
5	ъ	Solve the following problem. In a cross belt drive the diameter of the driver and follower pulleys are 200mm and 400mm respectively. The centre distance of the driver is 2m. The driver pulley rotates at 400rpm. Find the angle of contact between belt and both the pulleys and the length of the belt required .What is he power capacity of the driver, if the permissible tension in the belt is 1.2kn and coefficient of friction between belt and pulleys is 0.252.	К3	CO5	[14M]						



(AUTONOMOUS)

III B.Tech II Semester Regular Examinations, Month/Year: Sub Code: R20ME3201

SUBJECT NAME: DESIGN OF MACHINE ELEMENTS-II

(ME) MODEL PAPER-II

Time: 3 hours Max. Marks: 70

Note: Answer All **FIVE** Questions. All Questions Carry Equal Marks (5 X 14 = 70M)

Q. NO			QUESTION	KL	CO	MARKS
			Unit - I			
1	a	A journal has a bea journal is the diam lubricated operating kg/m-s. T the amour (ii) the difference the oil is	bearing of 50mm diameter and 80mm long, ring pressure of 6 N/mm ² , the speed of the 1000 rpm. The ratio of journal diameter to etral clearance is 1000. The bearing is with oil, whose absolute viscosity at the temperature of 75°C may be taken as 0.015 he room temperature is 25°C, Determine (i) at of artificial cooling required and mass of the coolant oil required, if the between the outlet and inlet temperature of 10°C, the specific heat of the oil is 1900 and heat dissipation coefficient is 500	K3	CO1	[14M]
		,,,,,,,	OR			<u> </u>
	b	A shaft ro variable lo subjected for 10 per kN for 30 time of cy is 20 × 10	following problem. tating at constant speed is subjected to bad. The bearings supporting the shaft are to stationary equivalent radial load of 3 kN cent of time, 2 kN for 20 per cent of time, 1 per cent of time and no load for remaining cle. If the total life expected for the bearing revolutions at 95 per cent reliability, dynamic load rating of the ball bearing.	К3	cO1	[14M]
		٠.	Unit - II			
2	a	A horizo	following problem. ontal four stroke diesel engine has the specification.	K3	CO2	[14M]

	1	Declaration of the control of the co					
		Brake power = 6 kW., Speed = 1800 r.p.m					
		Imep = 0.45 N/mm ² , Mechanical Efficiency = 75 %.					
	1	Calculate the dimensions of the cylinder and the	1	1			
		cylinder head.					
	\vdash	OR					
		Solve the following problem.					
		Design a plain carbon steel center crank shaft when					
		the crank is at dead center for a single acting 4 stroke single cylinder engine for the following data:					
ļ	b	Piston diameter =275 mm.	170				
l.	0	Stroke = 420 mm	K3	CO2	[14M]		
	ŀ	Maximum combustion pressure =2.5 N/mm ²					
		Weight of the fly wheel =15 kN.					
		Total belt pull =2.5 N					
	+-	Length of the connecting rod = 850 mm. Unit - III	<u> </u>	_l			
	a	Explain in detail the design procedure of Cylinder.	T 770	T ===			
			K2	CO3	[14M]		
	<u> </u>	OR					
		Solve the following problem.					
		Design a cast iron piston for a single acting four stroke engine for the following data:					
3		Cylinder bore = 100 mm; Stroke = 125 mm;					
	Ь	Maximum gas pressure = 5 N/mm2; Indicated mean	K3	000	F1 43 47		
		effective pressure = 0.75 N/mm ² ; Mechanical	K	CO3	[14M]		
		efficiency = 80%; Fuel consumption = 0.15 kg per					
		brake power per hour; Higher calorific value of fuel =					
		42×10^3 kJ/kg; Speed = 2000 r.p.m. Any other data required for the design may be assumed.					
		Unit - IV	<u> </u>				
		Solve the following problem.					
		A curved bar of rectangular section, initially					
		unstressed, is subjected to bending moment of 1400 N-					
	9	m which tends to straighten the bar. The section is 4	77.0	004	F4 43 53		
4	a	cm wide by 5 cm deep in the plane of bending, and the mean radius of curvature is 10 cm. Find the position of	K3	CO4	[14M]		
•		the neutral axis and magnitudes of the greatest bending					
		stress and draw a diagram to show approximately how	i				
		the stress varies across the section.					
		OR					
	ь	Derive an expression for radius of neutral axis for circular section.	K5	CO4	[14M]		
		Unit - V		L	[]		
E	a	What are different types of Flat Belt Drives? Explain	 -				
		any four with neat sketch.	K2	CO5	[14M]		
5	OR						
	ь	Solve the following problem.	K3	CO5	[14M]		
		Two pulleys, one 450 mm diameter and the other 200			[T.4TAT]		

	mm diameter, on parallel shafts 1.95 m apart are	
	connected by a crossed belt. Find the length of the belt	
	required and the angle	
i	of contact between the belt and each pulley. What	
	power can be transmitted by the belt when the larger	
	pulley rotates at 200 rev/min, if the maximum permissible tension in the belt is 1 kN, and the	
	coefficient of friction between the belt and pulley is	
	0.25?	

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Narasaraopeta Engineering College (Autonomous) Yallmanda (Post), Narasaraopet- 522601 Department of Mechanical Engineering

COURSE INFORMATION SHEET

PROGRAMME: B.TECH MECHANICAL ENGINEERING		
COURSE: DESIGN OF MACHINE	Semester: VI CREDITS: 3	
ELEMENTS -2		
(Design Data Hand Book Allowed)		
COURSE CODE:	COURSE TYPE (CORE /ELECTIVE /	
REGULATION: R19	BREADTH/ S&H): CORE	
COURSE AREA/DOMAIN: DESIGN	PERIODS: 5 Per Week.	

COURSE PRE-REQUISITES:

C.CODE	COURSE NAME	DESCRIPTION	SEM
19BME5TH04	Design of Machine Elements -1	Knowledge of design considerations in machine members and applying engineering principles in designing simple machine members (keys, cotters, couplings, springs)	V
19BME3TH06	Mechanics of Solids	Knowledge of stress, strain, shear force and bending moment, center of gravity, slope and deflection, torsion.	m
19BCC2TH11 Engineering Mechanics		Knowledge of free body diagram, kinematics and dynamics, impulse and momentum.	II
19RCC1TH03 Algebra &		An ability to solve ordinary and differential equations	I

COURSE OUTCOMES:

S.NO	COURSE OUTCOME STATEMENT
CO1	Apply the design procedure to engineering problems, including the consideration of
COI	technical and manufacturing constrains of bearings. [K3]
CO2	Identify the loads and stresses while designing the connecting rod and crank shaft.
CO2	[K3] '
CO3	Analyze stresses on the piston and cylinder depending upon Design and proportions.
COS	[K4]
CO4	Identify the loads and machine members subjected and calculate static and dynamic

	stresses to ensure safe design. [K3]
CO5	Compare capacities of power transmission of Belt, Rope and Chain Drives. [K2]

SYLLABUS:

UNIT	DETAILS
	DESIGN OF BEARINGS: Types of Journal bearings, Lubrication, Bearing Modulus, Full and partial bearings, Clearance ratio, Heat dissipation of bearings,
I "	bearing materials, Journal bearing design, Ball and roller bearings, Static loading of ball & roller bearings, Bearing life.
п	DESIGN OF I.C ENGINE COMPONENTS: Connecting Rod, Thrust in connecting rod, Stress due to whipping action on connecting rod ends, Cranks and Crank shafts, strength and proportions of over hung and centre cranks, Crank pins, Crank shafts.
ш	Pistons, Forces acting on piston, Construction, Design and proportions of piston, Cylinder. Cylinder liners.
IV	DESIGN OF CURVED BEAMS: introduction, stresses in curved beams, Expression for radius of neutral axis for rectangular, circular, trapezoidal and T-Section, C – clamps. Design of crane hook.
V	POWER TRANSMISSION SYSTEMS: Transmission of power by Belt and Rope drives, Transmission Efficiencies, Belts – Flat and V types – Ropes - pulleys for belt and rope drives, Materials, Chain drives.

TEXT BOOKS			
T	BOOK TITLE/AUTHORS/PUBLISHER		
T1	Machine Design, V.B. Bandari, Tata McGraw Hill Publishers .		
T2	Machine Design, Pandya & Shaw, Charotar publishers.		
REFI	REFERENCE BOOKS		
R	BOOK TITLE/AUTHORS/PUBLISHER		
R1	Machine Design, R.N. Norton, Pearson Publishers.		
R2	Design Data hand book. Mahadevan, CBS Publishers.		
R3	Mechanical Engineering Design, JE Shingly, Tata McGraw Hill Publishers.		

TOPICS BEYOND SYLLABUS/ADVANCED TOPICS:

S.NO	DESCRIPTION	Associated PO & PSO
1	Introduction to Design of Machine Tool Elements.	PO1, PO2, PO3, PSO3

WEB SOURCE REFERENCES:

1	Student Resources provided in NEC website student login: http://www.nrtec.ac.in
2	https://lecturenotes.in/subject/426/design-of-machine-members-dmm
3	https://nptel.ac.in/courses/112106137/
4	http://www.jntumaterials.co.in/2015/03/jntu-mechanical-engineering-design-of.html

5	https://files.jntufastupdates.com/download/design-of-machine-members-ii-lecture-notes/
6	https://nptel.ac.in/downloads/112105125/
7	https://india.oup.com/product/design-of-machine-elements-9780199477647
8	http://www.jntukelearn.in/design-machines-course-detail.html
9	https://www.sanfoundry.com/1000-machine-design-questions-answers/

DELIVERY/INSTRUCTIONAL METHODOLOGIES:

□1Chalk & Talk	□1PPT	□ \Active Learning
□ 1 Web Resources	☐ Students Seminars	☐Case Study
☐Blended Learning	□lQuiz	□\Tutorials * *
☐Project based learning	□1NPTEL/MOOCS	☐ Simulation
☐Flipped Learning	☐Industrial Visit	☐Model Demonstration
☐Brain storming	□Role Play	□Virtual Labs

MAPPING CO'S WITH PO'S:

	PO	PO	PO	PO	PO	PO	PO	PO	РО	PO	PO	PO	PSO	PSO	PSO
СО	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3
CO1	2	3 ;	3	2	-	-	-	-	-	-	-	-	-		3
CO2	2	3	3	2	-		-	-	-	-	-	-	-	-	3
CO3	2	3	3	2	-	-	, -	-	-	-	-	-	-	-	3
CO4	2	3	3	2	-	-	-	-	-	1	-	-	-	-	3
CO5	2	3	3	2	-	-	•	-	-		-	-	-	-	3
Average	2	3 1	3	2	-	-	-	-	-	•	-	-	-	-	3

MAPPING COURSE WITH POS & PSOs:

Cours	PO	PO	PO	PO	PO	PO	PO	PO	PO	PO1	PO1	PO1 2	PSO	PSO	PSO
e	1	2	3	4	5	6	7	8	9	0	1		1	2	3
C311	2	3	1 3	2	-	-	-	-	-	-	-	-	ŧ	-	3

Course	Outcome Assessn	Weighta			
Direct	Cumulative	Descriptive Test			
Assessment	İnternal	Objective Test	30%		Final
	Examinations (CIE)	Assignment Test	3070	90%	Course Outcome
	Semester End Ex	xaminations (SEE)	70%	70%	
Indirect Assessment	Course End Surv		10%		

Rubrics for overall attainment of course outcomes:

If 50% of the students crossed 50% of the marks: Attainment Level 1

If 60% of the students crossed 50% of the marks: Attainment Level 2

If 70% of the students crossed 50% of the marks: Attainment Level 3

Course Instructor

Course Coordinator

Module Coordinator

Head of the Department

ANNEXURE I:

(A) PROGRAM OUTCOMES (POs) Engineering Graduates will be able to:

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

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

mathematics, natural sciences, and engineering sciences.

3. 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.

4. Conduct investigations of complex problems: Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and

synthesis of the information to provide valid conclusions.

5. Modern tool usage: Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modeling to complex engineering activities with an understanding of the limitations.

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.

7. Environment and sustainability: Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the knowledge of, and need for sustainable development.

8. Ethics: Apply ethical principles and commit to professional ethics and responsibilities and

norms of the engineering practice.

9. Individual and team work: Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.

10. Communication: Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation, make effective presentations, and give and receive clear instructions.

- 11. Project management and finance: Demonstrate knowledge and understanding of the engineering and management principles and apply these to one's own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.
- 12. Life-long learning: Recognize the need for, and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.

(B) PROGRAM SPECIFIC OUTCOMES (PSOs):

- 1. The students will be able to understand the modern tools of machining which gives them good expertise on advanced manufacturing methods.
- 2. The students are able to design different heat transfer devices with emphasis on combustion and power production.
- 3. The students are able to design different mechanisms and machine components suitable to automation industry.

Cognitive levels as per Revised Blooms Taxonomy:

Cognitive Domain	Level	Key words
Remembering	K1	Choose, Define, Find, How, Label, List, Match, Name, Omit, Recall, Relate, Select, Show, Spell, Tell, What, When, Where, Which, Who, Why.
Understanding	K2	Classify, Compare, Contrast, Demonstrate, Explain, Extend, Illustrate, Infer, Interpret, Outline, Relate, Rephrase, Show, Summarize, Translate.
Applying	К3	Apply, Build, Choose, Construct, Develop, Experiment With, Identify, Interview, Make Use of, Model, Organize, Plan, Select, Solve, Utilize.
Analyzing	K4	Analyze, Assume, Categorize, Classify, Compare, Conclusion, Contrast, Discover, Dissect, Distinguish, Divide, Examine, Function, Inference, Inspect, List, Motive, Relationships, Simplify, Survey, Take part in, Test for, Theme
Evaluating	K5	Agree, Appraise, Assess, Award, Choose, Compare, Conclude, Criteria, Criticize, Decide, Deduct, Defend, Determine, Disprove, Estimate, Evaluate, Explain, Importance, Influence, Interpret, Judge, Justify, Mark, Measure, Opinion, Perceive, Prioritize, Prove, Rate, Recommend, Rule on, Select, Support, Value
Creating	K6	Adapt, Build, Change, Choose, Combine, Compile, Compose, Construct, Create, Delete, Design, Develop, Discuss, Elaborate, Estimate, Formulate, Happen, Imagine, Improve, Invent, Make up, Maximize, Minimize, Modify, Original, Originate, Plan, Predict, Propose, Solution, Solve, Suppose, Test, Theory

Unit wise Sample assessment questions

COURSE OUTCOMES: Students are able to

CO1: Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings. [K3]

CO2: Identify the loads and stresses while designing the connecting rod and crank shaft. [K3]

CO3: Analyze stresses on the piston and cylinder depending upon Design and proportions. **IK4**1

CO4: Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design. [K3]

CO5: Compare capacities of power transmission of Belt, Rope and Chain Drives. [K2]

S.NO	QUESTION	KNOWLEDGE LEVEL	СО
	UNIT 1		
1	 a) What are rolling contact bearings? Discuss their advantages over sliding contact bearings. b) A ball bearing subjected to a radial load of 5KN is expected to have a life of 8000 hours at 1450 rpm with a reliability of 99 per cent. Calculate dynamic load capacity of the bearing so that it can be selected from manufactures catalogue based on a reliability of 90 per cent. 	K1 & K3	CO1
2	 a) List the important physical characteristics of a good bearing material. b) The main bearing of a steam engine is 100 mm and 175 mm long. The bearing supports a load of 28 KN at 250 rpm. If the ratio of diametral clearance to the diameter is 0.001 and absolute viscosity of the lubricating oil is 0.015 kg/m-s. Find: i) the coefficient of friction and ii) heat generated at the bearing due to friction. 	K1 & K3	CO1
3	a) A 100 mm long and 60 mm diameter journal bearing supports a load of 2500 N at 600 rpm. If the room temperature is 20°C, what should be the viscosity of oil to limit the bearing surface temperature to 60°C? Diametral clearance = 0.06 mm. Energy dissipation coefficient based on the projected area of the bearing = 210 W/m ² °C b) A 250×250 mm bearing carries a load of 108 KN. The bearing rotates at 1500 rpm. The clearance ratio is 670. For full journal bearing the power lost in friction is 14.36 KW. Find the viscosity of the oil.	К3	CO1
4	What are the various terms used the journal bearings analysis and design. Give their definitions in brief.	K1	CO1
	UNIT 2		
1	The following data is given for the cap and bolts of the big end of the connecting rod. Engine speed=1500rpm length of the connecting rod =320mm length of stroke=140mm mass of reciprocating parts=1.75kg length of crank pin=54mm diameter of the crank pin=38mm permissible tensile stress for the bolts=120MPa	К3	CO2

	nomically 1.		
	permissible bending stress for cap=120MPa	_	
	Calculate the nominal diameter of the bolts and thickness of the cap for the big end.		
		<u> </u>	
	a) Explain the design of centre crankshaft when crank is at dead center.		
2	The state of the s		
	b) Explain the design of centre crankshaft when crank is at	K2	CO2
	an angle with the line of dead centre positions and subjected to maximum torsional moment.		
		<u> </u>	
	A connecting rod is to be designed for a high speed, four stroke IC engine. The following data are available.		
	Piston diameter=88mm;		
	Mass of reciprocating parts=1.6kg; Length of the		
	connecting rod (centre to centre) = 200 man Gtv. 10.5		
3	connecting rod (centre to centre)=300mm; Stroke=125mm; RPM=2200 (when developing 50kw);	КЗ	CO2
	Possible over apod—2000——— and the control of the c		002
	Possible over speed=3000rpm; compression ration = 6.8:1; Probable explosion pressure=3.5MPa.		
	Design and draw fully dimensioned describes and		
	Design and draw fully dimensioned drawing of the connecting rod.		
	UNIT 3	<u> </u>	
	The following data is given for the piston of a four stroke		
	diesel engine.		
	Cylinder bore=100 mm		
	Material of piston rings=grey cast iron		
	Allowable tensile stress=90 MPa		
	Allowable radial pressure on the cylinder wall=0.035MPa		
	Thickness of the piston head=16 mm		
	Number of piston rings=4		
1	Calculate (i) Radial width of piston rings (ii) Axial	К3	CO3
	thickness of the piston rings		005
	(iii) Gap between the free ends of the piston ring before		-
	assembly (iv) Gap between the free ends of the piston ring		
	after assembly		
	(v) Width of the top land (vi) Width of ring grooves		
	(vii) Thickness of piston barrel and (viii) Thickness of		
	barrel at top end		
	· · · · · · · · · · · · · · · · · · ·		
	Design a cast iron piston for a four stroke IC engine for the following specifications	<i>\$</i> *	
	Cylinder bore=120 mm		
	Stroke length=150 mm		
2	Maximum gas pressure=5 MPa		
_	Brake mean effective pressure=0.7 MPa	K3	CO3
	Fuel consumption=0.25 Kg/KW/hr		
	Speed=2400 rpm		
	Assume any other data necessary for the design		
	a) Explain the various types of cylinder liners.		
3	b) Discuss the design of piston for an internal combustion	179 0 377	000
_	engine.	K2 & K6	CO3
	The following data is given for the piston of a four stroke		
4	diesel engine.	K3	CO3
	10		1 1

	Cylinder bore = 120 mm	<u> </u>	
	Maximum gas pressure = 6 MPa		
	Allowable bearing pressure for skirt = 0.45 MPa		
	Ratio of side thrust on liner to maximum gas load on		
	piston=0.1		
	Width of top land =20 mm		
	Width of ring grooves = 3 mm		
	Total number of piston rings = 4		
	Axial thickness of piston rings = 3.5 mm		
	Calculate (i)length of skirt (ii) length of piston		
	UNIT 4	-	
	A punch press, used for stamping sheet metal, has a		
	punching capacity of 50 kN. The section of the frame is as		
	shown in Fig. Find the resultant stress at the inner and outer		
	fibre of the section.(All dimensions are in mm)		
1	₹ 800		
_	-/ (5-10		
1	x	K3	CO4
	→ 200 +8 ←		
	<u> </u>		
İ	100 300		
	<u> </u>		
	Section at X-X		
	Determine (i) location of neutral axis, (ii) maximum and		
	minimum stresses when a curved beam of circular section		
	of diameter 60 mm is subjected to pure bending moment of		
2	+ 1350 Nm. The bottom width is towards the centre of	К3	COA
	i I	KS	CO4
	curvature. The radius of curvature is 75 mm and beam is		
	curved in a plane parallel to depth. Also plot the variation		
	of stresses across the section.		
	A central horizontal section of a hook is symmetrical		
	trapezium 90 mm deep. The inner width being 90 mm and		
	outer being 45 mm. The hook carries a load of 67.5 KN, the		
3	load line passes at a distance of 40 mm from the inside edge	К3	CO4
-	of the section. The centre of curvature is in the load line.	_ 	
	Calculate the extreme intensities of stress. Also plot the		
	stress distribution across the section.		
	a) Write and derive the expression for Winkler-Bach		
	formula.		
4		K 1	CO4
	b) What are the assumptions made in the derivation of		
<u> </u>	stresses in a curved bar.		
	UNIT 5		
	A V belt drive is required for a 15 KW, 1440 rpm electric		
1	motor, which drives a centrifugal pump running at 360 rpm	К3	CO5
	for a service of 24 hours per day. From space		
	1		

	considerations, the center distance should be approximately 1m. Determine		T -
į	(i) Belt specifications		
	(ii) Number of belts		
	(iii) Correct center distance and		ļ.
	(iv) Pulley diameters		
2	It is required to select a v-belt drive to connect a 20 KW, 1440 rpm motor to a compressor running at 480 rpm for 15 hours per day. Space is available for a center distance approximately 1.2m. Determine i) The specifications of the belt ii) Diameter of the motor and compressor pulleys iii) The correct center distance and the number of belts	КЗ	CO5
3	A100 mm wide and 10 mm thick belt transmits 5 KW between two parallel shafts. The distance between the shaft centers is 1.5 m and the diameter of the smaller pulley is 440 mm. The driving and driven shafts rotate at 60 rpm and 150 rpm respectively. Find the stress in the belt, if the two pulleys are connected by i) open belt ii) a cross belt. The coefficient of friction is 0.22	К3	CO5



(AUTONOMOUS)

III B.Tech II Semester Regular Examinations, Month/Year:
Sub Code: 19BME6TH01 SUBJECT NAME: DESIGN OF MACHINE

ELEMENTS-II

(ME) MODEL PAPER-I

Time: 3 hours Max. Marks: 60

> Note: Answer All FIVE Questions. All Questions Carry Equal Marks (5 X 12 = 60M)

Q. NO		QUESTION	KL	СО	MARKS
		Unit - I	L		<u> </u>
1	a	Solve the following problem. Design a journal bearing for a centrifugal pump running at 1440 r.p.m. The diameter of the journal is 100 mm and load on each bearing is 20 kN. The factor ZN/p may be taken as 28 for centrifugal pump bearings. The bearing is running at 75°C temperature and the atmosphere temperature is 30°C. The energy dissipation coefficient is 875 W/m²/°C. Take diametral clearance as 0.1 mm.	К3	CO1	[12M]
		OR		<u> </u>	<u> </u>
-	b	Solve the following problem. Select a single row deep groove ball bearing for a radial load of 4000 N and an axial load of 5000 N, operating at a speed of 1600 r.p.m. for an average life of 5 years at 10 hours per day. Assume uniform and steady load.	К3	CO1	[12M]
		Unit - II			
2	a	Solve the following problem. Design a suitable connecting rod for a car with the following data: Piston diameter = 68mm. Stroke length = 80 mm. Length of connecting rod = 160mm. Maximum explosion pressure = 3.5 Mpa. Weight of the reciprocating parts = 2.5 kg Speed =4000 rpm. Compression ratio 8:1	K3	CO2	[12M]
		OR			
	b d	Explain the design of Center Crankshaft when the crank shaft is at dead center.	K2	CO2	[12M]
3		Unit - III			

2 E

	т-				, ,							
		Solve the following problem.										
		A horizontal four stroke diesel engine has the										
		following specification.										
	a	Brake power = 5 kW., Speed = 1200 r.p.m	K3	CO3	[12M]							
		Imep = 0.35 N/mm ² , Mechanical Efficiency = 80 %.										
		Calculate the dimensions of the explinder and the										
		cylinder head.										
		OR		·								
		Solve the following problem.										
		Design a plain carbon steel center crank shaft when										
		the crank is at dead center for a single acting 4 stroke										
		single cylinder engine for the following data:										
	Ь	Piston diameter =250 mm.	K3	CO3	[12M]							
	-	Stroke = 400 mm	113	003	[12141]							
		Maximum combustion pressure =2.5 N/mm2										
		Weight of the fly wheel =16 kN.										
		Total belt pull =3 N										
	 	Length of the connecting rod = 950 mm. Unit - IV		<u> </u>								
	-	Solve the following problem.		· · ·								
	ŀ	,										
		A curved beam of rectangular section of 30 mm width and 40 mm depth and mean radius of curvature of 60			ı							
		mm is initially unstressed. If a bending moment of 400										
	a	N-m is applied to the beam which tends to straighten	K3	CO4	[12M]							
	"	it, determine the stresses at the inner and outer surface	125	004	[12141]							
4		and sketch a diagram to show the variation of stresses										
		across the section. Also find the position of Neutral		į								
		axis.										
		OR										
	1-	What are the assumptions made in Winkler Bach										
	b	theory? Derive an expression for radius of neutral axis	K2	CO4	[12M]							
		for T section.										
		Unit - V										
	a	Derive the expression for length of on Open belt drive	K5	CO5	[12M]							
	-	& Cross belt drive.		,	[
		OR										
		Solve the following problem.										
		In a cross belt drive the diameter of the driver and										
		follower pulleys are 200mm and 400mm respectively.										
		The centre distance of the driver is 2m. The driver										
_	b	pulley rotates at 400rpm. Find the angle of contact between belt and both the pulleys and the length of the	K3	CO5	[12M]							
5		belt required .What is he power capacity of the driver,										
		if the permissible tension in the belt is 1.2kn and										
		coefficient of friction between belt and pulleys is										
		0.252.										

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(AUTONOMOUS)

III B.Tech II Semester Regular Examinations, Month/Year:
Sub Code: 19BME6TH01 SUBJECT NAME: DESIGN OF MACHINE

ELEMENTS-II

(ME) MODEL PAPER-II

Time: 3 hours Max. Marks: 60

> Note: Answer All FIVE Questions. All Questions Carry Equal Marks (5 X 12 = 60M)

Q. NO		QUESTION	KL	СО	MARKS
		Unit - I	<u> </u>	1	<u> </u>
1	A journ has a b journal the dia lubricat operatin kg/m-s. the amo (ii) the differen the oil	al bearing of 50mm diameter and 80mm long, earing pressure of 6 N/mm ² , the speed of the is 1000 rpm. The ratio of journal diameter to metral clearance is 1000. The bearing is ed with oil, whose absolute viscosity at the g temperature of 75°C may be taken as 0.015. The room temperature is 25°C, Determine (i) unt of artificial cooling required and mass of the coolant oil required, if the ce between the outlet and inlet temperature of is 10°C, the specific heat of the oil is 1900 and heat dissipation coefficient is 500 c.	K3	CO1	[12M]
		OR			
	A shaft variable subjecte for 10 pc kN for 3 time of c is 20 × 1	rotating at constant speed is subjected to load. The bearings supporting the shaft are d to stationary equivalent radial load of 3 kN er cent of time, 2 kN for 20 per cent of time, 1 0 per cent of time and no load for remaining cycle. If the total life expected for the bearing 06 revolutions at 95 per cent reliability, a dynamic load rating of the ball bearing.	К3	CO1	[12M]
		Unit - II			
2	a A horiz	e following problem. contal four stroke diesel engine has the g specification.	К3	CO2	[12M]

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		Brake power = 6 kW. , Speed = 1800 r.p.m				
		Imep = 0.45 N/mm ² , Mechanical Efficiency = 75 %.				
		Calculate the dimensions of the cylinder and the				
		cylinder head.				
		OR				
		Solve the following problem.				
		Design a plain carbon steel center crank shaft when				
		the crank is at dead center for a single acting 4 stroke				
		single cylinder engine for the following data:			•	
	b	Piston diameter =275 mm. Stroke = 420 mm	K3	CO2	[12M]	
		Maximum combustion pressure =2.5 N/mm ²				
		Weight of the fly wheel =15 kN.				
		Total belt pull =2.5 N				
		Length of the connecting rod = 850 mm.				
	<u> </u>	Unit - III		1		
	a	Explain in detail the design procedure of Cylinder.	K2	CO3	[12M]	
	\vdash	OR				
		Solve the following problem.				
		Design a cast iron piston for a single acting four stroke				
3		engine for the following data:				
3	b	Cylinder bore = 100 mm; Stroke = 125 mm;				
		Maximum gas pressure = 5 N/mm2; Indicated mean	K3	CO3	[12M]	
		effective pressure = 0.75 N/mm ² ; Mechanical				
		efficiency = 80%; Fuel consumption = 0.15 kg per				
		brake power per hour; Higher calorific value of fuel = 42×10^3 kJ/kg; Speed = 2000 r.p.m. Any other data				
		required for the design may be assumed.				
	 	Unit - IV		I	<u> </u>	
		Solve the following problem.	•			
	1	A curved bar of rectangular section, initially				
		unstressed, is subjected to bending moment of 1400 N-				
		m which tends to straighten the bar. The section is 4				
,	a	cm wide by 5 cm deep in the plane of bending, and the	K3	CO4	[12M]	
4		mean radius of curvature is 10 cm. Find the position of				
		the neutral axis and magnitudes of the greatest bending stress and draw a diagram to show approximately how				
		the stress varies across the section.				
		OR		l		
	ь	Derive an expression for radius of neutral axis for	V5	COA	[10] II	
	D	circular section.	K5	CO4	[12M]	
•		Unit - V				
	a	What are different types of Flat Belt Drives? Explain	K2	CO5	[12M]	
5	<u> </u>	any four with neat sketch.				
	OR Solve the following problem.					
	b	Two pulleys, one 450 mm diameter and the other 200	K3	CO5	[12M]	
		1 wo puncys, one 450 mm diameter and the other 200			_	

	mm diameter, on parallel shafts 1.95 m apart are		
	connected by a crossed belt. Find the length of the belt		
٠	required and the angle		
	of contact between the belt and each pulley. What		
	power can be transmitted by the belt when the larger		
ļ	pulley rotates at 200 rev/min, if the maximum		
	permissible tension in the belt is 1 kN, and the		
	coefficient of friction between the belt and pulley is		
	0.25?		

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ACADEMIC CALENDAR



ACADEMIC CALENDAR FOR HI B.TECH I & H SEMESTERS ACADEMIC YEAR 2021-22

ISEM	IESTER		Weeks
Description	From Date	To Date	AA CER2
Commencement of Class Work	04-10-2021		
ommencement of Sacratic Sacrat	04-10-2021	20-11-2021	1
1st Spell of Instructions	18-10-2021	23-10-2021	7 Weeks
I Assignment Examination	08-11-2021	13-11-2021	
II Assignment Examination	22-11-2021	27-11-2021	1 Week
I Mid Examination	28-11-2021	15-01-2022	
2 nd Spell of Instructions	13-12-2021	18-12-2021	7 Weeks
III Assignment Examination	03-01-2022	08-01-2022	
IV Assignment Examination	17-01-2022.	22-01-2022	1 Week
II Mid Examination	24-01-2022	29-01-2022	1 Week
Preparation & Practicals	31-01-2022	12-02-2022	2 Weeks
End Examinations	EMESTER		
	14-02-2022		AA
Commencement of II Semester Class work	14-02-2022	02-04-2022	
1st Spell of Instructions	28-02-2022	05-03-2022	7 Weeks
I Assignment Examination	21-03-2022	26-03-2022	1
II Assignment Examination	04-04-2022	09-04-2022	1 Week
I Mid Examination	11-04-2022	28-05-2022	
2 nd Spell of Instructions	25-04-2022	30-04-2022	7 Weeks
III Assignment Examination	16-05-2022	21-05-2022	1
IV Assignment Examination	30-05-2022	04-06-2022	1 Week
II Mid Examination	06-06-2022	11-06-2022	1 Week
Preparation & Practicals	13-06-2022	25-06-2022	2 Weeks
End Examinations Commencement of IV YEAR I SEM Class work			

PRINCIPAL



TIME TABLE

NARASARAOPETA ENGINEERING COLLEGE: NARASARAOPET (AUTONOMOUS) DEPARTMENT OF MECHANICAL ENGINEERING III B.TECH II SEM TIME TABLE

ROOM N	0:1321			TIT D' LECE	III SEM T	IME TARI	ETAREKIN	G			
	1	7			Section-B		71 0				
TIMINGS	9.10-	10.00-	BREAK	3				YY2 0			
	10.00	10.50	10.50-11.00	11 00 11 70	4		5	Wef: 12/	12/2022		
MON	D	M		11.00-11.50	11.50-12.40	12.40-1.30	120000	0	7	8	٦
† TUE	ROE	3&A	В	H		2.50	1.30-2.20	2.20-3.10	3.10-4.00		\dashv
WED			$egin{array}{cccccccccccccccccccccccccccccccccccc$. DOI	M			EES LAB		<u> </u>	
THU	DMI		A			L *	HT	DM	Υ	Honors/Minors	
	DO	M	· K	ROB		Ū		T	DME-II	Honors/Minors	1
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SAT	DME-II			DM	DME-II	C	ROB	&A	Honor	s/Minors	ľ
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CODE						<u> </u>			LAB	7	

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. ROB&A	
DMF	
MT&MD LAB	
HT LAB	
P&HVAC LAB	
EES LAB	
EITK	

Signature of HOD

SUBJECT

Design of Machine Elements -II

Heat Transfer

Dynamics of Machinery

Robotics & Applications

Digital Marketing

Machine Tools & Machine Dynamics Lab

Heat Transfer Lab

Pneumatics & HVAC Lab

English Employability Skills Lab

Essence of Indian Traditional Knowledge

FACULTY

Dr.S.Jaya Krishna

DOM

Mr.T.Narendar

Dr.M.Venkanna Babu

Dr.P.Suresh Babu

Dr.Y.Siva Reddy

Mr.P.Narendar/R.Chinna Rao

Mr.K.Govardhan Reddy/B.Ajay Kumar

Mr.K.Kiran Chand/MD.Taju

Mr.A.Venu Gopal

Mr.D.Satish

Signature of Principal

HT

Honors/Minors

NARASARAOPETA ENGINEERING COLLEGE: NARASARAOPET (AUTONOMOUS) DEPARTMENT OF MECHANICAL ENGINEERING III B.TECH II SEM TIME TABLE

RO	OM NO): 1320				Section-A		414			•	
TIM	IINGS	9.10-	2	BREAK	3	. 4	T	5	Wef: 12/	12/2022		
ļ		10.00	10.00-10.50	10.50-11.00	11.00-11.50	11.50-12.40	12.40-	 	6	7	8	
M	ION	RO	B&A		EITK		1.30	1.30-2.20	2.20-3.10	3.10-4.00		
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CODE

DME-II HT

DOM

· ROB&A

DM

MT&MD LAB

HTLAB

P&HVAC-LAB EES LAB

EITK

SUBJECT

Design of Machine Elements -II

Heat Transfer

Dynamics of Machinery

Robotics & Applications

Digital Marketing

Machine Tools & Machine Dynamics Lab

Heat Transfer Lab-

Pneumatics & HVAC Lab

English Employability Skills Lab

Essence of Indian Traditional Knowledge

FACULTY

Dr.T.Raja Santosh Kumar

Dr.D.Jägadish

Dr.B.Ravi Naik

Mr.T.V.Rao

Mrs:SD.Salma

Mr.M.Srinivasa Rao/N.Arun Kumar

Mr.K. K:John Babu /E.Saidulu

Mr. M. Venkaiah/P. Sravani Mr.A.Venu Gopal

Mr.D.Satish

Signature of HOD

Signature of Hrincipal

Minors



SYLLABUS COPY

Code: R20ME3201	2	1_1_	<u> </u>	DESIGN OF	MACHINE ELF Data Book is all		3	
II-SEMESTER	12	1	0	MARKS 30	MARKS MARKS		CREDITS	
III B.TECH	T	т	D	INTERNAL	EXTERNAL	TOTAL		

COURSE OUTCOMES:

- CO1: Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings.
- CO2: Identify the loads and stresses while designing the connecting rod and crank shaft.
- CO3: Analyze stresses on the piston and cylinder depending upon Design and proportions.
- CO4: Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design.
- CO5: Compare capacities of power transmission of Belt, Rope and Chain Drives.

UNIT - I:

ESIGN OF BEARINGS: Types of bearings, Lubrication, Bearing Modulus, Full and partial bearings, Clearance ratio, Heat dissipation of bearings, bearing materials, Journal bearing design, Ball and roller bearings, Static loading of ball & roller bearings, Bearing life.

UNIT - II:

DESIGN OF I.C ENGINE COMPONENTS: Connecting Rod, Thrust in connecting rod, Stress due to whipping action on connecting rod ends, Cranks and Crank shafts- strength and proportions of over hung and centre cranks, Crank pins.

UNIT - III: PISTONS AND CYLINDERS

Pistons, Forces acting on piston, Construction, Design and proportions of piston, Cylinder Cylinder liners.

UNIT-IV:

DESIGN OF CURVED BEAMS: introduction, stresses in curved beams, Expression for radius of neutral axis for rectangular, circular, trapezoidal and T-Section, C – clamps. Design of crane hook.



POWER TRANSMISSION SYSTEMS: Transmission of power by Belt and Rope drives, Transmission Efficiencies, Belts – Flat and V types – Ropes - pulleys for belt and rope drives, Materials, Chain Drives, Tensions of Belt and Chain Drives.

TEXT BOOKS:

- 1. Machine Design, V.B. Bandari, Tata McGraw Hill Publishers.
- 2. Machine Design, Pandya & Shaw, Charotar publishers.



REFERENCES:

- 1. Machine Design, R.N. Norton, Pearson Publishers.
- 2. Design Data hand book. S.Md.Jalaludeen, Anuradha Publishers.
- 3. Mechanical Engineering Design, JE Shingly, Tata McGraw Hill Publishers.

WEB REFERENCES:

- 1. https://www.digimat.in/nptel/courses/video/112105124/L01.html
- 2. http://nptel.vtu.ac.in/econtent/courses/ME/06ME61/index.php
- 3. http://creativestellars.blogspot.com/p/design.html
- 4. https://www.jntumaterials.co.in/2015/04/jntujntuk-design-of-machine-members-i.html
- 5. https://lecturenotes.in/subject/549/design-of-machine-members-i



Code: 19BME6TH01	(Design Data Book is allowed)								
	DESIGN OF MACHINE ELEMENTS -2								
	2	1	0	40	60	100	3		
III B.TECH-II-SEMESTER	L	ı	P	MARKS	MARKS	MARKS 100	CIGEDITS		
	r		D	INTERNAL	EXTERNAL	TOTAL	CREDITS		

COURSE OUTCOMES:

CO1: Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings.

CO2: Identify the loads and stresses while designing the connecting rod and crank shaft.

CO3: Analyze stresses on the piston and cylinder depending upon Design and proportions.

CO4: Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design.

CO5: Compare capacities of power transmission of Belt, Rope and Chain Drives.

UNIT-I:

DESIGN OF BEARINGS: Types of Journal bearings, Lubrication, Bearing Modulus, Full and partial bearings, Clearance ratio, Heat dissipation of bearings, bearing materials, Journal bearing design, Ball and roller bearings, Static loading of ball & roller bearings, Bearing life.

UNIT-II:

DESIGN OF I.C ENGINE COMPONENTS: Connecting Rod, Thrust in connecting rod, Stress due to whipping action on connecting rod ends, Cranks and Crank shafts- strength and proportions of over hung and centre cranks, Crank pins.

UNIT - III: PISTONS AND CYLINDERS

Pistons, Forces acting on piston, Construction, Design and proportions of piston, Cylinder liners.

UNIT-IV:

DESIGN OF CURVED BEAMS: introduction, stresses in curved beams, Expression for radius of neutral axis for rectangular, circular, trapezoidal and T-Section, C – clamps. Design of crane hook.

UNIT - V:

POWER TRANSMISSION SYSTEMS: Transmission of power by Belt and Rope drives, Transmission Efficiencies, Belts – Flat and V types – Ropes - pulleys for belt and rope drives, Materials.



TEXT BOOKS:

- 1. Machine Design, V.B. Bandari, Tata McGraw Hill Publishers .
- 2. Machine Design, Pandya & Shaw, Charotar publishers.

REFERENCES:

- 1. Machine Design, R.N. Norton, Pearson Publishers.
- 2. Design Data hand book. S.Md.Jalaludeen, Anuradha Publishers.
- 3. Mechanical Engineering Design, JE Shingly, Tata McGraw Hill Publishers.

WEB REFERENCES:

https://www.digimat.in/nptel/courses/video/112105124/L01.html

http://nptel.vtu.ac.in/econtent/courses/ME/06ME61/index.php

http://creativestellars.blogspot.com/p/design.html

https://www.jntumaterials.co.in/2015/04/jntujntuk-design-of-machine-members-i.html

https://lecturenotes.in/subject/549/design-of-machine-members-i





LESSON PLAN



Narasaraopeta Engineering College (Autonomous) Yallmanda (Post), Narasaraopet - 522601

DEPARTMENT OF MECHANICAL ENGINEERING LESSON PLAN

Course Code	Course Title (Regulation)	Sem	Branch	Contact Periods/Week	Sections
R20ME3201	DESIGN OF MACHINE ELEMENTS -2 (R20)	VI	Mechanical Engineering	-6	A & B

COURSE OUTCOMES: Students are able to

CO1: Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings. [K3]

CO2: Identify the loads and stresses while designing the connecting rod and crank shaft. [K3]

CO3: Analyze stresses on the piston and cylinder depending upon Design and proportions. [K4]

CO4: Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design. [K3]

O5: Compare capacities of power transmission of Belt, Rope and Chain Drives. [K2]

Unit No	Outcome		Topics/Activity	Ref Text book	Total Periods	Delivery Method	
		1	UNIT-1: DESIGN OF BE	ARINGS			
			Types of Journal bearings	T1, T2, R1,R3	2	PPT	
	CO 1.	1.2	,	T1, T2, R1,R3	2	Chalk, Talk, PPT	
	Apply the design procedure to engineering	1.3	Full and partial bearings, Clearance ratio	T1, T2, R1,R3	2	Chalk, Talk, PPT	
1	problems including the	1.4	Heat dissipation of bearings, bearing materials	T1, T2, R1,R3	2	PPT & Tutorial	
:		1.5	Journal bearing design	T1, T2,	2	Chalk, Talk, PPT &.Tutorial	
	•	1.6	Ball and roller bearings, Static loading of ball & roller bearings	T1, T2, R1,R3	2	Chalk, Talk, PPT & Tutorial	
		1.7	Bearing life.	T1, T2, R1,R3	2	PPT	
		UN	IT-2: DESIGN OF I.C ENGINE C	OMPONE	NTS		
		2.1	Connecting Rod, Thrust in connecting rod,	T1, T2, R1,R3	2	Chalk, Talk, PPT	
		2.2	Stress due to whipping action on connecting rod ends	T1, T2, R1,R3	3	PPT & Tutorial	
		2.3	Cranks and Crank shafts,	T1, T2, R1,R3	2	PPT	
		2.4	Strength and proportions of over hung crank shaft	T1, T2, R1,R3	2	Chalk, Talk	

		2.4	Strength and proportions of over hung crank shaft	T1, T2, R1,R3	2	Chalk, Talk
	<u>'</u>		Centre cranks, Crank pins, Crank shafts.	T1, T2, R1,R3	3	PPT & Tutorial
•	<u>CO 3.</u>		UNIT-3. PISTON AND CY	LINDER	•	1
3	Analyze stresses on the piston and cylinder	3.1	Pistons	T1, T2, R1,R3	3	Chalk, Talk,
	depending upon Design and proportions. [K4]	3.2	Forces acting on piston, Construction	T1, T2, R1,R3	3	PPT & Tutorial
		IEX	AMINATION DURING SIXTH W	EEK		
3	Analyze stresses on the piston and cylinder	3.3	Design and proportions of piston	T1, T2, R1,R3	3	Chalk, Talk,
	depending upon Design and proportions. [K4]	3.4	Cylinder. Cylinder liners. UNIT - 4: DESIGN OF CURV	T1, T2, R1,R3	3	PPT & Tutorial
	CO 4. *	4:1	Introduction,	T1, T2, R1,R3'	2	Chalk, Talk,
	Identify the loads and machine members	4.2	Stresses in curved beams.	T1, T2, R1,R3	2	PPT & Tutorial
4	subjected and calculate static and dynamic stresses to ensure safe	4.3	Expression for radius of neutral axis for rectangular, circular, trapezoidal and T-Section,	T1, T2, R1,R3	3	Chalk, Talk, & Tutorial
·	design. [K3]	4.4	C – Clamps.	T1, T2, R1,R3	2	PPT
		4.5	Design of crane hook.	T1, T2, R1,R3	3	Chalk, Talk
	<u>CO 5.</u>	5.1	Transmission of power by Belt and Rope drives	T1, T2, R1,R3	2	Chalk, Talk,
5	Compare capacities of power transmission of	5.2	Transmission Efficiencies	T1, T2, R1,R3	2	PPT & Tutorial
	Belt, Rope and Chain Drives. [K2]	5.3	Belts – Flat and V types	T1, T2, R1,R3	3	Chalk, Talk,
			Ropes - pulleys for belt and rope drives, Materials,	T1, T2, R1,R3	3	PPT
		5.5	Chain drives.		2	PPT
	MID II EX	KAM	INATION DURING EIGHTEENT	H WEEK		•
			END EXAMINATIONS			

Text Books:

- Machine Design, V.B. Bandari, Tata McGraw Hill Publishers. Machine Design, Pandya & Shaw, Charotar publishers.

Reference Books:

- Machine Design, R.N. Norton, Pearson Publishers. R1
- R2 Design Data hand book. Mahadevan, CBS Publishers.
- Mechanical Engineering Design, JE Shingly, Tata McGraw Hill Publishers.



Narasaraopeta Engineering College (Autonomous)

Yallmanda (Post), Narasaraopet - 522601

DEPARTMENT OF MECHANICAL ENGINEERING LESSON PLAN

Course Code	Course Title (Regulation)	Sem	Branch &	Contact Periods/Week	Sections
19BME6TH01	DESIGN OF MACHINE ELEMENTS -2 (R19)	VI	Mechanical Engineering	5	A & B

COURSE OUTCOMES: Students are able to

CO1: Apply the design procedure to engineering problems, including the consideration of technical and manufacturing constrains of bearings. [K3]

CO2: Identify the loads and stresses while designing the connecting rod and crank shaft. [K3]

CO3: Analyze stresses on the piston and cylinder depending upon Design and proportions. [K4]

CO4: Identify the loads and machine members subjected and calculate static and dynamic stresses to ensure safe design. [K3]

CO5: Compare capacities of power transmission of Belt, Rope and Chain Drives. [K2]

Unit No	Outcome		Topics/Activity	Ref Text book	Total Periods	Delivery Method		
		ļ	UNIT-1: DESIGN OF BEARINGS					
		1.1	Types of Journal bearings	T1, T2, R1,R3				
	CO 1. Apply the design	1.2	Lubrication, Bearing Modulus	T1, T2, R1,R3				
	procedure to engineering problems,	1.3	Full and partial bearings, Clearance ratio	T1, T2, R1,R3				
1	including the	consideration of	1.4	Heat dissipation of bearings, bearing materials	T1, T2, R1,R3	12	Chalk, Talk,	
		1.5	Journal bearing design	T1, T2, R1,R3		PPT & Tutorial		
		1.6	Ball and roller bearings, Static loading of ball & roller bearings	T1, T2, R1,R3				
		1.7	Bearing life.	T1, T2, R1,R3				
	i	UN	T-2: DESIGN OF I.C ENGINE	COMPO	NENTS			
	<u>CO 2.</u>	2.1	Connecting Rod, Thrust in connecting rod,	T1, T2, R1,R3				
2	Identify the loads and stresses while designing	2.2	Stress due to whipping action on connecting rod ends	T1, T2, R1,R3		Chalk, Talk,		
	the connecting rod and crank shaft. [K3]	2.3	Cranks and Crank shafts,	T1, T2, R1,R3	12	PPT & Tutorial		
			Strength and proportions of over hung crank shaft	T1, T2, R1,R3		- 333741		

		2.5	Centre cranks, Crank pins,	T1, T2,				
	CO 3	-	Crank shafts.	R1,R3				
	CO 3.	-	UNIT-3. PISTON AND CYLINDER					
3	Analyze stresses on the	3.1	Pistons	T1, T2,		Chalk,		
	piston and cylinder depending upon Design			R1,R3	6	Talk,		
	and proportions. [K4]		Forces acting on piston,	T1, T2,	0	PPT &		
		TIXE A	Construction	R1,R3		Tutorial		
	MIDI	EXA	MINATION DURING SIXTH	WEEK		•		
	Analyze stresses on the	3.3	Design and proportions of	T1, T2,		Chalk,		
3	piston and cylinder		piston	R1,R3		Talk,		
	depending upon Design	3.4	Cylinder. Cylinder liners.	T1, T2,	6	PPT &		
	and proportions. [K4]			R1,R3		Tutorial		
		-	UNIT – 4: DESIGN OF CUR	VED BEA	MS	7		
		4.1	Introduction,	T1, T2,				
	CO 4. Identify the loads and			R1,R3				
		4.2	Stresses in curved beams.	T1, T2,				
	machine members			R1,R3				
4	subjected and calculate	$\begin{vmatrix} 4 & 3 \end{vmatrix}$ n	Expression for radius of			Chalk,		
	static and dynamic stresses to ensure safe		neutral axis for rectangular,	T1, T2,	12	Talk,		
			circular, trapezoidal and T-Section,	R1,R3	12	PPT &		
	design. [K3]		Section,			Tutorial		
		4.4	C – Clamps.	T1, T2,				
			•	R1,R3				
	_	4.5	Design of crane hook.	T1, T2, R1,R3				
		TT						
			NIT - 5: POWER TRANSMISS	SION SYS	ГЕMS			
	<u>CO 5.</u>	5.1	Transmission of power by Belt and Rope drives	T1, T2,				
	Compare capacities of		and Rope drives	R1,R3				
5	power transmission of	5.2	Transmission Efficiencies	T1, T2,		Chalk,		
	Belt, Rope and Chain Drives. [K2]			R1,R3	12	Talk,		
		5.3	Belts – Flat and V types	T1, T2,	12	PPT &		
-2				R1,R3		Tutorial		
			Ropes - pulleys for belt and	T1, T2,				
			rope drives, Materials,	R1,R3				
	MID II PY	5.5	Chain drives.					
	WIID II EXA	MINA	ATION DURING EIGHTEENT	TH WEEK				
]	END EXAMINATIONS					

Text Books:

- T1 Machine Design, V.B. Bandari, Tata McGraw Hill Publishers.
- T2 Machine Design, Pandya & Shaw, Charotar publishers.

Reference Books:

- R1 Machine Design, R.N. Norton, Pearson Publishers.
- R2 Design Data hand book. Mahadevan, CBS Publishers.
- R3 Mechanical Engineering Design, JE Shingly, Tata McGraw Hill Publishers.

Principal

HOD

Faculty



CO-POs & CO-PSOs MAPPING (COURSE ARTICULATION MATRIX)



DEPARTMENT OF MECHANICAL ENGINEERING COURSE ARTICULATION MATRIX

R20-REGULATION

Explanation of Course Articulation Matrix Table to be ascertained:

- Course Articulation Matrix correlates the individual COs of a course with POs and PSOs.
- > The Course Outcomes are mapped with POs and PSOs in the scale of 1 to 3.
- > The strength of correlation is indicated as 3 for Substantial (High) correlation, 2 for Moderate (Medium) correlation, and 1 for Slight (Low) correlation.

III B.Tech II SEMESTER DESIGN OF MACHINE ELEMENTS -II

i	rse Co C321	ode:		C	ourse	Nan	ie: D	ESIG	N OI	F MAC	CHINE	ELEN	MENT	S-II	
CO-					1.0			POs &				,	~~~~		
COs	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO ₂	PSO3
C321.1	2	3	3	2	-	-	-	-	-	-	1	-	2	2	3
C321.2	2	3	3	2	-	-	-	-	-	-	-	-	2	2	3
C321.3	2	3	3	2	-	_	-	-	-	-	-	-	2	2	3
C321.4	2	3	3	2	-	-	-	-	-	-	-	-	2	-	3
C321.5	2	3	3	-	-	-	-	-	-	-	-	-	-	-	3
C321	2.00	3.00	3.00	2.00	-	-	-	-	-	-	-	1	2.00	2.00	3.00



WEB REFERENCES

DESIGN OF MACHINE ELEMENTS-2

(Design Data Hand Book Allowed)

List of E-Resources:

1	Student Resources provided in NEC website student login: http://www.nrtec.ac.in
2	https://lecturenotes.in/subject/426/design-of-machine-members-dmm
3	https://nptel.ac.in/courses/112106137/
4	http://www.jntumaterials.co.in/2015/03/jntu-mechanical-engineering-design-of.html
5	https://files.jntufastupdates.com/download/design-of-machine-members-ii-lecture-
	notes/
6	https://nptel.ac.in/downloads/112105125/
7	https://india.oup.com/product/design-of-machine-elements-9780199477647
8	http://www.jntukelearn.in/design-machines-course-detail.html
9	https://www.sanfoundry.com/1000-machine-design-questions-answers/



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5	19471A0305	CHAVA ASHOK
6	19471A0306 +	CHIRUGURI KARUNAKAR
7	19471A0307	DURGAMPUDI MAHESH REDDY
8	19471A0308	GANGAVARAPU SRI CHANDRASEKHAR
9	19471A0309	GANNEPALLI RAVI
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16	19471A0317	JONNALAGADDA MADHU
17	19471A0318	KAKANI NAGENDRA BABU
18	19471A0319	KAMBAMPATI AJITHKUMAR
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22	19471A0323	MAHANKALI RAKESH
23	19471A0324	MALLAVARAPU PRABHAKAR
24	19471A0326	MELAM STEPHEN WILLIAMS
25	19471A0327	NARENDRA BABU SADHE
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27	19471A0329	ONTERU VEERANJANEYULU
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31	19471A0335	RAMAR SATISH KUMAR

	32	19471A0336	SAVALAM MANI KUMAR
	33	19471A0337	SHAIK JILANI
	34	19471A0338	SHAIK MAHAMMAD BILAL
· ·	35	19471A0339	SHAIK MAHAMMAD RIYAZ
, d	36	19471A0340	SHAIK SUBHANI
	37	19471A0341	TALAKAYALA VINAY KUMAR
	38	19471A0342	VADLAMUDI YASWANTH SAI
	39	19471A0343	VEERLA KOTESWARA RAO
0	40	19471A0344	VEJARLA AVINASH
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	48	20475A0361	KUMMARA PARAMESWARA RAO
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6	20475A0306	THAPPETA RADHAKRISHNA .
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24	20 <mark>475A0324</mark>	NUNNA BALA NAVEEN
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26	20475A0326	NAGISETTY RAKESH
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35	20475A0336	MIDDELA BAJIVALI
36	20475A0337	VATTIGORLA YOGANJANEYULU
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45	20475A0346	KANAPARTHI VENKATA KRISHNA
46	20475A0347	SHAIK AMEER
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HAND WRITTEN/PRINTED LECTURE NOTES

UNIT 1 JOURNAL BEARINGS

1.1 WHY TO STUDY FRICTION, WEAR & LUBRICATION?

Moving parts of every machine is subjected to friction and wear. Friction consumes and wastes energy. Wear causes changes in dimensions and eventual breakdown of the machine element and the entire machine. The loss of just a few milligrams of material in the right place, due to wear can cause a production machine or an automobile to be ready for replacement. If we imagine the amount of material rendered useless by way of wear, it is startling! Lots of materials ranging from Antimony to zinc, including titanium, vanadium, iron, carbon, copper, aluminum etc., would be lost. It is therefore essential to conserve the natural resources through reduction in wear. Lubrication plays a vital role in our great and complex civilization.

1.2 BEARINGS

A bearing is machine part, which support a moving element and confines its motion. The supporting member is usually designated as bearing and the supporting member may be journal. Since there is a relative motion between the bearing and the moving element, a certain amount of power must be absorbed in overcoming friction, and if the surface actually touches, there will be a rapid wear.

1.2.1 Classification: Bearings are classified as follows:

1. Depending upon the nature of contact between the working surfaces:-

- a) Sliding contact bearings
- b) Rolling contact bearings.

a) SLIDING BEARINGS:

- · Hydrodynamically lubricated bearings
- Bearings with boundary lubrication
- Bearings with Extreme boundary lubrication.
- Bearings with Hydrostatic lubrication.

b) ROLLING ELEMENT BEARINGS:

- Ball bearings
- Roller bearings
- Needle roller bearings

1. Based on the nature of the load supported:

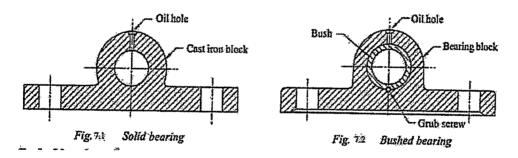
- Radial bearings Journal bearings
- Thrust bearings
 - Plane thrust bearings
 - Thrust bearings with fixed shoes
 - Thrust bearings with Pivoted shoes
- Bearings for combined Axial and Radial loads.

JOURNAL BEARING:

It is one, which forms the sleeve around the shaft and supports a bearing at right angles to the axis of the bearing. The portion of the shaft resting on the sleeve is called the journal. **Example of journal bearings are-** Solid bearing, Bushed bearing, and Pedestal bearing.

Solid bearing:

A cylindrical hole formed in a cast iron machine member to receive the shaft which makes a running fit is the simplest type of solid journal bearing. Its rectangular base plate has two holes drilled in it for bolting down the bearing in its position as shown in the figure 1.1. An oil hole is provided at the top to lubricate the bearing. There is no means of adjustment for wear and the shaft must be introduced into the bearing endwise. It is therefore used for shafts, which carry light loads and rotate at moderate speeds.



Bushed bearing:

It consists of mainly two parts, the cast iron block and bush; the bush is made of soft material such as brass, bronze or gunmetal. The bush is pressed inside the bore in the cast iron block and is prevented from rotating or sliding by means of grub- screw as shown if the figure 1.2. When the bush gets worn out it can be easily replaced. Elongated holes in the base are provided for lateral adjustment.

Pedestal bearing:

It is also called Plummer block. Figure 1.3 shows half sectional front view of the Plummer block. It consists of cast iron pedestal, phosphor bronze bushes or steps made in two

halves and cast iron cap. A cap by means of two square headed bolts holds the halves of the steps together. The steps are provided with collars on either side in order to prevent its axial movement. The snug in the bottom step, which fits into the corresponding hole in the body, prevents the rotation of the steps along with the shaft. This type of bearing can be placed any where along the shaft length.

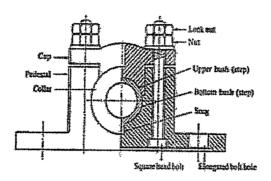


Fig 1.3: Pedestal Bearing

Thrust bearing:

It is used to guide or support the shaft, which is subjected to a load along the axis of the shaft. Since a thrust bearing operates without a clearance between the conjugate parts, an adequate supply of oil to the rubbing surfaces is extremely important. Bearings designed to carry heavy thrust loads may be broadly classified in to two groups-

FOOT STEP BEARING, AND COLLAR BEARING

Footstep bearing: Footstep bearings are used to support the lower end of the vertical shafts. A simple form of such bearing is shown in fig 1.4. It consists of cast iron block into which a gunmetal bush is fitted. The bush is prevented from rotating by the snug provided at its neck. The shaft rests on a concave hardened steel disc. This disc is prevented from rotating along with the shaft by means of pin provided at the bottom.

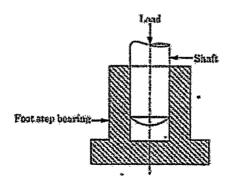


Fig:1.4 Foot step Bearing

Collar bearing:

The simple type of thrust bearing for horizontal shafts consists of one or more collars cut integral with the shaft as shown in fig.1.5. These collars engage with corresponding bearing surfaces in the thrust block. This type of bearing is used if the load would be too great for a step bearing, or if a thrust must be taken at some distance from the end of the shaft. Such bearings may be oiled by reservoirs at the top of the bearings.

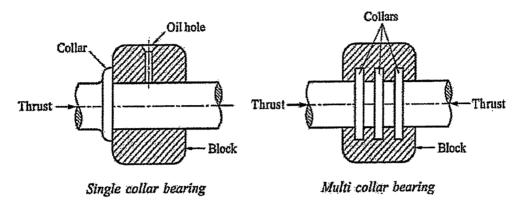


Fig.1.5 Collar bearings

Thrust bearings of fixed inclination pad and pivoted pad variety are shown in figure 1.6 a & b. These are used for carrying axial loads as shown in the diagram. These bearings operate on hydrodynamic principle.

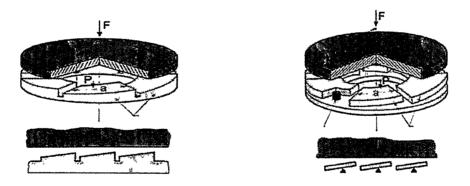


Fig. 1.6a Fixed-incline-pads thrust bearing

Fig.1.6b Pivoted-pads thrust bearing

Rolling contact bearings:

The bearings in which the rolling elements are included are referred to as rolling contact bearings. Since the rolling friction is very less compared to the sliding friction, such bearings are known as anti friction bearings.

Ball bearings:

It consists of an inner ring which is mounted on the shaft and an outer ring which is carried by the housing. The inner ring is grooved on the outer surface called inner race and the outer ring is grooved on its inner surface called outer race. In between the inner and outer race there are number of steel balls. A cage pressed steel completes the assembly and provides the means of equally spacing and holding the balls in place as shown in the figure 1.7. Radial ball bearings are used to carry mainly radial loads, but they can also carry axial loads.

Cylindrical roller bearings

The simplest form of a cylindrical roller bearing is shown in fig 1.8. It consists of an inner race, an outer race, and set of roller with a retainer. Due to the line contact between the roller and the raceways, the roller bearing can carry heavy radial loads.

Tapered roller bearings:

In tapered roller bearings shown in the fig. 1.9, the rollers and the races are all truncated cones having a common apex on the shaft centre to assure true rolling contact. The tapered roller bearing can carry heavy radial and axial loads. Such bearings are mounted in pairs so that the two bearings are opposing each other's thrust.

1.2.2 ADVANTAGES OF SLIDING CONTACT BEARINGS:

- They can be operated at high speeds.
- They can carry heavy radial loads.
- They have the ability to withstand shock and vibration loads.
- Noiseless operation.

Disadvantages:

- High friction losses during staring.
- More length of the bearing.
- · Excessive consumption of the lubricant and high maintenance.

1.2.3 ADVANTAGES ROLLING CONTACT BEARINGS:

- Low starting and less running friction.
- It can carry both radial as well as thrust loads.
- Momentary over loads can be carried without failure.
- Shaft alignment is more accurate than in the sliding bearings.

Disadvantages:

More noisy at high speeds.

Low resistance to shock loads.

High initial cost.

Finite life due to eventual failure by fatigue

1.3 SOLID FRICTION

- 1. Resistance force for sliding
 - Static coefficient of friction
 - Kinetic coefficient of friction
- 2. Causes
 - Surface roughness (asperities)
 - Adhesion (bonding between dissimilar materials)
- 3. Factors influencing friction
 - Sliding friction depends on the normal force and frictional coefficient, independent of the sliding speed and contact area
- 4. Effect of Friction
 - Frictional heat (burns out the bearings)
 - Wear (loss of material due to cutting action of opposing motion)
- 5. Engineers control friction
 - Increase friction when needed (using rougher surfaces)
 - Reduce friction when not needed (lubrication)

The coefficients of friction for different material combinations under different conditions are given in table 1.1.

TABLE 1.1
COEFFICIENTS OF FRICTION

OCCITION TO CITICION	
Material	μ
Perfectly clean metals in	Seizure µ > 5
vacuum	•
Clean metals in air	0,8-2
Clean metals in wet air	0.5-1.5
Steel on dry bearing metals	0.1-0.5
(e.g. fead, bronze)	
Steel on ceramics	0.1-0.5
Ceramics on ceramics	0.05-0.5
(e.g. carbides on carbides)	
Polymers on polymers	0.05-1.0
Metals and ceramics on	0.04-0.5
polymers	
(PE, PTFE, PVC)	
Boundary lubrication of	0.05-0.2
metals	
High-temperature lubricants	0.05-0.2
(MoS ₂ , graphite)	
Hydrodynamic lubrication	0.001-0.005

1.4 LUBRICATION:

Prevention of metal to metal contact by means of an intervening layer of fluid or fluid like material.

Types of sliding lubrication:

- · Sliding with Fluid film lubrication.
- · Sliding with Boundary lubrication.
- Sliding with Extreme boundary lubrication.
- Sliding with clean surfaces.

1.4.1 HYDRODYNAMIC / THICK FILM LUBRICATION / FLUID FILM LUBRICATION

Metal to Metal contact is prevented. This is shown in figure 1.10. Friction in the bearing is due to oil film friction only. Viscosity of the lubricant plays a vital role in the power loss, temperature rise & flow through of the lubricant through the bearing. The principle operation is the Hydrodynamic theory. This lubrication can exist under moderately loaded bearings running at sufficiently high speeds.



Fig. 1.10 Thick Film Lubrication

1.4.2 BOUDARY LUBRICATION (THIN FILM LUBRICATION)

During starting and stopping, when the velocity is too low, the oil film is not capable of supporting the load. There will be metal to metal contact at some spots as shown in figure 1.11. Boundary lubrication exists also in a bearing if the load becomes too high or if the viscosity of the lubricant is too low. Mechanical and chemical properties of the bearing surfaces and the lubricants play a vital role.



Fig.1.11 Boundary Lubrication

Oiliness of lubricant becomes an important property in boundary lubrication. Anti oxidants and Anti-corrosives are added to lubricants to improve their performance. Additives are added to improve the viscosity index of the lubricants.

Oiliness Agents

- Increase the oil film's resistance to rupture, usually made from oils of animals or vegetables.
- The molecules of these oiliness agents have strong affinity for petroleum oil and for metal surfaces that are not easily dislodged.
- Oiliness and lubricity (another term for oiliness), not related to viscosity, manifest itself under boundary lubrication; reduce friction by preventing the oil film breakdown.

Anti-Wear Agents

Mild EP additives protect against wear under moderate loads for boundary lubrications Anti-wear agents react chemically with the metal to form a protective coating that reduces friction, also called as anti-scuff additives.

1.4.3 Extreme boundary lubrication

Under certain conditions of temperature and load, the boundary film breaks leading to direct metal to metal contact as shown in figure 1.12. Seizure of the metallic surfaces and destruction of one or both surfaces begins. Strong intermolecular forces at the point of contact results in tearing of metallic particles. "Plowing" of softer surfaces by surface irregularities of the harder surfaces. Bearing material properties become significant. Proper bearing materials should be selected.

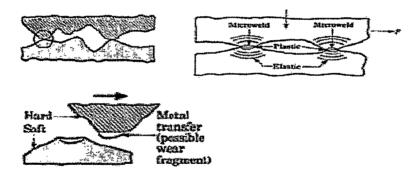


Fig.1.12 Extreme Boundary Lubrication

Extreme-Pressure Agents

Scoring and pitting of metal surfaces might occur as a result of this case, seizure is the primarily concern. Additives are derivatives of sulphur, phosphorous, or chlorine. These additives prevent the welding of mating surfaces under extreme loads and temperatures.

Stick-Slip Lubrication

1

A special case of boundary lubrication when a slow or reciprocating action exists. This action is destructive to the full fluid film. Additives are added to prevent this phenomenon causing more drag force when the part is in motion relative to static friction. This prevents jumping ahead phenomenon.

1.4.4 Solid film lubrication

When bearings must be operated at extreme temperatures, a solid film lubricant such as graphite or molybdenum di-sulphide must be used because the ordinary mineral oils are not satisfactory at elevated temperatures. Much research is currently being carried out in an effort to find composite bearing materials with low wear rates as well as small frictional coefficients.

1.4.5. Hydrostatic lubrication

Hydrostatic lubrication is obtained by introducing the lubricant, which is sometimes air or water, into the load-bearing area at a pressure high enough to separate the surfaces with a relatively thick film of lubricant. So, unlike hydrodynamic lubrication, this kind of lubrication does not require motion of one surface relative to another. Useful in designing bearings where the velocities are small or zero and where the frictional resistance is to be an absolute minimum.

1.4.6 Elasto Hydrodynamic lubrication

Elasto-hydrodynamic *lubrication* is the phenomenon that occurs when a lubricant is introduced between surfaces that are in rolling contact, such as mating gears or rolling bearings. The mathematical explanation requires the Hertzian theory of contact stress and fluid mechanics.

1.5 Newton's Law of Viscous Flow

In Fig. 1.13 let a plate A be moving with a velocity U on a film of lubricant of thickness h. Imagine the film to be composed of a series of horizontal layers and the force F causing these layers to deform or slide on one another just like a deck of cards. The layers in contact with the moving plate are assumed to have a velocity U; those in contact with the stationary surface are assumed to have a zero velocity. Intermediate layers have velocities that depend upon their distances y from the stationary surface.

Newton's viscous effect states that the shear stress in the fluid is proportional to the rate of change of velocity with respect to y.

Thus

T = F/A = Z (du/dy).

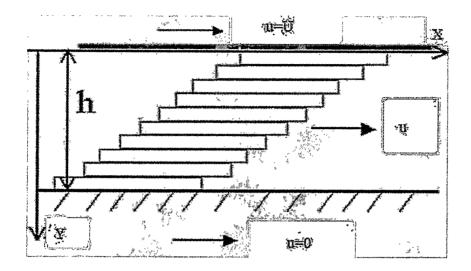


Fig.1.13 Viscous flow

where Z is the constant of proportionality and defines absolute viscosity, also called dynamic viscosity. The derivative du/dy is the rate of change of velocity with distance and may be called the rate of shear, or the velocity gradient. The viscosity Z is thus a measure of the internal frictional resistance of the fluid.

For most lubricating fluids, the rate of shear is constant, and du/dy = U/h. Fluids exhibiting this characteristic are known as a Newtonian fluids.

Therefore $\tau = F/A = Z(U/h)$.

The absolute viscosity is measured by the pascal-second (Pa \cdot s) in SI; this is the same as a Newton-second per square meter.

The poise is the cgs unit of dynamic or absolute viscosity, and its unit is the dyne second per square centimeter ($dyn \cdot s/cm2$). It has been customary to use the centipoises (cP) in analysis, because its value is more convenient. The conversion from cgs units to SI units is as follows:

$$Z(Pa \cdot s) = (10)^{-3} Z(cP)$$

Kinematic Viscosity is the ratio of the absolute Viscosity to the density of the lubricant.

$$Z_k = Z/\rho$$

The ASTM standard method for determining viscosity uses an instrument called the Saybolt Universal Viscosimeter. The method consists of measuring the time in seconds for 60 mL of lubricant at a specified temperature to run through a tube 17.6 micron in diameter and 12.25 mm long. The result is called the *kinematic viscosity*, and in the past

the unit of the square centimeter per second has been used. One square centimetre per second is defined as a **stoke**.

The kinematic viscosity based upon seconds Saybolt, also called *Saybolt Universal viscosity* (SUV) in seconds, is given by:

$$Zk = (0.22t - 180/t)$$

where Z_k is in centistokes (cSt) and t is the number of seconds Saybolt.

1.6 Viscosity -Temperature relation

Viscous resistance of lubricating oil is due to intermolecular forces. As the temperature increases, the oil expands and the molecules move further apart decreasing the intermolecular forces. Therefore the viscosity of the lubricating oil decreases with temperature as shown in the figure.1.14. If speed increases, the oil's temperature increases and viscosity drops, thus making it better suited for the new condition. An oil with high viscosity creates higher temperature and this in turn reduces viscosity. This, however, generates an equilibrium condition that is not optimum. Thus, selection of the correct viscosity oil for the bearings is essential.

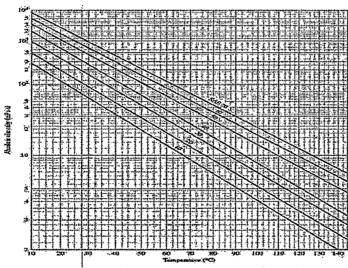


Fig. 1.14 Viscosity temperature relationship

Viscosity index of a lubricating oil

Viscosity Index (V.I) is value representing the degree for which the oil viscosity changes with temperature. If this variation is small with temperature, the oil is said to have a high viscosity index. The oil is compared with two standard oils, one having a V.I. of 100 and the other Zero. A viscosity index of 90 indicates that the oil with this value thins out less rapidly than an oil with V.I. of 50.

1.7 Types of lubricants

- Vegetable or Animal oils like Castor oil, Rapeseed oil, palm oil, Olive oil etc.
- Animal oils like lard oil, tallow oil, whale oil, etc.
- Mineral oils-petroleum based-Paraffinic and Naphthenic based oils

Properties of lubricants

- Availability in wide range of viscosities.
- · High Viscosity index.
- Should be Chemically stable with bearing material at all temperatures encountered.
- Oil should have sufficient specific heat to carry away heat without abnormal rise in temperature.
- Reasonable cost.

Selection Guide for Lubricants

The viscosity of lubricating oil is decisively for the right thickness of the lubricating film (approx. 3-30µm) under consideration of the type of lubricant supply

Low sliding speed High Viscosity

High sliding speed Low viscosity

High bearing clearance High Viscosity

High load (Bearing pressures) Higher Viscosity

1.8 Bearing materials

Relative softness (to absorb foreign particles), reasonable strength, machinability (to maintain tolerances), lubricity, temperature and corrosion resistance, and in some cases, porosity (to absorb lubricant) are some of the important properties for a bearing material.

A bearing element should be *less than one-third as hard* as the material running against it in order to provide **embedability** of abrasive particles.

A bearing material should have high compression strength to withstand high pressures without distortion and should have good fatigue strength to avoid failure due to pitting. e.g. in Connecting rod bearings, Crank shaft bearings, etc. A bearing material should have conformability. Soft bearing material has *conformability*. Slight misalignments of bearings can be self-correcting if plastic flow occurs easily in the bearing metal. Clearly there is a compromise between load-bearing ability and conformability.

In bearings operating at high temperatures, possibility of oxidation of lubricating oils leading to formation of corrosive acids is there. The bearing material should be corrosion resistant. Bearing material should have easy availability and low cost. The bearing material

should be soft to allow the dirt particles to get embedded in the bearing lining and avoid further trouble. This property is known as **Embeddability.**

Different Bearing Materials

- Babbitt or White metal usually used as a lining of about 0.5mm thick bonded to bronze, steel or cast iron.
 - Lead based & Tin based Babbitt's are available.
 - · Excellent conformability and embeddability
 - · Good corrosion resistance.
 - Poor fatigue strength
- Copper Based alloys most common alloys are copper tin, copper lead, phosphor bronze: harder and stronger than white metal: can be used <u>un-backed as a solid</u> <u>bearing</u>.
- Aluminum based alloys running properties not as good as copper based alloys but cheaper.
 - Ptfe suitable in very light applications
 - Sintered bronze Sintered bronze is a porous material which can be impregnated with oil, graphite or Ptfe. Not suitable for heavily loaded applications but useful where lubrication is inconvenient.
 - Nylon similar to Ptfe but slightly harder; used only in very light applications.

Triple-layer composite bearing material consists of 3 bonded layers: steel backing, sintered porous tin bronze interlayer and anti-wear surface as shown in figure 1.15. High load capacities and low friction rates, and are oil free and anti-wear.

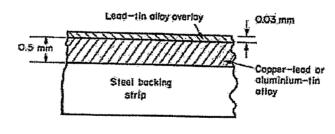


Fig.1.15 Tri-metal Bearing

If oil supply fails, frictional heating will rapidly increase the bearing temperature, normally lead to metal-to-metal contact and eventual seizure. Soft bearing material (low melting point) will be able to shear and may also melt locally. **Protects the journal** from severe surface damage, and helps to avoid component breakages (sudden locking of mating surfaces).

1.9 Petroff's Equation for lightly Loaded Bearings

The phenomenon of bearing friction was first explained by Petroff on the assumption that the shaft is concentric. This can happen when the radial load acting on the bearing is zero or very small, speed of the journal is very high and the viscosity of the lubricant is very high. Under these conditions, the eccentricity of the bearing (the offset between journal center and bearing center) is very small and the bearing could be treated as a concentric bearing as shown in figure 1.16

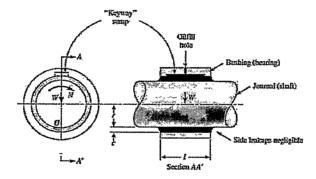


Fig.1.16 Concentric Bearing

Let us now consider a shaft rotating in a guide bearing. It is assumed that the bearing carries a very small load, that the clearance space is completely filled with oil, and that leakage is negligible (Fig. 7.16). Let the radius of the shaft be r, and the length of the bearing by l. If the shaft rotates at N' rev/s, then its surface velocity is $U = 2\pi r$ N'. Since the shearing stress in the lubricant is equal to the velocity gradient times the viscosity,

$$\tau = Z U/h = 2\pi r N' Z/c$$

where the radial clearance c has been substituted for the distance h.

F= Frictional force= τ A= $(2\pi rN'Z/c)(2\pi rI)$ = $(4\pi 2r^2|ZN'/c)$

Frictional torque= $Fr = (4\pi 2r^3 IZN'/c)$

The coefficient of friction in a bearing is the ratio of the frictional force F to the Radial load W on the bearing. $f = F/W = (4\pi 2r^3 IZN'/cW)$

The unit bearing pressure in a bearing is given by p= W/2rL= Load/ Projected Area of the Bearing.

Or W= 2prL

Substituting this in equation for f and simplifying

$$f = 2\pi^2 (ZN'/p) (r/c)$$

This is the Petroff's equation for the coefficient of Friction in Lightly Loaded bearings.

Example on lightly loaded bearings

E1. A full journal bearing has the following specifications:

- Journal Diameter:46 mm
- · Bearing length: 66 mm
- Radial clearance to radius ratio: 0.0015
- speed: 2800 r/min
- Radial load: 820 N.
- Viscosity of the lubricant at the operating temperature:8.4 cP

Considering the bearing as a lightly loaded bearing, Determine (a) the friction torque (b) Coefficient of friction under given operating conditions and (c) power loss in the bearing.

Solution:

Since the bearing is assumed to be a lightly loaded bearing, Petroff's equation for the coefficient of friction can be used.

$$f = 2\pi^2 (ZN'/p) (r/c)$$

N = 2800/60 = 46.66 r/sec.

$$Z = 8.4 \text{ cP} = 8.4 \times 10^{-3} \text{ Pa.sec}$$

$$r = 46/2 = 23 \text{ mm} = 0.023 \text{ m}$$

P=w/2rL=820/2X0.023X0.066=270092 Pa.

Substituting all these values in the equation for f, f = 0.019

T=Frictional torque: Frictional force x Radius of the Journal

- = (fW) r
- $= 0.019 \times 820 \times 0.023$
- = 0.358 N-m
 - = 0.358 x 46.66/ 1000
 - = 0.016 kW

1.10 HYDRODYNMIC JOURNAL BEARINGS Concept

The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing.

One type occurs when the rate of shear across the oil film is a constant value and the line representing the velocity distribution is a straight line. In the other type the velocity distribution is represented by a curved line, so that the rate of shear in different layers across the oil film is different. The first type takes place in the case of two parallel surfaces having a relative motion parallel to each other as shown in Fig.1.19.

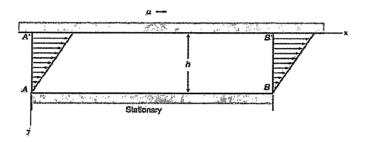


Fig. 1.19 Velocity profiles in a parallel-surface slider bearing.

There is no pressure development in this film. This film cannot support an external Load. The second type of velocity distribution across the oil film occurs if pressure exists in the film. This pressure may be developed because of the change of volume between the surfaces so that a lubricant is squeezed out from between the surfaces and the viscous resistance of flow builds up the pressure in the film as shown in Fig 1.20 or the pressure may be developed by other means that do not depend upon the motion of the surfaces or it may develop due to the combination of factors. What is important to note here is the fact that pressure in the oil film is always present if the velocity distribution across the oil film is represented by a curved line

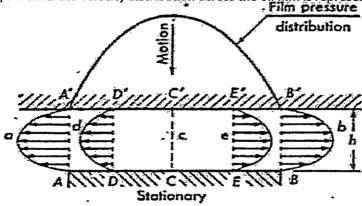


Fig.1.20 Flow between two parallel surface

Plate AB is stationary while A' B' is moving perpendicular to AB.

Note that the velocity distribution is *Curvilinear*. This is a *pressure induced flow*.

This film can support an External load.

Hydrodynamic film formation

Consider now the case of two non parallel planes in which one is stationary while the other is in motion with a constant velocity in the direction shown in Fig 1.21. Now consider the flow of lubricant through the rectangular areas in section AA' and BB' having a width equal to unity in a direction perpendicular to the paper.

The volume of the lubricant that the surface A'B' tends to carry into the space between the surfaces AB and A'B' through section AA' during unit time is AC'A'. The volume of the lubricant that this surface tends to discharge from space through section BB' during the same period of time is BD'B'. Because the distance AA' is greater than BB' the volume AC'A' is greater than volume BC'B' by a volume AEC'. Assuming that the fluid is incompressible and that there is no flow in the direction perpendicular to the motion, the actual volume of oil carried into the space must be equal to the discharge from this space. Therefore the excess volume of oil is carried into these space is squeezed out through the section AA' and BB' producing a constant pressure — induced flow through these sections.

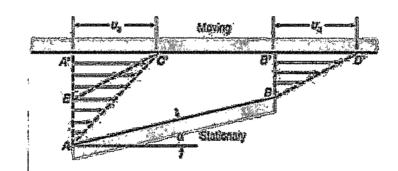


Fig. 1.21 Velocity distribution only due to moving plate

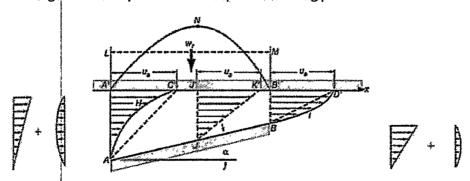


Fig.1.22 Resultant Velocity Distribution

The actual velocity distribution in section AA' and BB' is the result of the combined flow of lubricant due to viscous drag and due to pressure –induced flow. The resultant velocity distributions across these sections are as shown in Fig 1.22.

The curve A'NB' shows the general character of the pressure distribution in the oil film and the line LM shows the mean pressure in the oil film. Because of the pressure developed in the oil film the, plane A'B' is able to support the vertical load W applied to this plane, preventing metal to metal contact between the surfaces AB and A'B'. This load is equal to the product of projected area of the surface AB and mean pressure in the oil film.

Conditions to form hydrodynamic lubrication

There must be a wedge-shaped space between two relative moving plates;

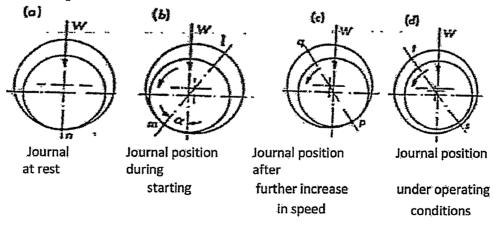
There must be a relative sliding velocity between two plates, and the lubricant must flow from big entrance to small exit in the direction of the moving plate;

The lubricant should have sufficient viscosity, and the supply of the lubricant is abundant.

Formation of oil film in a Journal bearing

Imagine a journal bearing with a downward load on the shaft that is initially at rest and then brought up to operating speed. At rest (or at slow shaft speeds), the journal will contact the lower face of the bearing as shown in the figure 1.23. This condition is known as boundary lubrication and considerable wear can occur. As shaft speed increases, oil dragged around by the shaft penetrates the gap between the shaft and the bearing so that the shaft begins to "float" on a film of oil. This is the transition region and is known as thin-film lubrication. The journal may occasionally contact the bearing particularly when shock radial load occur. Moderate wear may occur at these times. At high speed, the oil film thickness increases until there comes a point where the journal does not contact the bearing at all. This is known as thick film lubrication and no wear occurs because there is no contact between the journal and the bearing.

The various stages of formation of a hydrodynamic film is shown in figure 1.23.



Pressure distribution around an idealised journal bearing

A typical pressure distribution around the journal in a hydrodynamic bearing is as shown in the Fig. 1.24.

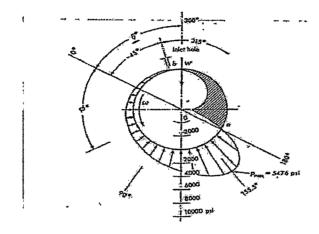
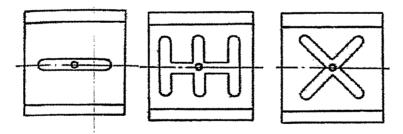


Fig.1.24 Bearing pressure distribution in a journal bearing

Typical oil groove patterns



Some typical groove patterns are shown in the above figure. In general, the lubricant may be brought in from the end of the bushing, through the shaft, or through the bushing. The flow may be intermittent or continuous. The preferred practice is to bring the oil in at the center of the bushing so that it will flow out both ends, thus increasing the flow and cooling action.

1.13 Thermal aspects of bearing design

Heat is generated in the bearing due to the viscosity of the oil. The frictional heat is converted into heat, which increases the temperature of the lubricant. Some of the lubricant that enters

the bearing emerges as a side flow, which carries away some of the heat. The balance of the lubricant flows through the load-bearing zone and carries away the balance of the heat generated. In determining the viscosity to be used we shall employ a temperature that is the average of the inlet and outlet temperatures, or

$$Tav=(Ti+T)/2$$

where= $(T_1 + T_2)$ is the inlet temperature and T is the temperature rise of the lubricant from inlet to outlet. The viscosity used in the analysis must correspond to T_{av} .

Self contained bearings:

These bearings are called *selfcontained* bearings because the lubricant sump is within the bearing housing and the lubricant is cooled within the housing. These bearings are described as *pillow-block* or *pedestal* bearings. They find use on fans, blowers, pumps, and motors, for example. Integral to design considerations for these bearings is dissipating heat from the bearing housing to the surroundings at the same rate that enthalpy is being generated within the fluid film.

Heat dissipated based on the projected area of the bearing:

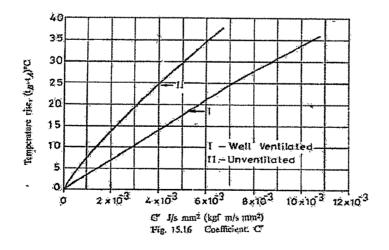
Heat dissipated from the bearing, J/S $H_D = CA (t_B-t_A)$

Where C= Heat dissipation coefficient from data hand book

Another formula to determine the heat dissipated from the bearing $H_D = Id (T+18)^2 / K_3$

Where $K_3 = 0.2674 \times 10^6$ for bearings of heavy construction and well ventilated = 0.4743×10^6 for bearings of light construction in still air air

T= t_B - t_A
Where,
t_B = Bearing surface temperature
t_A = Ambient temperature



For good performance thye following factors should be considered.

Surface finish of the shaft (journal): This should be a fine ground finish and preferably lapped.

Surface hardness of the shaft: It is recommended that the shaft be made of steel containing at least 0.35-0.45% carbon. For heavy duty applications shaft should be hardened.

Grade of the lubricant: In general, the higher the viscosity of the lubricant the longer the life. However the higher the viscosity the greater the friction, so high viscosity lubricants should only be used with high loads. In high load applications, bearing life may be extended by cutting a grease groove into the bearing so grease can be pumped in to the groove.

Heat dissipation: Friction generates heat and causes rise in temperature of the bearing and lubricant. In turn, this causes a reduction in the viscosity of the lubricating oil and could result in higher wear. Therefore the housing should be designed with heat dissipation in mind. For example, a bearing mounted in a Bakelite housing will not dissipate heat as readily as one mounted in an aluminium housing.

Shock loads: Because of their oil-cushioned operation, sliding bearings are capable of operating successfully under conditions of moderate radial shock loads. However excessive prolonged radial shock loads are likely to increase metal to metal contact and reduce bearing life. Large out of balance forces in rotating members will also reduce bearing life.

Clearance: The bearings are usually a light press fit in the housing. A shouldered tool is usually used in arbour press. There should be a running clearance between the journal and the bush. A general rule of thumb is to use a clearance of 1/1000 of the diameter of the journal.

Length to diameter ratio(I/d ratio): A good rule of thumb is that the ratio should lie in the range 0.5-1.5. If the ratio is too small, the bearing pressure will be too high and it will be difficult to retain subricant and to prevent side leakage. If the ratio is too high, the friction will be high and the assembly misalignment could cause metal to metal contact.

Examples on journal bearing design

Example EI:

Following data are given for a 360° hydrodynamic bearing:

Radial load=3.2 kN
Journal speed= 1490 r.p.m.
Journal diameter=50 mm
Bearing length=50mm
Radial clearance=0.05 mm
Viscosity of the lubricant= 25 cP

Assuming that the total heat generated in the bearing is carried by the total oil flow in the bearing, calculate:

- Power lost in friction:
- The coefficient of friction;
- Minimum oil film thickness
- Flow requirement in 1/min; and
- Temperature rise.

Solution:

P= W/Ld = 3.2x1000/ (50x50) = 1.28 MPa.= $1.28x10^6$ Pa Sommerfeld number = $S = (ZN'/p) (r/c)^2$

r/c = 25/0.0.05 = 500Z= 25 cP = 25x10⁻³ Pa.sec = 1490/60= 24.833 r/sec. Substituting the above values, we get **S=0.121**

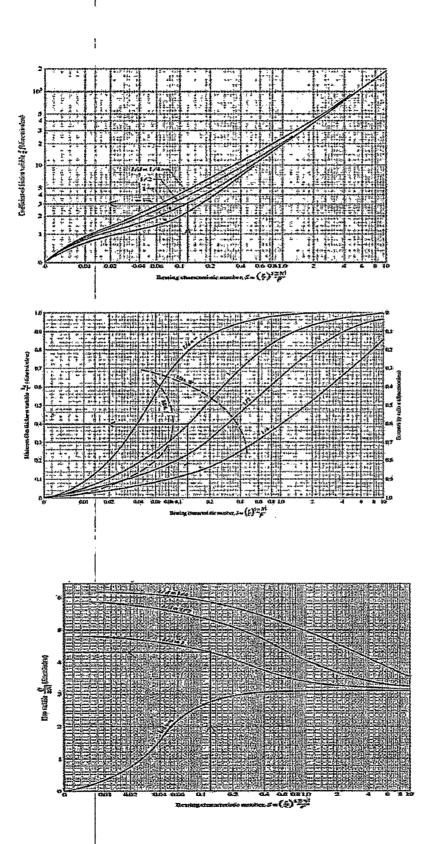
For S= 0.121 & L/d=1, Friction variable from the graph= (r/c) f= 3.22 Minimum film thickness variable= h_o /c =0.4 Flow variable= Q/rcN L= 4.33

f = 3.22x0.05/25= 0.0064 Frictional torque= T= fWr = 0.0064x3200x 0.025 = 0.512 N-m Power loss in the Bearing= 2π N T/ 1000 kW = 0.080 kW

 $h_o = 0.4 \times 0.05 = 0.02 \text{ mm}$ $Q/r c N^I L = 4.33 \text{ from which we}$ get, $Q = 6720.5 \text{ mm}^3 / \text{sec.}$

Ex 2

Determination of dimensionless variables is shown in the following figures. Assume that all the heat generated due to friction is carried away by the lubricating oil. Heat generated = 80 watt = mC_pT where: m= mass flow rate of lubricating oil= pQ in kg/sec Cp= Specific heat of the oil= 1760 J/kg ^{o}C T= temperature rise of the oil p= 860x10 $^{-9}$ kg/mm 3 Substituting the above values, T= 7.9 ^{o}C The Average temperature of the oil= $T_1+T/2=27+(7.9/2)=30.85$ ^{o}C



Example E2:

A 50 mm diameter hardened and ground steel journal rotates at 1440 r/min in a lathe turned bronze bushing which is 50 mm long. For hydrodynamic lubrication, the minimum oil film thickness should be five times the sum of surface roughness of journal bearing. The data about machining methods are given below:

Machining method

surface Roughness(c.l.a)

Shaft

grinding

turning/boring

1.6 micron

0.8 micron

Bearing

The class of fit is H8d8 and the viscosity of the lubricant is 18 cP. Determine the maximum radial load that the journal can carry and still operate under hydrodynamic conditions.

Solution:

Min. film thickness = h_0 = 5 [0.8+1.6] = 12 micron = 0.012

mm For H8 d8 fit, referring to table of tolerances,

Ø50 H8 = Min. hole limit = 50.000 mm

Max. hole limit = 50.039 mm

Mean hole diameter= 50.0195 mm

 \emptyset 50 d8 = Max. shaft size = 50- 0.080= 49.920

mm Min. shaft size = 50-0.119= 49.881 mm

Mean shaft diameter= 49.9005 mm.

Assuming that the process tolerance is centered,

Diametral clearence= 50.0195- 49.9005= 0.119 mm

Radial clearence= 0.119/2= 0.0595 mm

 $h_0/c = 0.012/0.0595 = 0.2$

L/d = 50/50 = 1

From the graph, Sommerfeld number= 0.045

 $S=(ZN'/p)(r/c)^2=0.045$

r/c= 25/0.0595= 420.19

Z= 18 cP= 18x10⁻³ Pa.sec

N'= 1440/60= 24 r/sec

From the above equation, Bearing pressure can be calculated.

 $p=1.71x10^6$ Pa = 1.71 MPa.

The load that the bearing can carry:

W= pLd = 1.71x 50x 50= 4275 N

Example E3:

The following data are given for a full hydrodynamic journal bearing:

Radial load=25kN

Journal speed=900 r/min.

Unit bearing pressure= 2.5 MPa

(I/d) ratio= 1:1

Viscosity of the lubricant=20cP

Class of fit=H7e7

Calculate: 1.Dimensions of bearing

2. Minimum film thickness and

3. Requirement of oil flow

Solution:

N = 900/60 = 15 r/sec

P=W/Ld

2.5= 25000/Ld=25000/d²

d= 100 mm & L=100 mm

For H7 e7 fit, referring to table of tolerances,

Ø100 H7 = Min. hole limit = 100.000 mm

Max. hole limit = 100.035 mm

Mean hole diameter= 100.0175 mm

Ø 100 e7 = Max. shaft size = 100-0.072= 99.928

mm Min. shaft size = 100-0.107=

99.893 mm

Mean shaft diameter= 99,9105 mm

Assuming that the process tolerance is centered, Diametral clearence= 100-0175- 99.9105= 0.107 mm

Radial clearence= 0.107/2= 0.0525mm

Assume r/c = 1000 for general bearing applications.

C=r/1000=50/1000=0.05 mm. Z=20 cP= $20x10^{-3}$ Pa.sec

N' = 15 r/sec P= 2.5 MPa= 2.5 x10⁶ Pa S= (ZN'/p) (r/c)² =0.12

For L/d=1 & S=0.12, Minimum Film thickness variable= h_o /c = 0.4

 $h_0 = 0.4 \times 0.05 = 0.02 \text{ mm}$

Example E4:

A journal bearing has to support a load of 6000N at a speed of 450 r/min. The diameter of the journal is 100 mm and the length is 150mm. The temperature of the bearing surface is limited to 50 °C and the ambient temperature is 32 °C. Select a suitable oil to suit the above conditions.

Solution:

```
N^{I} = 450/60 =7.5 r/sec, W=6000 N, L=150mm, d=100 mm, t_{A} = 32 ^{\circ}C, t_{B} = 50 ^{\circ}C. Assume that all the heat generated is dissipated by the bearing.
```

Use the Mckee's Equation for the determination of coefficient of friction.

f=Coefficient of friction= K_a (ZN^I/p) (r/c) 10^{-10} +f

```
p= W/Ld= 6000/100x150 = 0.4 MPa. K_a = 0.195x \ 10^6 for a full bearing f = 0.002 r/c= 1000 assumed U = 2\pi rN^1 = 2x3.14x \ 50x7.5 = 2335 mm/sec= 2.335 m/sec. f = 0.195x \ 10^6 x (Z * 7.5 / 0.4) x 1000 \ x \ 10^{-10} +0.002 f = 0.365z + 0.002
```

Heat generated= f *W*U Heat generated= (0.365Z+ 0.002)x6000x2.335

Heat dissipated from a bearing surface is given by:

```
H_{\rm D} = Id (T+18)^2 / K_3
```

Where K_3 = 0.2674x 10^6 for bearings of heavy construction and well ventilated = 0.4743x 10^6 for bearings of light construction in still air air

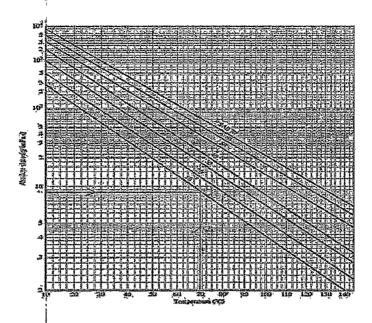
```
T = t_B - t_A = 50-32 = 18 C

H_D = 150 \times 100 (18+18)^2 / 0.2674 \times 10^6 = 72.7 Watt
```

 H_D = Hg for a self contained bearing.

72.7 = (0.365Z+ 0.002)x6000x2.335 Z= 0.0087 Pa.Sec.

Relation between oil temp, Amb. temp, & Bearing surface temperature is given by $t_B-t_A=\%$ (t_0-t_A) t_0 = oil temperature= 68 $^\circ$ C



Select SAE 10 Oil for this application

[Journal bearing] It is required to design a journal bearing for a centrifigal pump running at 1200 P.P.M. The journal is 8cm and load on each journal is 1800 kg. The factor $\frac{2N}{p}$ for bearing is 3600. The length and diameter vatio is 1.9. Take correction factor as 0.002. The bearing and atmospheric temperatures are 80°C and 30°C respective a) To calculate bearing pressure: d = Diameter of journal = 8cm. L= Longth of journal, = 1.2. >> L= 1.2d = 1.2 ×8 = 9.6cm P= Load on bearing = 1800kg: -A=Projected area = Lxd = 9.6 x8 = 76.8 cm2. Pa Bearing Pressure. P=pxA => P= A = 1800 = 93.4 kg/cm2. b, To find Viscosity: -N= R.P.M of journal = 1200 ZN = 3600 (given) >) . Z = D x 3600 = 33.4 x 3600 Z = 50.7 centipolise. 1 centipoise= 100 poise E) To calculate coefficient of fiction: u= coefficient of Riction M= 33 (2N) (d) +k - where = = clearance votio = 0.001 (say) K = Correction factor = 0.002 (given) $M = \frac{33}{100} \times 2600 \times \frac{1}{0.001} + 0.002 = 0.01058$ اعداد، و ا) ते) अवृत्य

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d, To calculate heat generated and heat dissipated: H= Heat generated H= 4.P.V kcal/min. и = 0·0 lo58 P=Load on bearing = 1800 kg. V= 100 = Tx 100 x 1200 = 301.44m/min-:. H = 0.01058 × 1800 × 301.44 = 13.44 kcal /min. Hd = Heat dissipated Hd= C, Ld. (T2-T1) C, = Energy dissipation coefficient = 0.0012 (assume). 72 = Bearing temperature = 80°C T, = Atmospheric temperature = 30°c. :. Hd = 0.0012 ×9.6 × 8 (80-30) = 4.6 kcal /min -As the rate of heat dissipation is less than rate of heat generation, therefore the bearing should be coded artificially. -) A journal bearing of somm diameter and somm long has a bearing pressure of 60/mm2, the speed of the journal is 100 orpm. The ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil, whose absolute viscosity at the operat temperature of 75°c may be taken as 0.015/cg/m-s. The room temperature is osc, Determine i, The amount of artificial cooling required. in. The mass of coolant oil required, if he différence between the

outlet and inlet temperature of oil is 10°C. The specific heat

```
of the oil is 1900 I /1eg/oc and heat dissipation coefficient is
500 W/ 2/0c.
- G.D: - Diameter of journal, d = 50mm.
        length of joynal, L= 80mm.
         Beoring pressure, P=60 mm2;
        Speed N= 1000 rpm.
         Ratio, d/c = 1000
        Oil Temperature, To = 75°C
        Viscosity of oil, Z = 0.015 kg/m-s.
       Room temperature, Ta = 25°c.
Difference between outlet and inlet temperatures, T=10°C.
       Specific heat of oil, &= 19005/109/0c.
        Heat dissipation coefficient, C = 500 W/m2/oc.
       Ratio of length to bearing diameter (1) = 80 = 1.6.
 Coefficient of Riction M= 32 ZN Pk.
     = \frac{33}{10^8} \left| \frac{0.015 \times 1000}{6} \right| \left[ 1000 \right] + 0.002
                               [: k=0.002 for 1 = 0.75 to 28]
      = 0.000895+0.002 = 0.002825 > [H=0.002825]
 Load acting on bearing, W= Pxdxl=6x50x80= 24,0000.
 Heat generated Dig = MWV = MXWX TIdN
                        =0.002825 ×24,000 × 11×0.05×1000
                 : Qg = 177.55/g
Heat discipated Qu = C.A (Tb-Ta)
             But (T_b - T_a) = \frac{1}{2} (T_o - T_a) = \frac{1}{2} (75 - 25) = \frac{1}{2} (50)
                    Tb-Ta=25°C
```

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Qy = 500 × 0.05 × 0.08 × 25 = 501/6. :. Of = 202/8 1) Amount of Artificial Cooling Required: -Quartificial = Heat generated - Heat dissipated = Qg - Q1 : [Quartificial = 197.5] 2) Mass of Coolant oil Required; -Let in be the mass of the oil required to carry away exces heat generated over heat dissipated abortificial = mxsxT 127.5 = m × 1900 × 10 7 m = 0.007 kg/s. 1: m = 0.403 kg/min A journal bearing somm in diameter and 120mm long runs at 600pm. It uses oil having viscosity of 60 centipoises. Radial clearance provided is 0.15mm. Determine the sakload, booking can take. Assure Sommertell Number = 1.43 × 109. 1 pascal second G.D: - Journal diameter d= 80mm= 0.08m. = 1000 centi poigeion Length of bearing, L = 120mm = 01012m. = 1kg/m-s ... Speed, N= 600 .v.p.m. Viscosity of oil, Z=60 contipoise = 0.06/cg/in-s. Radial clearance, = 0.15mm. : c = 0.30mm

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"Sommerfield number = 1.43 × 10°. $S = \frac{20}{p} \left(\frac{d}{c}\right)^2 \text{ where } c = \text{diametral clearance}.$

ire = 1.43 x109.

 $\frac{0.06 \times 600}{P} \left(\frac{80}{0.3} \right)^{2} = 1.43 \times 10^{9}.$

" b=1.74×10-3 n/mg.

Newton-second = /cg-m/s. N-9 = leg-m/s. Nis/me = kg/mis

Sale load on the bearing W=P.A.

= 1.79 x120 x80

1 /W= 17.184N

A somm diameter 75mm long journal bearing is loaded with bearing pressure of 2 mpa. The shaft in bearing votates at soorp.m Heat is last from surface at the vate of 11.6 Joules In 2 per secpers. The housing area is 8 times the projected area. If the room temperature is 98°C, determine the surface temperature of bearing. Coefficient of Friction 15 0.0015.

G.D:- Diameter d= 50mm, Length L=75mm

Bearing pressure Pb=2mpor, Speed N=500 r.p.m

Heat lost, C= 11.6 J/m2/sec/°c.

Bearing temperature tb = 28°c

Coefficient of fiction 1=0.0015

Rodial Lord is given by W=pxlxd = 2x75x50=7500N.

Velocity, V= TDN = TT x 50 x 500 = 1.309 m/sec.

Heat generated is given by Dig = MWV = 0.0015 × 7500 × 1.309.

w=I/s=10-40/seq. ag = 14.726 I/s or Watts. Heat dissipated is given by Q1 = 8Cxlxdx (ts-tb) = 8x11.6x75x50x(ts-28) = 348000 (ts-28). Equating ag = al 14.726 = 34,8000 (ts-28) N-m = J/2/gec/or xmm (ts-28) = 14.726 × 1000 × 1000 ts-28=4222. Ls=4932+28=70-292c. : Girbace temperature of bearing, to = 70.293c. -> Design a journal booking for a centrifyed primp running at 1440pm. The diameter of the journal is 100mm and load on each bearing 19 Jokn - The Pactor 2N/p may be taking 28 for Centrifyed pump bearings. The bearing is running at 75°c temperature and the atmospheric temperature is 30°c. The energy dissipation coefficient is 875 W/m²/c. Take diametrial degrana as a limin. G.D: - Speed of journal N= 1440vpm. Dia of journel of = 100mmy = 0.1m. hood on the bearing, W= 20/cN= 20×103N. P= 28.1 Temperature of bearing to=15°C Temperature of atmosphere ta=30°c. Energy dissipation coefficient, C= 87.5 W/m/oc. Diameter deavance c= D-d=0.1min. Diameter clearance vetro, $\frac{c}{d} = \frac{D-d}{d} = \frac{0.1}{100} = 0.001$ Let I be the length of the journal, 1 for certifical pump (from dota book) = 1.5.

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. melio = monoel = 1 Larmol all to Agnod :

·mmos1 = 001x51 = P51 =7

```
Speed, N=1440r.p.m.
At mospheric temperature, Ta = 16°c.
                                                     certipoise= 1000 kg/m-s
Operating temperature of oil To= 60°C.
Absolute viscosity, Z = 23 centipoise = 0.023/cg/m-s.
 1/2 valio for Centrifugel pump is 1.6 (from data book)
            1=1.6x.d=1.6x75=120mm.
Bearing pressure P= 1/2000 = 1:332/min?.
Bearing pressure for centrifyed pump is 1.40/mm2, therefore the:
design is sale.
 Bearing characteristic number \frac{ZN}{p} = \frac{0.093 \times 14.40}{1.33} = 94.9
    ZN for centritugal pump is 28 (from data booktable)
Bearing modules at point of minimum friction is given as.
           \frac{ZN}{P} = 3\frac{1}{3}\left(\frac{2N}{P}\right) = \frac{1}{3}(28) = 9.33
Since the value of ZN = 24.9 is more than 9.33 the bearing is
said to be hydrodynamic:
Coefficient of friction u = \frac{33}{108} \left[ \frac{2N}{P} \right] \left[ \frac{d}{c} \right] + k.
                              = 108 (24.9) (0-0013) +0.002.
                                [: 42 Per certifiged pump is 0.0013].
                                [kfor 42=0.6 is taken as 0.00].
                     :. H= 0.0063+0.00 2 = 0.0083
                             1:M = 0:0083
Heat generated any = MWV
                       = 0.0083 × 12000 × TX × 0.0075 × 1440
```

Heat dissipation coefficient for well ventilated bearing is =1200 W/m/2

: Qd = 1200 x 0 : 012 x 0 : 075 x 22 = 237. 6 walts. Heat generated is more than heat dissipated so artifical cooling required Amount of artificial cooling required = Host generated - Host dissipated = Org - Ord = 563.9-237.6=395.6W = 325.6w = 0,325kw.

A journal bearing with a diameter of somm and length 150 mm Carries a load of solar, when the journal speed is 150 mp. m. The diametral clearance ratio is 0.0015. If possible, the bearing is to operate at 35°C. ambient temperature without external aboling with a maximum oil temperature of 90°C. It external cooling is required, it is to be as · little as possible to minimize the required oil flaw rate and heat oxchanger Size.

(a) what type of oil do you recommend?

b) Will be bearing operate without external cooling?

c) If the bearing operates without external cooling, determine the operating oil temperature?

d) If the bearing operates with external cooling, determine the amount of oil in Eglmin required to cong away the occas heat generated over heat dissipated, when the oil temperature vises from 85°c to 90°c, when passing through the bearing.

```
G.D:- Dia of jamal, d= 200mm, Length of jamal, L= 150mm."
   Load, w= 20/cn = 20×1030.
    Diametral clearance vation = 0:0015.
    Oil temperature, To = 90°C
   Ambient temperature, Ta= 35°c,
   Speed, N= 150 ripim.
 Assume that the bearing is used in marine steam engines, (cr) by I ratio.
   = = 0.0015, 1 = 150 = 0.75 / 1Z = 0.03 N-5/m2(a) kg/m-s 15
 a, From table and graph SAE 70 type of oil is recommended.
 by Bearing pressure, P = W = 20,000 = 0.667 N/min.
      Sommerfeld number = \frac{2N}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6 (For design purpose).
       \frac{2N}{p} = \frac{14.3 \times 10^6}{\left(\frac{d}{2}\right)^2} = 14.3 \times 10^6 \times (0.0015)^2 = 39.175.
  : Coefficient of fration, H = \frac{33}{108} \left[\frac{20}{P}\right] \left[\frac{1}{c}\right] + k
                                                   [:: k=0.002 for 41=0.75]
          \mu = \frac{3^{\frac{3}{2}}}{10^{\frac{3}{2}}} \times 32.175 \left[ \frac{1}{0.0015} \right] + 0.002 = 0.009 \Rightarrow \mu = 0.009
 Heat generated, ag = LINV
               = 0.009 × 20,000 × TX0.2 × 150
                                                               TIEN m/s
         : Dig = 282.743W.
Heat dissipated, Que C.A (Tb-Ta)
             But, Tb-Ta = = (To-Ta) = = (90-35) = 27.5°c.
           Qy = 280 ×0.2×0.15×27.5
                                                [.. A = Lxd and assuming]
                                                        C= 280 W/ma/oc
        -, QT = 331M
Heat dissipated is less than the heat generated, so external cooling
is required.
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Therefore, bearing will operate with external cooling. s) If the bearing operates without external cooling, The operating temperature d'oil = 90°c. d, Amount of external cooling required, QL = Heat generated - Heat dissipated. Que = Qg - Qy = 282.743-231 .. Qt = 51.743W Let, in be the mass of oil required to comy acray exass heat War 3 = mx 7/4g/g x oc generated over heat dissipated. And, T=90-85=5°C (given) Heat taken away by heail, Qt = mxsxT. [specific heat of oil,]

g=1840 to 2100 J/kg/oc]

ec 51.743 = mx1900x5 m= 51.743 =0.005 lg/sec). m=0.34 kg/min Select a suitable dimensions for a journal bearing which is subjected to 3:5 kN radial load. The diameter, and bugh of the bearing are 75mm each. The journal votate's at 400 r.pm. G.D: - Radial Load, W=2.5kN=3.5x102N. Length = dioneter = 75mm (of bearing) speed of journal N=400v.p.m. Consider the journal bearing design for centraged pump. Bearing pressure p = W = 2500 = 0.622 N/mm2. The bearing pressure is less than permissible value (i.e from standard) Take lubricant as SAE20 at 55°C . The operating value, i.e ZN = 0.0023 ×400 = 14.79. (Z=0.023=0.025 N=5/me)

From table operating value is 28 [i-e 3N = 28] Bearing modules at minimum point of faction is given as, $\lambda = \frac{1}{3} \left(\frac{2N}{P} \right) = \frac{1}{3} (28) = 9.33.$ The induced value i.e 14.79 is greater than the bearing wedness (i.e.9.33). Therefore, the bearing can operate under hydrodynamic conditions. c = 0.001/2 for centrifigal pumps. : Coefficient of friction is given by $M = \frac{33}{108} \left[\frac{2N}{P} \right] \left[\frac{d}{c} \right] + k$. = 33 (14.79) [1 0.002. (... Take k=0.002) 1. M=0.006 Dimensions of Bearing: -Diameter of journal (d), = 0.0013>) c=0.0013d (d= p. do / / / / 0. / 84/5) Diametrial dearance is given by c= D-d 0.0013 2=75-2. >> d(0.0013+1)=75 $d = \frac{75}{1.0012}$. d= 74.903mm :. Diametrial dearance, C= D-d = 75-74-903 C=0:0972000. Radial clearance, cr = = = 0.091 Cr = 0.0485mm. Minimum oil film thickness, tr= = = 0.097 = 0.0242mm tp =0.0242mm. Eccentricity e=C,-tp=0.0485-0.0242=0.0243mm e=0.0243mm.

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· 5dr 69.94 = mdr 0086 = N' poods mmE0.0= en unimos= an essential mig muminim Bearing length L=50mm. 5-100ml = Journal diomate & Somm. | Catepoir = I millipostical . grand sing babbabance bearing. provided so that when the journal is robuling at 2800 mpm, the minimum of the thickness is 30 microns. Evaluate the concepanding certificiant of Viscosity of oil = 20 mpg = Specifi the redict clearance that needs to be Rodid Lad = 2 kn; Tourned diameter = 50mm; Bearing langth = 50mm; The following date is given for a 360 hydrodynomic bearing. Lis length of beging, D is diameter of bearing. Disolar religio 21 00 Local Leiligo 21 W P is book per unit of projected bearing over a) we as a sold of the Bods of ail (vo) (Leeffect of dynamic (doesdo) viscosity (vo) a is the radial decreme. I which (or) coefficient of absolute nacert Evibor Hode all zin Gis the commerteld Number or bearing characteristic Number Sommerfeld Numbers. $S = \frac{(x)^3 \text{ A.C.}}{W} = \frac{(x)^3 \text{ A.C.}}{W} = \frac{2N}{Q} \left(\frac{d}{d}\right)^3 = 2$ 105.0=3/ · los·0 = 5840.0 = 3

Successibility ratio (Pater or altitude) 2 = 2 = 1 = 32 = 1 - 349

(absolute viscosity)
(coefficient of friction) M = 20 mpas = 20 × 10 9 N-Sec/mm2. / = 1" Performance parameters: Pressure P = W = 2×103 = 0.8N/mm2. The Sommerfell Number is given by $S = (\frac{r}{c})^2 (\frac{\mu \cdot N}{P})$ S= (252) 20×10 9×46.667 = 729×106 5 = 729×10-G The commented number depends on the ratio (ho) Assume [ho] = 0.6. Assure (ho) =0.9 C= ho = 0.03 = 0.05mm > 1 x C = ho | 0.03 =0.033 → 1). At c=0.033, 5=1.33. At C=0.05mm, S=0.264 Thus c2= 729 ×106 Thus, C2 = 729 × 10-6 From eq () ED it is observed that From eq O EO it is observed that assured rake is approximately equal to assumed value differs from calculated value of radia desance, calculated value of vadial charance (C) Coefficient of friction: From standards, (x) 4=5.79 (C) 4=5.79x (C) = 5,79 × (0.05)

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A journal bearing 100mm x 100mm Eupports a shaft narring at 1500 mpm. Radial clearance is 0.1mm, minimum film thickness is 0.02mm. Calculate the maximum radial load the bearing can support and for that load calabate the power last infriction and oil temperature vise. Consider absolute viscosity = 20e3 As. Specific gravity of oil = 0.88 and specific heat of oil = 2.1 10/10gk. G.D: - Diameter d = length l = loomm. [: Yd=1] Speed N= 1500r.p.m., Radiol clearance, C====0.1mm. :, C = 2×Cr = 2×0·1=0·3 mm. Minimum film thickness ho = 0.02 mm. Absolute viscosity, Z = 20e pars = 20 x103 = 0.03 pars. Specific quanty of oil Siz = 0.88, Specific heat of oil, C=9.1 K/19k From the table, For 1 = 1 and ho = 0.02 = 0.2 Gomes field number 5=0.0446. 1) Maximum Radial Load: Sommerfeld humber, $\tilde{S} = \frac{2N^{1}}{p} \times \left(\frac{d}{c}\right)^{2}$ (, N = 1500 HB) $0.0446 = \frac{125000}{P} > P = 2.8 \times 10^{6} Pa$ $1Pa = 10^{-6}N/m^{2}$ $1pa = N/m^{2}$ 0.0149= 0.05×1200 (100) 5

Bearing pressure P= W > Max radial load W= Pxlxd. M = 5.8 × 106 × (100) × (1000) = 58000 N. w= 2800000cm) 28 km.

@ Power last in friction: - It is gran by Heat generated by Dig = uwv From table ho = 0.2, ar is 1.70

Minimum oil film thickness = 22.5 microns

Inlet temperature = 40°c.

Bearing material = Babbit.

Determine the length of the bearing and select a suitable oil.

Given that :-

Journal diameter de 75 mm,

Radial Load WR = 10/2N = 10×103N.

Tournal speed, N= 1440r.p.m

Minimum oil film thickness ho = 22.5 microns. = 22.5 x103 mm = 0.025

Inlet temperature Ti = 40°c.

Bearing's material - Babbitt

Length of Bearing: -

Bearing pressure, P= WR Lxd.

Assume bearing pressure p=2N/mm2 for inachine tool applications.

$$L = \frac{W_R}{P \times d} = \frac{10 \times 10^3}{2 \times 75} = 66.67 \text{mm}.$$

$$\frac{1}{15} = \frac{66.67}{15} = 0.89$$

Assume standard value, & =1. > l=H=75mm.

Selection of Lubrant:-

Bearing pressure, P= NR = 10×103 = 1.78 N/mm2.

Radial dearance:

The standard radial clearance value for babbit material is,

For Machine Tools = = Cr = 0.00%

C+ = (0.001) x x = (0.001) x 75 = 0.0375mm

Minimum oil film thickness, variable, ho = 0.0295 = 0.6.

: From the dimensionless performance parameter for full journal bearing with side flow table.

Sommerfeld number, 5=0.264 and or vent = 3.99 where, q-Flow of the lubriant (mm3/s) and N= 1440 = 248-pis Sommerfeld number is given by $S = (x)^2 \frac{2N}{p}$ $Z = \frac{0.264 \times 1.78}{(000)^{3} \times 34} = \frac{1.78}{1.78} \times \frac{2 \times 34}{1.78} = \frac{1.78}{1.78} \times \frac{3.10}{1.78} = \frac{3.10}{1.78} \times \frac{3.10}{1.78} = \frac{3.10}{$ Z=19,58×10-9N-sec/mm2 or 19.58 Centipoise. solution of Lubrianti-Coefficient of friction variable, CF.V = (T) M = (d) M = 5.79.7 About variable F.V = TONE = 3.99 + (from data book) = Temperature vise $\Delta t = \frac{8.3p(cFV)}{F.V}$ (From bandhair Textbook) At = 18.3x1.78x6.79 = 21.44°C Average temperature of lubricating oil or operating temperature, Tavg = T; + (A) =40+21.44 = 50.782 From the viscosity-temperature graph, the value of viscosity for SAE-10 is 22 certipoise at 50.79°C. Therefore, for his operation SAE-10 oil is selected to neet the. winimum viscosity of 19.53 Entipoise.

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-> Select the suitable dimensions for a journal bearing which is is ubjected to 3.5km radial had. The diameter and length of the bearings are 75 mm each. The journal votates at 400 rpm. -: St. - G.D :- Radial Load, W= 3.5kN = 3.5 x 103N Length of bearing L=75mm, Diameter of bearing, d=75mm. Speed of journal, N=400 pm. Consider the journal bearing design for centrifuged pumps. : Bearing Pressure, P is given as, P= W = 3500 = 0.600 N/mm2 . The bearing pressure is less than permissible value (i.e from standards) Assume the Lubricant as SAERO at 55°C. .. The operating value, in ZN = 0.023 × 400 = 14.79. (: z=0.023 from tables) Bearing modulus at minimum point of friction is given as $k = \frac{3}{3} \left[\frac{A}{2N} \right] = \frac{3}{3} (38) = 9.33.$ The induce value i-e 14.79 is greater than the bearing modules (i.e. 9.33). Therefore, the bearing can operate under hydrodynamic Frontables, for Centifly pumps, = 00013. .. Coefficient of hiction (4) is given by, $M = \frac{33}{108} \left[\frac{2N}{P} \right] \left[\frac{d}{c} \right] + k$. M = 33 (14.79) (14.79) (10.0013) +0.002 = 0.006 (... Talak=0.002) Dimension of Beging: Diameter of journal (d), C=0.013>) C=0.00131 Diametral decrance is given as, c=D-d 0.00131 = 75-1

d(1+0.0013)=75 > d= 75 1.0013 Ę > d=74.903mm Diametral dearance, C= D-d =75-74.903 C=0-097mm Radial dearance, Cr = = = 0,097 Cr = 0.0485 mm) Minimum oil film thickness, te = = = 0.097 to = 0. DEHErry eccentricity e=er-tp. = 0.0485-0.0242. @=0.0843mm Eccentricity votio, $E = \frac{e}{c_T} = \frac{0.0943}{0.0485}$ ·· [8= 0.501] ...

" C = 15000H. For C= 161466.04 W. d= 75mm, Bearing No. 6315 is suitable. · 1740.904101=> (2 pringe 11pd ret E=4:) Dynamic load Gerach : 201 /1c = 8980 [140 x166] (grilled 21 200 voint 1= V ...) 408P8 = W (000E) \$1+0008 x 1 x 99.0 = AWY + 9WVX = W , bod Britished Indorup3 STE.O = ODE = AW = 1440×106 xx or 1440millionxu 00008×0081×09=711 Life Ly = 20,000 hous, y Life in remultions: 17×0×09=7 - mdroos1 = N 7 = 1500 mbm. shoff diameter d = 15mm. Axial Load Lous = Alw = 3x103h. Rodial Land WR = 8 kn = 8 x103 w bearing for this application. acceptable dienster of the shaft is 15 mm; select a switable ball The corrected life of the beginny is 20,000 hours. The minimum A single now deep grove ball bearing is subjected to a radial load of 3km. The shaft volates at 1900 rpm.

| Springs | Isd |

Wa = 3000 = 0.04167. From tables of dynamically located bearings, For WA 0.375 and Wa = 0.04167 , e=0.24. and WA = 0.375>e = 0.24. To find value & Y'. By interpolation, $(y-1.8) = \frac{1.8-1.6}{(0.07-0.04)} \times (0.04167-0.04) = (.811)$ Dynamic load Capacity, W=×WR+YWA =0.56x8000+ 1.811 ×3000 = 9913 N. C = W [106] 1/6 = 9913 [1440×106] = 111941.88 D. C=111941.81 N, d=75mm Co = 72000 N are near to the properties of bearing No 6315 -: For the given data; Bearing No 6315 (C=112000A) is suitable. Select a single vow deep groove ball bearing for a radial load of 4000N and on avial load of 5000N, operating at a speed of 1600 pm for an average life of 5 years at 10 hours per day. Assume uniform and steady load. G.D: - Radial Load, WR = 4000N. Axial Load, WA = 5000N.

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M=XNMB+YMA.

= 0.56 ×1×4,000 + 1.6×5000

= 1024DN = 10.24kN.

Dynamic load capacity, C= W(Lob) 1/k = 10240 (17.52 × 108) 1/3

C = 133446.275N = 193.446kN

C = 133446.275N = 193.446kN

21 4 21291 = 2 gring rolls redmy C= 19615 Nie · 07051.91=0 N560.05191=D [Bunosa llod ref E= 10] EN [801 x H 8.0] 0085 = ball bearing is 1. Therefore, the bearing should be selected for a bod bod of 5800h. Assuming uniform and stock boat boats frieter (les) for . CMD85 = M (1=v:) (oodex1) + (oodex1x320) = A'WY + gWVX = CN .: 1=x b no 32:0= x \2.0=0)/AW but 1200 = 0000 = AW = 40 = 100 Les des Los mon Assume, WA Le = 0.5 and bearing be simple now deep grove ball bearing. : AWA+9WVX = W boot loads traderings mirrorly seed 124 01× 46.0= 0001 × 004 × 09 = 7 Like of the bearing in revolutions, L= 60×10×2H Speed, N=400 rip.m, Expected Life of bearing, LH=1000hvs. Rodied Load We= 5000 U. Axial thust WA= 30000 *P*: G.D:-Diameter of shaft D=40mm. Life if the bearing is expected to be at least toochus. at boat temil wase a boat look had a soon brust lead. The Due to a benel gear mounted on the shaft, the bearing will have Select a bearing for a homm diameter shaft that rotates at 400 rpm

A ball bearing is subjected to radial forces of 2500N and an axial force of 100010. The dynamic load carrying capacity of the bearing is 7.350 N. The values of X and Y factors are 0.56 and 1.6 respectively. The shaft is rotating raorpm. Determine the like of being. Stil- G.D:- Radial force, WR = 25001- 19 peed N = 720 pm. Axial fora, WA = 1000 N. Dynamic load Conging Capacity 1, C= 7350N. Factors: x=0.56 (Rodial force); y=1.6 (Axialforce). Accuming that the load is winiform and steady, thus service factor is=1. Life of bearing is, LF [C] × 106 -> 0 Design dynamic equivalent load is given by, = [(x.V. WR) + (YWA)] xks, [: V=1 (for bearings)] W'= Wxks = [(0.56×1×2500)+ (1.6×1000)] ×1 W = 3000Nsubstituting above value in equ $L = \left[\frac{c}{W}\right]^{\frac{1}{2}} \times 10^{\frac{6}{3000}} = \left[\frac{7350}{3000}\right]^{\frac{3}{2}} \times 10^{\frac{6}{5}}$ [: k=3 for ball bearings] 1 = 147.06 1 x10 5 vev. 2. Like of bearing in hours is given by, LH = LOXN LH = Com. = 147.061 × 105 = 340.418 hvs 1. LH = 240. 418 hrs

Load aycle for 312 bearing is as follows. Radial load 5000N and axial load of 3000N at 900 rpm for 25% of time Radial Load of 60000 and axial load of 100000 at 1200 pm for 25% of time. Radial Load of 4000M and axial load of 1500M at 1000mm for remaining time. The loads are steady. Find the rating life of this bearing in year with

8: C.D: -

thrs perday.

,	W D				
	Type of load	Rodiol(n)	Thrustor Anial	Speed (R.p.m	Fraction of Cycle
	Uniform	5000	3000	1300 doo	85/100 = 0-35
	steady boad	4000	1500	1000	0.5

Assume radial and avial load factor to be land 1.5 respectively.

The basic dynamic equivalent radial load is given by.

W= (xV.WR+YWA).kg. [:: x=1, 7=1.5]

:. W= WR+1.5WA.

[ks=1 for steady load]

For different operating cycles, we have

W, = (WR, + 1.5 WA,) = (5000+ 1.5 ×3000) = 9500 N. Similary.

Wa = (6000 + 1.5 × 1000) = 7500 N"

W3= (4000 + 1.5 x 1500) = 6250N.

Life of bearing in revolutions is given by. L=60NLH.

But LH = 300 x 2 = 600 hours (: Assum year = 300 days working)

L = 60 x N x 600 = 36000 N vev.

Like of bearing for different operating cycle,

L=36000 × 900 × (0.25) tine fredien = 8.1×106 rev. 1

L2 = 36000 x (0.25) x 1200 = 10.8 x 10 Ger.

L3 = 36000 x 0.5 x 1000 = 18 x 106 ver.

The dynamic equivalent load is given by $N = \left[\frac{L_1(w_1)^3 + (L_2(w_2)^3 + L_3(w_3)^3}{L_1 + L_2 + L_3}\right]^{\frac{1}{3}}$

$$W = \begin{bmatrix} 8.1 \times 10^{6} \times (9500)^{3} + 10.8 \times 10^{6} \times (7500)^{3} + 18 \times 10^{6} (6250)^{3} \\ (8.1 \times 10^{6}) + (10.8 \times 10^{6}) + (18 \times 10^{6}) \end{bmatrix}$$

$$= \begin{bmatrix} 1.589 \times 10^{19} \\ 36.9 \times 10^{6} \end{bmatrix}^{1/3} = 7551.488$$

I. $W = 7551.488$

For bearing no 312, the basic dynamic load apacty
$$C = 64 \times 10^{3} \text{ Av}$$

The rating like of bearing in revolutions is given by
$$L_{R} = \begin{bmatrix} C \\ W \end{bmatrix} \times 10^{6}$$
But $k = 3$ for ball bearing,
$$L_{R} = \begin{bmatrix} 64 \times 10^{3} \\ 1551.488 \end{bmatrix}^{3} \times 10^{6} = 608.755 \times 10^{6} \text{ FeV}$$

$$L_{R} = \begin{bmatrix} 608.755 \times 10^{6} \text{ FeV} \end{bmatrix}$$

$$L_{R} = \begin{bmatrix} 608.755 \times 10^{6} \text{ FeV} \end{bmatrix}$$

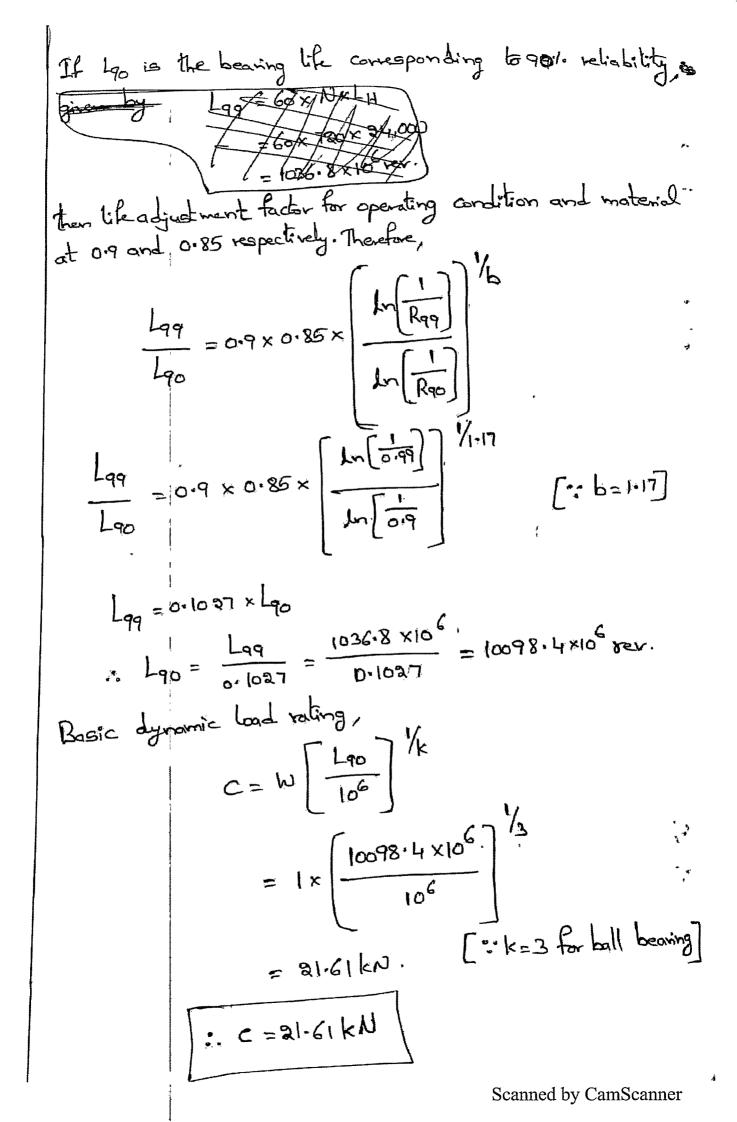
A ball bearing is operating on a work after consisting of three parts - a radial load of 3000N at 1440 rpm for one quarter again, a radial load of 5000N at 720 rpm for one half cycles and cycle, a radial load of 5000N at 1440 rpm for the remaining cycle. The radial load of 9500N at 1440 rpm for the remaining cycle. The radial load of the bearing is 10000 hrs. Calculate the dynamic load Canying Capacity of the bearing.

Resolutions N in Element Radial Speed Load P(N) (Minute) (hbm) Number 1/4 (1440) = 360res (N) 1/4 = 0.25 3000 1440 /2 (780) = 360 rev>1) 1/0 =0.5 720 1/4 (1440) = 360ren) 1 1/4 = 0:25 1440 3 500 1080 verolity

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Equirdent load Wis greenby $W = \frac{2}{N_1 + N_2 + N_2} \frac{N_1 + N_2 + N_3 + N_3 + N_3}{N_1 + N_2 + N_3}$ $W = 3 / \frac{360 (3000)^3 + 360 (5000)^3 + 360 (8500)^3}{360 + 360 + 360}$ = 3823.014N. According to load life relationship, Lio = GONLIOH, where Lio - Rated bearing (in millions revolutions). LIOH = Rated bearing like (hours). N=Speed of rotation (r.p.m). L10 = 60 (1080)(10000) = 648 million rev. Also, Dynamic load apacity C=W(Lio) 1/3. Dynamic load Capacity C=3893.014 (648) = 33082.442N :. C= 33082.442N -) A nothing contact ball bearing is to be selected to support an overhung countershaft. The shaft speed is 720 r.p.m. The bearing is to have 99%. reliability corresponding to a life of 24,000 hours. The bearing is subjected to an equivalent radial load of 1kN. Consider like adjustment factors for operating condition and material as 0.9 and G.D:- Shaft-speed N= Taor.p.m. Life in hours LH = 24,000 hours. 0.85 respectively. Equivalent radial Land W=1kN. The bearing like corresponding to 99.7. reliability is given by Lgg = Go x NxLH = 60 x 720 x 24,000 = 1036.8 ×106 vev.

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A shaft with two supports is located in such a way that the axial thrust on each bearing is soon and the radial thrust is 2000N and 1000N respectively at left and light hand bearing. The shaft volutes at 1440rpm. The beging is to be selected on the basis of 3 has penday for 10 years. The shoft diameter is Homm. Select he suitable bearings at both the ends.

10: G.D: - Axial houst, WA = 500N-Radial thrust on bearing-1, WR, = 20001 Radial thurst on bearing - 2, WR2 = 1000N. Speed N= 1440 pm, Diameter d=40mm

like = 3hours I day for 10 years.

מספב אנע WA = 500 N wi R==10000 WRIE SOON

The like of bearing in hours is given by LH = 3×365×10 = 10950 hours (. 1 pen = 365 days)

Like of bearing in revolutions.

L=60NLH = 60 x 1440 x 16950 1=946.08 × 106 zer = 946.08 million You

Right Hand Bearing:

Let, equivalent load be 1000N (WRE), : Dynamic load copacity, L= [wp.]

946.08 = 03 F. C = 9816.936N

Let us schet bearing number 6303 which has the following apacities, i.e C=1,3500N and G=6550N

WA = 500 = 0.076, e=0.27

Also, WA = 500 = 0.5 (: V=]

From standards, x=0.56 and y=1.575. Equivalent radial load, w= (x. v wR2) + (J.WA) = (0.56 ×1 × 1000) + (-575 ×500) W= 1347.5N. : like L= \(\frac{13500}{1347-5}\right)^{\frac{3}{2}} = 1005.576 million ver : Bearing number 6203 having C= 1350010 may be selected. Left Hand Bearing :-Let equivalent load be 30000 (WA) Dynamic had capacity is given by L= [] 946.08 = (300)3 > (C= 29450.808N) Let us select bearing number 6010 which has the following capacities. C=31600N, Co=13200N. : WA = 500 = 0.038, e < 0.24. Also WA >0.24 from standards x = 0.56 and y= 1.82. Eguiralent load, W= (x.V. WR,)+ (y.WA). = (1x0.56x2000)+(1.82x500) ≈ 8°3°N . Life, L= (21600)

= 1204-684 million ker which is approximately equal to 946.08 millian sev. .. The bearing number 6010 may be selected at left hand bearing.

Introduction

As the name implies, the internal combustion engines (briefly written as I. C. engines) are those engines in which the combustion of fuel takes place inside the engine cylinder. The I.C. engines use either petrol or diesel as their fuel. In petrol engines (also called *spark ignition engines* or *S.I engines*), the correct proportion of air and petrol is mixed in the carburettor and fed to engine cylinder where it is ignited by means of a spark produced at the spark plug. In diesel engines (also called *compression ignition engines* or *C.I engines*), only air is supplied to the engine cylinder during suction stroke and it is compressed to a very high pressure, thereby raising its temperature from 600°C to 1000°C. The desired quantity of fuel (diesel) is now injected into the engine cylinder in the form of a very fine spray and gets ignited when comes in contact with the hot air. The operating cycle of an I.C. engine may be completed either by the two strokes or four strokes of the

piston. Thus, an engine which requires two strokes of the piston or one complete revolution of the crankshaft to complete the cycle, is known as two stroke engine. An engine which requires four strokes of the piston or two complete revolutions of the crankshaft to complete the cycle, is known as four stroke engine.

The two stroke petrol engines are generally employed in very light vehicles such as scooters, motor cycles and three wheelers. The two stroke diesel engines are generally employed in marine propulsion.

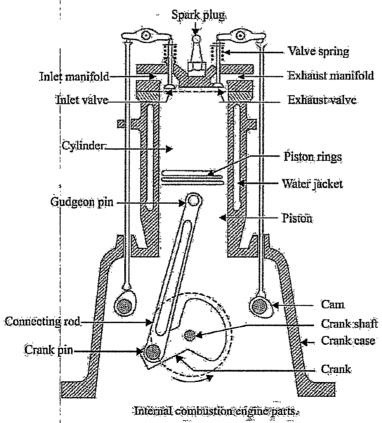
The four stroke petrol engines are generally employed in light vehicles such as cars, jeeps and also in aeroplanes. The four stroke diesel engines are generally employed in heavy duty vehicles such as buses, trucks, tractors, diesel locomotive and in the earth moving machinery.

Principal Parts of an Engine

The principal parts of an I.C engine, as shown in Fig. 32.1 are as follows:

1. Cylinder and cylinder liner, 2. Piston, piston rings and piston pin or gudgeon pin, 3. Connecting rod with small and big end bearing, 4. Crank, crankshaft and crank pin, and 5. Valve gear mechanism.

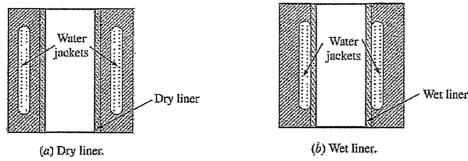
The design of the above mentioned principal parts are discussed, in detail, in the following pages.



Cylinder and Cylinder Liner

The function of a cylinder is to retain the working fluid and to guide the piston. The cylinders are usually made of cast iron or cast steel. Since the cylinder has to withstand high temperature due to the combustion of fuel, therefore, some arrangement must be provided to cool the cylinder. The single cylinder engines (such as scooters and motorcycles) are generally air cooled. They are provided with fins around the cylinder. The multi-cylinder engines (such as of cars) are provided with water jackets around the cylinders to cool it. In smaller engines, the cylinder, water jacket and the frame are made as one piece, but for all the larger engines, these parts are manufactured separately. The cylinders are provided with cylinder liners so that in case of wear, they can be easily replaced. The cylinder liners are of the following two types:

1. Dry liner, and 2. Wet liner.



Dry and wet liner,

A cylinder liner which does not have any direct contact with the engine cooling water, is known as dry liner, as shown in Fig. (a). A cylinder liner which have its outer surface in direct contact with the engine cooling water, is known as wet liner, as shown in Fig.

The cylinder liners are made from good quality close grained cast iron (i.e. pearlitic cast iron), nickel cast iron, nickel chromium cast iron. In some cases, nickel chromium cast steel with molybdenum may be used. The inner surface of the liner should be properly heat-treated in order to obtain a hard surface to reduce wear.

Design of a Cylinder

In designing a cylinder for an I. C. engine, it is required to determine the following values:

- 1. Thickness of the cylinder wall. The cylinder wall is subjected to gas pressure and the piston side thrust. The gas pressure produces the following two types of stresses:
 - (a) Longitudinal stress, and (b) Circumferential stress.

Since these two stressess act at right angles to each other, therefore, the net stress in each direction is reduced.

The piston side thrust tends to bend the cylinder wall, but the stress in the wall due to side thrust is very small and hence it may be neglected.

 D_0 = Outside diameter of the cylinder in mm,

D =Inside diameter of the cylinder in mm,

p = Maximum pressure inside the engine cylinder in N/mm².

t =Thickness of the cylinder wall in mm, and

1/m = Poisson's ratio. It is usually taken as 0.25.

The apparent longitudinal stress is given by

$$\sigma_i = \frac{\text{Force}}{\text{Area}} = \frac{\frac{\pi}{4} \times D^2 \times p}{\frac{\pi}{4} [(D_0)^2 - D^2]} = \frac{D^2 \cdot p}{(D_0)^2 - D^2}$$

and the apparent circumferential stresss is given by

$$\sigma_{it} = \frac{\text{Force}}{\text{Area}} = \frac{\hat{D} \times l \times p}{2t \times l} = \frac{D \times p}{2t}$$

... (where I is the length of the cylinder and area is the projected area)

 \therefore Net longitudinal stress = $\sigma_i - \frac{\sigma_c}{m}$

and net circumferential stress $= \sigma_{c} - \frac{\sigma_{l}}{m}$ The thickness of a cylinder wall (i) is usually obtained by using a thin cylindrical formula, i.e.,

$$t = \frac{p \otimes D}{2\sigma_n} + C$$

where

p = Maximum pressure inside the cylinder in N/min².

D =Inside diameter of the cylinder or cylinder bore in mm.

 $\sigma_c = \text{Permissible circumferential or hoop stress for the cylinder material}$ in MPa or N/mm². Its value may be taken from 35 MPa to 100 MPa depending upon the size and material of the cylinder.

C = Allowance for reboring.

The allowance for reboring (C) depending upon the cylinder bore (D) for L.C. engines is given in the following table:

Table 32.1. Allowance for reboring for I. C. engine cylinders.

D (mm),	75 ,	100	150	200	250	300-	350	400	450	500;
C'(mm).	1.5	24	4.0	6.3	8.0	9.5	11.0	12:5	12.5	12.5

The thickness of the cylinder wall usually varies from 4.5 mm to 25 mm or more depending upon the size of the cylinder. The thickness of the cylinder wall (t) may also be obtained from the following empirical relation, i.e.

$$t = 0.045 D + 1.6 \text{ mm}$$

The other empirical relations are as follows:

Thickness of the dry liner

= 0.03 D to 0.035 D

Thickness of the water jacket wall

= 0.032 D + 1.6 mm or t/3 m for bigger cylinders and 3t/4 for smaller cylinders

Water space between the outer cylinder wall and inner jacket wall

= 10 mm for a 75 mm cylinder to 75 mm for a 750 mm cylinder or $0.08 D \pm 6.5$ mm

2. Bore and length of the cylinder. The bore (i.e. inner diameter) and length of the cylinder may be determined as discussed below:

Let

 p_m = Indicated mean effective pressure in N/mm²,

D = Cylinder bore in mm,

A = Cross-sectional area of the cylinder in mm^2 ,

 $=\pi D^2/4$

l =Length of stroke in metres,

N =Speed of the engine in r.p.m., and

n =Number of working strokes per min

 $= N_s$ for two stroke engine

= N/2, for four stroke engine.

We know that the power produced inside the engine cylinder, i.e. indicated power,

$$I.P. = \frac{p_m \times l \times A \times n}{60}$$
 watts.

From this expression, the bore (D) and length of stroke (D) is determined. The length of stroke is generally taken as (1.25 D) to (2D).

Since there is a clearance on both sides of the cylinder, therefore length of the cylinder is taken as 15 percent greater than the length of stroke. In other words,

Length of the cylinder, $L = 1.15 \times \text{Length of stroke} = 1.15 t$

Notes: (a) If the power developed at the crankshaft, i.e. brake power (B. P.) and the mechanical efficiency (η_m) of the engine is known, then

$$LP_i = \frac{B_i P_i}{\eta_{ai}}$$

- (b) The maximum gas pressure (p) may be taken as 9 to 10 times the mean effective pressure (p_m) .
- 3. Cylinder flange and studs. The cylinders are cast integral with the upper half of the crank-case or they are attached to the crankcase by means of a flange with studs or bolts and nuts. The cylinder flange is integral with the cylinder and should be made thicker than the cylinder wall. The flange thickness should be taken as 1.2 t to 1.4 t, where t is the thickness of cylinder wall.

The diameter of the stude or bolts may be obtained by equating the gas load due to the maximum pressure in the cylinder to the resisting force offered by all the stude or bolts. Mathematically,

$$\frac{\pi}{4} \times D^2 \cdot p = n_s \times \frac{\pi}{4} (d_c)^2 \sigma_i$$

where

D = Cylinder bore in mm,

 $p = \text{Maximum pressure in N/mm}^2$,

 $n_r = \text{Number of studs.}$ It may be taken as 0.01 D + 4 to 0.02 D + 4

 $d_{\rm e}$ = Core or minor diameter, $k\dot{e}$, diameter at the root of the thread in mm.

 σ_t = Allowable tensile stress for the material of studs or bolts in MPa or N/mm². It may be taken as 35 to 70 MPa.

The nominal or major diameter of the stud or bolt (d) usually lies between 0.75 t_f to t_f where t_f is the thickness of flange. In no case, a stud or bolt less than 16 mm diameter should be used.

The distance of the flange from the centre of the hole for the stud or bolt should not be less than d+6 mm and not more than 1.5 d, where d is the nominal diameter of the stud or bolt.

In order to make a leak proof joint, the pitch of the study or bolts should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$, where d is in mm.

4. Cylinder head. Usually, a separate cylinder head or cover is provided with most of the engines. It is, usually, made of box type section of considerable depth to accommodate ports for air and gas passages, inlet valve, exhaust valve and spark plug (in case of petrol engines) or atomiser at the centre of the cover (in case of diesel engines).

The cylinder head may be approximately taken as a flat circular plate whose thickness (t_h) may be determined from the following relation:

$$t_h = D \sqrt{\frac{C.p}{\sigma_c}}$$

where

D = Cylinder bore in mm,

p = Maximum pressure inside the cylinder in N/mm²,

 σ_c = Allowable circumferential stress in MPa or N/mm². It may be taken as 30 to 50 MPa, and

C =Constant whose value is taken as 0.1.

The studs or bolts are screwed up tightly along with a metal gasket or asbestos packing to provide a leak proof joint between the cylinder and cylinder head. The tightness of the joint also depends upon the pitch of the bolts or studs, which should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$. The pitch circle diameter (D_p) is usually taken as D+3d. The studs or bolts are designed in the same way as discussed above.

Example 32.1. A four stroke diesel engine has the following specifications:

Brake power = 5 kW; Speed = 1200 rp.m.; Indicated mean effective pressure = 0.35 N/mm²; Mechanical efficiency = 80 %.

Determine: 1. bore and length of the cylinder; 2. thickness of the cylinder head; and 3. size of studs for the cylinder head.

Solution. Given: B.P. = 5kW = 5000 W; N = 1200 r.p.m. or n = N/2 = 600; $p_m = 0.35 \text{ N/mm}^2 / \eta_m = 80\% = 0.8$

1. Bore and length of cylinder

Let

D =Bore of the cylinder in mm,

$$A = \text{Cross-sectional area of the cylinder} = \frac{\pi}{4} \times D^2 \text{ mm}^2$$

I =Length of the stroke in m.

$$= 1.5 D \, \text{mm} = 1.5 \, D / 1000 \, \text{m}$$

We know that the indicated power, $I.P = B.P. / \eta_m = 5000 / 0.8 = 6250 \text{ W}$

We also know that the indicated power (I.P.),

$$6250 = \frac{P_m L.A.n}{60} = \frac{0.35 \times 1.5D \times \pi D^2 \times 600}{60 \times 1000 \times 4} = 4.12 \times 10^{-3} D^3$$

...(: For four stroke engine, n = N/2) $D^3 = 6250/4.12 \times 10^{-3} = 1517 \times 10^3$ or D = 115 mm Ans.

....(Assume)

and

$$l = 1.5 D = 1.5 \times 115 = 172.5 \text{ mm}$$

Taking a clearance on both sides of the cylinder equal to 15% of the stroke, therefore length of the cylinder, $L = 1.15 l = 1.15 \times 172.5 = 198$ say 200 mm Ans.

2. Thickness of the cylinder head

Since the maximum pressure (p) in the engine cylinder is taken as 9 to 10 times the mean effective pressure (p.), therefore let us take

$$p = 9 p_m = 9 \times 0.35 = 3.15 \text{ N/mm}^2$$

We know that thickness of the cyclinder head,

$$t_h = D\sqrt{\frac{C.p}{\sigma_t}} = 115\sqrt{\frac{0.1 \times 3.15}{42}} = 9.96 \text{ say } 10 \text{ mm Ans.}$$

...(Taking C = 0.1 and $\sigma_i = 42$ MPa = 42 N/mm²)

3. Size of study for the cylinder head

Let

d = Nominal diameter of the stud in mm,

 d_c = Core diameter of the stud in mm. It is usually taken as 0.84 d.

σ, = Tensile stress for the material of the stud which is usually nickel

 $n_{\star} = \text{Number of studs}.$

We know that the force acting on the cylinder head (or on the studs)

$$= \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (115)^2 3.15 = 32.702 \text{ N}$$
 ...(i)

The number of stude (n_s) are usually taken between 0.01 D+4 (i.e. 0.01 \times 115 \pm 4 = 5.15) and 0.02D + 4 (i.e. $0.02 \times 115 + 4 = 6.3$). Let us take n = 6.

We know that resisting force offered by all the studs

=
$$n_s \times \frac{\pi}{4} (d_c)^2 \sigma_t = 6 \times \frac{\pi}{4} (0.84d)^2 65 = 216 d^2 N$$
 ...(ii)

...(Taking $\sigma = 65 \text{ MPa} = 65 \text{ N/mm}^2$)

From equations (i) and (ii), $d^2 = 32.702/216 = 151$ or d = 12.3 say 14 mm

The pitch circle diameter of the study (D_p) is taken D+3d.

$$D_n = 115 + 3 \times 14 = 157 \text{ mm}$$

$$D_p = 115 + 3 \times 14 = 157 \text{ mm}$$
We know that pitch of the studs
$$= \frac{\pi \times D_p}{n_s} = \frac{\pi \times 157}{6} = 82.2 \text{ mm}$$

We know that for a leak-proof joint, the pitch of the study should lie between $19\sqrt{d}$ to $28.5\sqrt{d}$, where d is the nominal diameter of the stud.

:. Minimum pitch of the stude = $19\sqrt{d}$ = $19\sqrt{14}$ = 71.1 mm

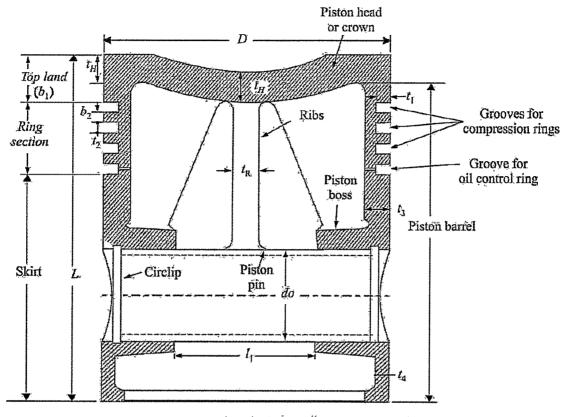
and maximum pitch of the study = $28.5\sqrt{d}$ = $28.5\sqrt{14}$ = 106.6 mm.

Since the pitch of the study obtained above (i.e. 82.2 mm) lies within 71.1 mm and 106.6 mm, therefore, size of the stud (d) calculated above is satisfactory.

$$d = 14 \,\mathrm{mm}$$
 Ans.

Piston

The piston is a disc which reciprocates within a cylinder. It is either moved by the fluid or it moves the fluid which enters the cylinder. The main function of the piston of an internal combustion engine is to receive the impulse from the expanding gas and to transmit the energy to the crankshaft through the connecting rod. The piston must also disperse a large amount of heat from the combustion chamber to the cylinder walls.



Piston for I.C. engines (Trunk type).

The piston of internal combustion engines are usually of trunk type as shown in Fig. 32.3. Such pistons are open at one end and consists of the following parts:

- 1. Head or crown. The piston head or crown may be flat, convex or concave depending upon the design of combustion chamber. It withstands the pressure of gas in the cylinder.
- 2. Piston rings. The piston rings are used to seal the cyliner in order to prevent leakage of the gas past the piston.
- 3. Skirt. The skirt acts as a bearing for the side thrust of the connecting rod on the walls of cylinder.
- 4. Piston pin. It is also called gudgeon pin or wrist pin. It is used to connect the piston to the connecting rod.

Design Considerations for a Piston

In designing a piston for I.C. engine, the following points should be taken into consideration:

- 1. It should have enormous strength to withstand the high gas pressure and inertia forces.
- 2. It should have minimum mass to minimise the inertia forces.
- 3. It should form an effective gas and oil sealing of the cylinder.
- 4. It should provide sufficient bearing area to prevent undue wear.
- 5. It should disprese the heat of combustion quickly to the cylinder walls.
- 6. It should have high speed reciprocation without noise.
- 7. It should be of sufficient rigid construction to withstand thermal and mechanical distortion.
- 8. It should have sufficient support for the piston pin.

Material for Pistons

The most commonly used materials for pistons of I.C. engines are cast iron, cast aluminium, forged aluminium, cast steel and forged steel. The cast iron pistons are used for moderately rated engines with piston speeds below 6 m/s and aluminium alloy pistons are used for highly rated engines running at higher piston speeds. It may be noted that

- 1. Since the *coefficient of thermal expansion for aluminium is about 2.5 times that of cast iron, therefore, a greater clearance must be provided between the piston and the cylinder wall (than with cast iron piston) in order to prevent siezing of the piston when engine runs continuously under heavy loads. But if excessive clearance is allowed, then the piston will develop 'piston slap' while it is cold and this iendency increases with wear. The less clearance between the piston and the cylinder wall will lead to siezing of piston.
- 2. Since the aluminium alloys used for pistons have high **heat conductivity (nearly four times that of cast iron), therefore, these pistons ensure high rate of heat transfer and thus keeps down the maximum temperature difference between the centre and edges of the piston head or crown.

Notes: (a) For a cast from piston, the temperature at the centre of the piston head (T_c) is about 425°C to 450°C under full load conditions and the temperature at the edges of the piston head ($T_{\rm E}$) is about 200°C to 225°C.

- (b) For aluminium alloy pistons, T_C is about 260°C to 290°C and T_E is about 185°C to 215°C.
- 3. Since the aluminium alloys are about ***three films lighter than cast iron, therfore, its mechanical strength is good at low temperatures, but they lose their strength (about 50%) at temperatures above 325°C. Sometimes, the pistons of aluminium alloys are coated with aluminium exide by an electrical method:

32.8 Piston Head or Crown

The piston head or crown is designed keeping in view the following two main considerations, i.e.

- 1. It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and
 - 2. It should dissipate the heat of combustion to the cylinder walls as quickly as possible.

On the basis of first consideration of straining action, the thickness of the piston head is determined by treating it as a flat circular plate of uniform thickness, fixed at the outer edges and subjected to a uniformly distributed load due to the gas pressure over the entire cross-section.

The thickness of the piston head $(t_{\rm H})$, according to Grashoff's formula is given by

$$z_{\rm H} = \sqrt{\frac{3p.D^2}{16\sigma_t}} \text{ (in rnm)}$$
 ...(1)

where

 $p = \text{Maximum gas pressure or explosion pressure in N/mm}^2$,

D = Cylinder bore or outside diameter of the piston in mm, and

 σ_{i} = Permissible bending (tensile) stress for the material of the piston in MPa or N/mm². It may be taken as 35 to 40 MPa for grey cast iron, 50 to 90 MPa for nickel cast iron and aluminium alloy and 60 to 100 MPa for forged steel.

On the basis of second consideration of heat transfer, the thickness of the piston head should be such that the heat absorbed by the piston due combustion of fuel is quickly transferred to the cylinder walls. Treating the piston head as a flat ciucular plate, its thickness is given by

$$t_{\rm H} = \frac{H}{12.56 k(T_{\rm C} - T_{\rm E})}$$
 (in mm) ...(ii)

The coefficient of thermal expansion for aluminium is 0.24 × 10⁻⁶ m f°C and for cast iron it is 0.1 × 10⁻⁶ m f°C.

The heat conductivity for aluminium is 174.75 W/m/°C and for cast from it is 46.6 W/m f°C.

The density of aluminium is 2700 kg/m³ and for east iron it is 7200 kg/m³.

where

H = Heat flowing through the piston head in kJ/s or watts,

k = Heat conductivity factor in W/m/°C. Its value is 46.6 W/m/°C for grey cast iron, 51.25 W/m/°C for steel and 174.75 W/m/°C for aluminium alloys.

 $T_{\rm C}$ = Temperture at the centre of the piston head in $^{\rm o}$ C, and

 $T_{\rm E}$ = Temperature at the edges of the piston head in °C.

The temperature difference $(T_{\rm C}-T_{\rm E})$ may be taken as 220°C for cast from and 75°C for aluminium. The heat flowing through the position head (H) may be determined by the following expression, i.e.,

$$H = C \times HCV \times m \times B.P.$$
 (in kW)

C =Constant representing that portion of the heat supplied to the engine which is absorbed by the piston. Its value is usually taken

HCV = Higher calorific value of the fuel in kJ/kg. It may be taken as 45×10^3 kJ/kg for diesel and 47×10^3 kJ/kg for petrol,

m = Mass of the fuel used in kg per brake power per second, and

B.P. = Brake power of the engine per cylinder

Notes: 1. The thickness of the piston head (t_H) is calculated by using equations (i) and (ii) and larger of the two values obtained should be adopted.

- 2. When $t_{\rm H}$ is 6 mm or less, then no ribs are required to strengthen the piston head against gas loads. But when $t_{\rm H}$ is greater then 6 mm, then a suitable number of ribs at the centre line of the boss extending around the skirt should be provided to distribute the side thrust from the connecting rod and thus to prevent distortion of the skirt. The thickness of the ribs may be takes as $t_{\rm H}/3$ to $t_{\rm H}/2$.
- 3. For engines having length of stroke to cylinder hore (L/D) ratio upto 1.5, a cup is provided in the top of the piston head with a radius equal to 0.7 D. This is done to provide a space for combustion chamber.

32.9 Piston Rings

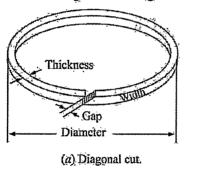
The piston rings are used to impart the necessary radial pressure to maintain the seal between the piston and the cylinder bore. These are usually made of grey cast iron or alloy cast iron because of their good wearing properties and also they retain spring characteristics even at high temperatures. The piston rings are of the following two types:

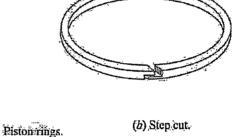
1. Compression rings or pressure rings, and 2. Oil control rings or oil scraper.

The compression rings or pressure rings are inserted in the grooves at the top portion of the piston and may be three to seven in number. These rings also transfer hear from the piston to the cylinder liner and absorb some part of the piston fluctuation due to the side thrust.

The oil control rings or oil scrapers are provided below the compression rings. These rings provide proper lubrication to the liner by allowing sufficient oil to move up during upward stroke and at the same time scraps the lubricating oil from the surface of the liner in order to minimise the flow of the oil to the combustion chamber.

The compression rings are usually made of rectangular cross-section and the diameter of the ring is slightly larger than the cylinder bore. A part of the ring is cut-off in order to permit it to go into the cylinder against the liner wall. The diagonal cut or step cut ends, as shown in Fig. 32.4 (a) and (b) respectively, may be used. The gap between the ends should be sufficiently large when the ring is put cold so that even at the highest temperature, the ends do not touch each other when the ring expands, otherwise there might be buckling of the ring.





The radial thickness (t_1) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring. From bending stress consideration in the ring, the radial thickness is given by

$$t_1 = D \sqrt{\frac{3p_w}{\sigma_p}}$$

where

D = Cylinder bore in mm,

 $p_{_{\rm IP}}$ = Pressure of gas on the cylinder wall in N/mm². Its value is limited from 0.025 N/mm² to 0.042 N/mm², and

σ, = Allowable bending (tensile) stress in MPa. Its value may be taken from 85 MPa to 110 MPa for cast iron rings.

The axial thickness (t_2) of the rings may be taken as $0.7 t_1$ to t_1 .

The minimum axial thickness (f_2) may also be obtained from the following empirical relation:

$$t_2 = \frac{D}{10n_R}$$
 where $n_{\hat{R}} = \text{Number of rings.}$

The width of the top land (i.e. the distance from the top of the piston to the first ring groove) is made larger than other ring lands to protect the top ring from high temperature conditions existing at the top of the piston,

 \therefore Width of top land, $b_1 = t_H$ to 1.2 t_H

The width of other ring lands (i.e. the distance between the ring grooves) in the piston may be made equal to or slightly less than the axial thickness of the ring (t_2) .

s. Width of other ring lands, $b_2 = 0.75 t_2$ to t_2

The depth of the ring grooves should be more than the depth of the ring so that the ring does not take any piston side thrust.

The gap between the free ends of the ring is given by 3.5 t_1 to 4 t_1 . The gap, when the ring is in the cylinder, should be 0.002 D to 0.004 D.

32.10 Piston Barrel

It is a cylindrical portion of the piston. The maximum thickness (t_3) of the piston barrel may be obtained from the following empirical relation:

$$t_{\pi} = 0.03 D + b + 4.5 \text{ mm}$$

where

 $b = \text{Radial depth of piston ring groove which is taken as 0.4 mm larger than the radial thickness of the piston ring <math>(f_1)$

$$= t_1 + 0.4 \, \text{min}$$

Thus, the above relation may be written as

$$t_3 = 0.03 D + t_1 + 4.9 \text{ mm}$$

The piston wall thickness (t_4) towards the open end is decreased and should be taken as $0.25 t_3$ to $0.35 t_4$.

32.11 Piston Skirt

The portion of the piston below the ring section is known as *piston skirt*. In acts as a bearing for the side thrust of the connecting rod. The length of the piston skirt should be such that the bearing pressure on the piston bairel due to the side thrust does not exceed 0.25 N/mm² of the projected area for low speed engines and 0.5 N/mm² for high speed engines. It may be noted that the maximum thrust will be during the expansion stroke. The side thrust (R) on the cylinder liner is usually taken as 1/10 of the maximum gas load on the piston.

We know that maximum gas load on the piston,

$$P = p \times \frac{\pi D^2}{4}$$

.. Maximum side thrust on the cylinder,

 $R = P/10 = 0.1 \ p \times \frac{\pi D^2}{4} \qquad \qquad ...(0)$

where

 $p = \text{Maximum gas pressure in N/mm}^{\frac{1}{2}}$, and

D = Cylinder bore in mm.

The side thrust (R) is also given by

R =Bearing pressure \times Projected bearing area of the piston skirt

$$= p_b \times D \times l$$

where

$$l = \text{Length of the piston skirt in mm.}$$
 ...(ii)

From equations (i) and (ii), the length of the piston skirt (l) is determined. In actual practice, the length of the piston skirt is taken as 0.65 to 0.8 times the cylinder bore. Now the total length of the piston (L) is given by

L =Length of skirt + Length of ring section + Top land

The length of the piston usually varies between D and 1.5 D. It may be noted that a longer piston provides better bearing surface for quiet running of the engine, but it should not be made unnecessarily long as it will increase its own mass and thus the inertia forces.

32.12 Piston Pin

The piston pin (also called gudgeon pin or wrist pin) is used to connect the piston and the connecting rod. It is usually made hollow and tapered on the inside, the smallest

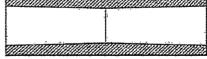
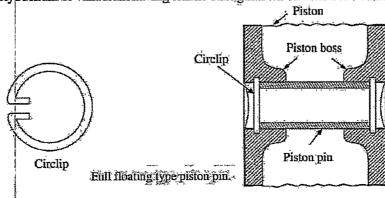


Fig.32.5. Piston pin.

inside diameter being at the centre of the pin, as shown in Fig. 32.5. The piston pin passes through the bosses provided on the inside of the piston skirt and the bush of the small end of the connecting red. The centre of piston pin should be $0.02\,D$ to $0.04\,D$ above the centre of the skirt, in order to off-set the turning effect of the friction and to obtain uniform distribution of pressure between the piston and the cylinder lines.

The material used for the piston pin is usually case hardened steel alloy containing nickel, chromium, molybdenum or vanadium having fensile strength from 710 MPa to 910 MPa.

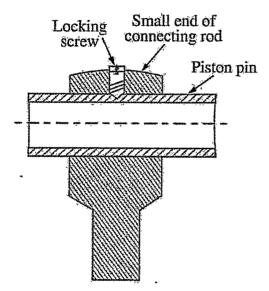


The connection between the piston pin and the small end of the connecting rod may be made either *full floating type* or *semi-floating type*. In the full floating type, the piston pin is free to turn both in the *piston bosses and the bush of the small end of the connecting rod. The end movements of the piston pin should be secured by means of spring circlips, as shown in Fig. 32.6, in order to prevent the pin from touching and scoring the cylinder liner.

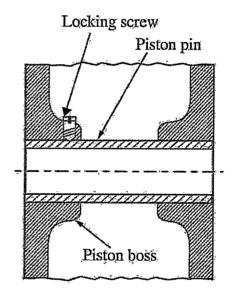
In the semi-floating type, the piston pin is either free to turn in the piston bosses and rigidly secured to the small end of the connecting rod, or it is free to turn in the bush of the small end of the connecting rod and is rigidly secured in the piston bosses by means of a screw, as shown in Fig. 32.7

The piston pin should be designed for the maximum gas load or the inertia force of the piston, whichever is larger. The bearing area of the piston pin should be about equally divided between the piston pin bosses and the connecting rod bushing. Thus, the length of the pin in the connecting rod bushing will be about 0.45 of the cylinder bore or piston diameter (D), allowing for the end clearance

of the pin etc. The outside diameter of the piston pin (d_0) is determined by equating the load on the piston due to gas pressure (p) and the load on the piston pin due to bearing pressure (p_{b1}) at the small end of the connecting rod bushing.



(a) Piston pin secured to the small end of the connecting rod.



(b) Piston pin secured to the boss of the piston.

Fig. 32.7. Semi-floating type piston pin.

Let

 d_0 = Outside diameter of the piston pin in mm

 l_1 = Length of the piston pin in the bush of the small end of the connecting rod in mm. Its value is usually taken as 0.45 D.

 p_{b1} = Bearing pressure at the small end of the connecting rod bushing in N/mm². Its value for the bronze bushing may be taken as 25 N/mm².

We know that load on the piston due to gas pressure or gas load

$$=\frac{\pi D^2}{4} \times p \qquad \qquad ..(i)$$

and load on the piston pin due to bearing pressure or bearing load

= Bearing pressure
$$\times$$
 Bearing area = $p_{b1} \times d_0 \times l_1$...(ii)

From equations (i) and (ii), the outside diameter of the piston pin (d_0) may be obtained.

* The mean diameter of the piston bosses is made 1.4 d_0 for east iron pistons and 1.5 d_0 for aluminium pistons, where d_0 is the outside diameter of the piston pin. The piston bosses are usually tapered, increasing the diameter towards the piston wall.

The piston pin may be checked in bending by assuming the gas load to be uniformly distributed

over the length l_1 with supports at the centre of the bosses at the two ends. From Fig. 32.8, we find that the length between the supports,

$$l_2 = l_L + \frac{D - l_1}{2} = \frac{l_1 + D}{2}$$

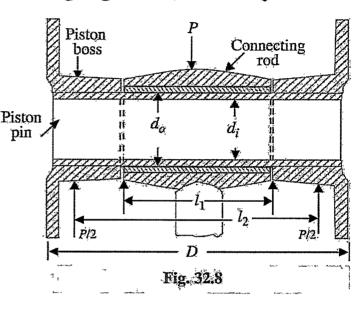
Now maximum bending moment at the centre of the pin,

$$M = \frac{P}{2} \times \frac{l_2}{2} - \left| \frac{P}{l_1} \times \frac{l_1}{2} \times \frac{l_1}{4} \right|$$

$$= \frac{P}{2} \times \frac{l_2}{2} - \frac{P}{2} \times \frac{l_1}{4}$$

$$= \frac{P}{2} \left(\frac{l_1 + D}{2 \times 2} \right) - \frac{P}{2} \times \frac{l_1}{4}$$

$$= \frac{P \cdot l_1}{8} + \frac{P \cdot D}{8} - \frac{P \cdot l_1}{8} = \frac{P \cdot D}{8}$$



...(iii)

We have already discussed that the piston pin is made hollow. Let d_0 and d_i be the outside and inside diameters of the piston pin. We know that the section modulus,

$$Z = \frac{\pi}{32} \left[\frac{(d_0)^4 - (d_i)^4}{d_0} \right]$$

We know that maximum bending moment,

$$M = Z \times \sigma_b = \frac{\pi}{32} \left[\frac{(d_0)^4 - (d_i)^4}{d_0} \right] \sigma_b$$

where

 σ_b = Allowable bending stress for the material of the piston pin. It is usually taken as 84 MPa for case hardened carbon steel and 140 MPa for heat treated alloy steel.

Assuming $d_i = 0.6 d_0$, the induced bending stress in the piston pin may be checked.

Example 32.2. Design a cast iron piston for a single acting four stroke engine for the following data:

Cylinder bore = 100 mm; Stroke = 125 mm; Maximum gas pressure = 5 N/mm²; Indicated mean effective pressure = $0.75 \,\mathrm{N/mm^2}$; Mechanical efficiency = 80%; Fuel consumption = $0.15 \,\mathrm{kg}$. per brake power per hour; Higher calorific value of fuel = 42×10^3 kJ/kg; Speed = 2000 r.p.m.

Any other data required for the design may be assumed.

Solution. Given: D = 100 mm; L = 125 mm = 0.125 m; $p = 5 \text{ N/mm}^2$; $p_m = 0.75 \text{ N/mm}^2$; $\eta_m = 80\% = 0.8$; m = 0.15 kg. / BP / h = 41.7 × 10⁻⁶ kg. / BP / s; $HCV = 42 \times 10^3$ kJ / kg; N = 2000 r.p.m.

The dimensions for various components of the piston are determined as follows:

1. Piston head or crown

The thickness of the piston head or crown is determined on the basis of strength as well as on the basis of heat dissipation and the larger of the two values is adopted.

We know that the thickness of piston head on the basis of strength,

$$I_{\rm H} = \sqrt{\frac{3p.D^2}{16 \, \sigma_t}} = \sqrt{\frac{3 \times 5(100)^2}{16 \times 38}} = 15.7 \, \text{say } 16 \, \text{mm}$$

...(Taking of for cast iron = 38 MPa = 38 N/mm2)

Since the engine is a four stroke engine, therefore, the number of working strokes per minute,

$$n = N/2 = 2000/2 = 1000$$

and cross-sectional area of the cylinder,

$$A = \frac{\pi D^2}{4} = \frac{\pi (100)^2}{4} = 7855 \text{ mm}^2$$

We know that indicated power

$$IP = \frac{p_m.L.A.n}{60} = \frac{0.75 \times 0.125 \times 7855 \times 1000}{60} = 12.270 \text{ W}$$
$$= 12.27 \text{ kW}$$

.. Brake power,

$$BP = IP \times \eta_{in} = 12.27 \times 0.8 = 9.8 \text{ kW}$$

 $m(: \eta_m = BP/IP)$

....(Taking C = 0.05)

We know that the heat flowing through the piston head,

$$H = C \times HCV \times m \times BP$$

= 0.05 × 42 × 10³ × 41.7 × 10⁻⁶ × 9.8 = 0.86 kW = 860 W

: Thickness of the piston head on the basis of heat dissipation,

$$t_{\rm H} = \frac{H}{12.56 k (T_{\rm C} - T_{\rm E})} = \frac{860}{12.56 \times 46.6 \times 220} = 0.0067 \,\text{m} = 6.7 \,\text{mm}$$

...(\because For cast iron , k = 46.6 W/m/°C, and $T_C - T_E = 220 \text{ °C}$)

Taking the larger of the two values, we shall adopt $t_{H} = 16 \text{ mm}$ Ans.

Since the ratio of L/D is 1.25, therefore a cup in the top of the piston head with a radius equal to 0.7 D (i.e. 70 mm) is provided.

2. Radial ribs

The radial ribs may be four in number. The thickness of the ribs varies from $t_{\rm H}/3$ to $t_{\rm H}/2$.

 \therefore Thickness of the ribs, $t_R = 16/3$ to 16/2 = 5.33 to 8 mm

Let us adopt

$$t_{\hat{\mathbf{R}}} = 7 \,\mathrm{mm}$$
 Ans.

3. Piston rings

Let us assume that there are total four rings (i.e. $n_r = 4$) out of which three are compression rings and one is an oil ring.

We know that the radial thickness of the piston rings

$$t_1 = D\sqrt{\frac{3p_w}{\sigma_t}} = 100\sqrt{\frac{3 \times 0.035}{90}} = 3.4 \text{ mm}$$

... (Taking $p_{s} = 0.035 \text{ N/min}^2$, and $\sigma_{r} = 90 \text{ MPa}$)

and axial thickness of the piston rings
$$t_2 = 0.7 t_1$$
 to $t_1 = 0.7 \times 3.4$ to 3.4 mm = 2.38 to 3.4 mm

Let us adopt

$$t_0 = 3 \, \text{mm}$$

We also know that the minimum axial thickness of the pistion ring,

$$t_2 = \frac{D}{10 n_p} = \frac{100}{10 \times 4} = 2.5 \text{ mm}$$

Thus the axial thickness of the piston ring as already calculated (i.e. $t_2 = 3$ mm) is satisfactory. Ans.

The distance from the top of the piston to the first ring groove, i.e. the width of the top land,

$$b_1 = t_H \text{ to } 1.2 t_H = 16 \text{ to } 1.2 \times 16 \text{ mm} = 16 \text{ to } 19.2 \text{ mm}$$

and width of other ring lands,

$$b_2 = 0.75 t_2$$
 to $t_2 = 0.75 \times 3$ to 3 mm = 2.25 to 3 mm

Let us adopt

 $b_1 = 18 \text{ mm}$; and $b_2 = 2.5 \text{ mm Ans}$.

We know that the gap between the free ends of the ring,

$$G_1 = 3.5 t_1 \text{ to } 4 t_1 = 3.5 \times 3.4 \text{ to } 4 \times 3.4 \text{ mm} = 11.9 \text{ to } 13.6 \text{ mm}$$

and the gap when the ring is in the cylinder,

$$G_2 = 0.002 D$$
 to 0.004 $D = 0.002 \times 100$ to 0.004 × 100 mm = 0.2 to 0.4 mm

Let us adopt

 $G_1 = 12.8 \text{ mm}$; and $G_2 = 0.3 \text{ mm Ans.}$

4. Piston barrel

Since the radial depth of the piston ring grooves (b) is about 0.4 mm more than the radial thickness of the piston rings (t_1) , therefore, $b = t_1 + 0.4 = 3.4 + 0.4 = 3.8$ mm

We know that the maximum thickness of barrel,

$$t_3 = 0.03 D + b + 4.5 \text{ mm} = 0.03 \times 100 + 3.8 + 4.5 = 11.3 \text{ mm}$$

and piston wall thickness towards the open end,

$$t_4 = 0.25 t_3$$
 to $0.35 t_3 = 0.25 \times 11.3$ to $0.35 \times 11.3 = 2.8$ to 3.9 mm $t_4 = 3.4$ mm

5. Piston skirt

Let us adopt

Let

l =Length of the skirt in mm.

We know that the maximum side thrust on the cylinder due to gas pressure (p),

$$R = \mu \times \frac{\pi D^2}{4} \times p = 0.1 \times \frac{\pi (100)^2}{4} \times 5 = 3928 \text{ N}$$
(Tables)

We also know that the side thrust due to bearing pressure on the piston barrel (p_{μ}).

$$R = p_b \times D \times l = 0.45 \times 100 \times l = 45 lN$$

...(Taking $p_6 = 0.45 \text{ N/mm}^2$)

From above, we find that

$$45 l = 3928 \text{ or } l = 3928 / 45 = 87.3 \text{ say } 90 \text{ mm Ans.}$$

.. Total length of the piston,

L = Length of the skirf + Length of the ring section + Top land =
$$I + (4 t_2 + 3b_2) + b_T$$
 = $90 + (4 \times 3 + 3 \times 3) + 18 = 129$ say 130 mm Ans.

6. Piston pin

Let

 $d_0 = \text{Outside diameter of the pin in mm}$

 $l_1=$ Length of pin in the bush of the small end of the connecting rod in mm. and

 p_{b1} = Bearing pressure at the small end of the connecting rod bushing in N/mm². It value for bronze bushing is taken as 25 N/mm².

We know that load on the pin due to bearing pressure

= Bearing pressure
$$\times$$
 Bearing area = $p_{b1} \times d_0 \times l_1$
= $25 \times d_0 \times 0.45 \times 100 = 1125 d_0 \text{ N}$... (Taking $l_1 = 0.45 D$)

We also know that maximum load on the piston due to gas pressure or maximum gas load

$$= \frac{\pi D^2}{4} \times p = \frac{\pi (100)^2}{4} \times 5 = 39.275 \text{ N}$$

From above, we find that

1125
$$d_0 = 39$$
 275 or $d_0 = 39$ 275 / 1125 = 34.9 say 35 mm Ans. $d_i = 0.6 \times 35 = 21$ mm Ans. The inside diameter of the pin (d) is usually taken as $0.6 d_0$.

Let the piston pin be made of heat treated alloy steel for which the bending stress (σ_b) may be taken as 140 MPa. Now let us check the induced bending stress in the pin.

We know that maximum bending moment at the centre of the pin,

$$M = \frac{P.D}{8} = \frac{39.275 \times 100}{8} = 491 \times 10^{3} \text{ N-mm}$$

We also know that maximum bending moment (M),

$$491 \times 10^{\frac{1}{3}} = \frac{\pi}{32} \left[\frac{(d_0)^4 - (d_i)^4}{d_0} \right] \sigma_b = \frac{\pi}{32} \left[\frac{(35)^4 - (21)^4}{35} \right] \sigma_b = 3664 \sigma_b$$

 $\sigma_b = 491 \times 10^3 / 3664 = 134 \text{ N/mm}^2 \text{ or MPa}$

Since the induced bending stress in the pin is less than the permissible value of 140 MPa (i.e. 140 N/mm^2), therefore, the dimensions for the pin as calculated above (i.e. $d_0 = 35 \text{ mm}$ and $d_i = 21 \text{ mm}$) are satisfactory.

-> (1) Design a suitable connecting rod for a cox with the following Connecting wol 17 dotta: Piston diameter = 68mm; Stroke length = 80mm; Length of the connecting rod = 160mm; Maximum explosion pressure = 3.5 Mpa; Weight of the reciprocating Parts = 2.5kg Speed = 4000 r.p.m; Compression ratio = 8:1 Given that, Drameter of piston, D=68mm Stroke length, L=80mm Crank length = stroke length = 80 = 40mm Length of connecting rod, Lc = 160mm Maximum explosion pressure, p=3.5N/mm2 Weight of secipsocating pasts, MR = 2.5kg Speed, N = 4000 8.p.m Compression ratio = 8:1 1. Cross-section of connecting rod. Let, thickness of flange and web=t With of section, B=4t Depth, H = 5t $\frac{T_{xx}}{T} \approx 3.2$, the section chosen is satisfactory. Load on the paton = TxD2xp Assuming factor of safety, F.S=5

Buckling load,
$$W_B = 10.710.88 \times 5$$

$$= 63554.4N$$
Radius of gyration about $x = ax15$

$$kxx = \sqrt{\frac{1}{Ax}}$$
Area, $t = 2(4txt) + 3txt = 11t^2$
Moment of inertia about $x = ax15$

$$\int_{xx} \frac{1}{12} \left[(4t)(5t)^3 - 3x1.5t \times 3t^3 \right]$$

$$= \frac{419}{12} t^4$$

$$kxx = \sqrt{\frac{419}{12}} t^4 = 1.78t$$
Using Renkine's formula for buckling load,
$$W_B = \frac{O_C \times A}{1 + O_C \cdot L} = \frac{1}{A}$$

$$L = \text{Equivalent length, since both the ends ase hinged.}$$

$$L = L_C \text{ (length of Connecting rad)}$$

$$W_B = \frac{320 \times 11t^2}{1 + \frac{1}{1007}}$$

$$63554.4 = \frac{3520t^2}{1 + \frac{1007}{12}}$$

$$63554.4 = \frac{3520t^2}{1 + \frac{1007}{12}}$$

$$63554.4 = \frac{3520t^2}{1 + \frac{1007}{12}} = 3520t^2$$

$$t^2 + 1.077 = \frac{3520}{63554.4}$$

$$co5538t^4 - t^2 - 1.077 = 0$$

$$t^2 = 1 + \sqrt{\frac{124}{124} + 20.05538 \times 1.077}$$

Scanned by CamScanner

t = 19.076

t = 4.36

t & 5mm

The dimension of section are,

t = 5mm

B=4t=4x5=20mm

H = 5t = 5x5 = 25 mm

Depth near big end,

H1 = 1.2H = 1.2 x25 =30000

Depth max small and

Ha = 0.7H to 0.9H

Taking it as 0.85H

H2 = 0.85 x 25 = 21.25 mm

Dimensions at big end = 30mm x 20mm

Dimensions at small end = 21.25 mm x 20mm

2. Design of the Big end Bearing or crank pin

Assuming the bearing pressure, Po = 12N/mm2

Let, d = Diameter of the big end bearing

Li = Length of the big end boosing

= 1.3 d, (Assume L1 = 1.3)

: Load on the big end bearing

= Projected area x Bearing Pressure

= d1 × L1 × Pb

= d, x1.3d, x12

= 15.6d2

Equating the load on the big end bearing or crank

Pen to the maximum gas pressure

15.6 dia = 12710.88

d2 = 814.8

```
d1 = 28.544 mm = 29mm
           L = 1.3 d = 13 x 28.544
              = 37.10mm & 38mm
3. Design of small End of the connecting Rod Piston pin
   Let,
    Diameter of piston pin = da
    Length of piston pin = La = 2 da (Assume La = 2)
   Also, assuming that booking pressure = 12 N/mm2
      Load on the piston pin = of 12 x12
                              = $ x2d2 x12
    Equating to the maximum force on the piston, we get
            24 da = 12,710.88
               da = 889 529.62
               da = 23.013 mm
               da & stomm
              La = 8d2 = 8 x 83.013
                 =46.026-mm 12 47mm
4. Size of the Bolts for securing Big End cap
     Let,
           dcb = Core diameter of the bolts
            ot = Allocable tensile stress
                = Go N/mod (Assume)
             Mr = Number of botts
   inforce on botts = I (dcb) x of xmb
                  = \frac{\pi}{4} (d<sub>Cb</sub>) & x 60 x 2 (Assume n_b = 2)
    :Inostia force of the reciprocating Parts
             Et = WE COS X & [COSO + COSO]
```

Fig. 2.5 × $\left(\frac{271 \times 4000}{60}\right)^2$ × 0.040 × $\left[1 + \frac{0.040}{0.160}\right]$ = 2.5 × 175459.63 × 0.040 × $\left[1 + 0.25\right]$ = 21932.45N

Equating the force on boths to the inertia force, $\frac{11}{4} \left(\frac{1}{4}\right)^2 \times 60 \times 2 = 21932.45$ $\frac{1}{4} \left(\frac{1}{4}\right)^2 \times 60 \times 2 = 21932.45$ And $\frac{1}{4} = 232.71$ And $\frac{1}{4} = \frac{15.25}{0.84} = 18.15 × 19 mm$ Nominal diameter = $\frac{15.25}{0.84} = 18.15 × 19 mm$ Design a mild steel connecting rod with an 1-section for a single cylinder spook ignition engine from the following data,

→(2)

Design a mild Steel Connecting rod with an 1-section for a Single cylinder spook ignition engine from the following data, Diameter of Piston, D=0. lowing Weight of reciprocating parts = 18.2N.

Length of Connecting rod from centre to Centre = 0.314m Stroke length = 0.14m

Stroke length = 0.14m

Speed of the engine = 1500 rev/min

Moximum explosion pressure = 2.28Mpg gauge

Assume that the maximum throat takes place at T.D.C during the explosion Stroke.

-Ans:

Given that,

Diameter of piston, D = lowmon

Weight of reciprocating parts, WR = 18.2N

Mass of reciprocating parts, MR = $\frac{18.2}{9.81}$ = 1.855 kg

Length of Connecting rod, L = 314 mm

Stroke = 140mm

i Grank length, 8 = 140 = 70mm

Speed, N = 1500 8.p.m

Explosion pressure, p= a. 28 N/mma Dimension of I - Section Let width of the section, B=4t Height of the section, H=5t : Asea = 11t2, Txx = 419+4, Tyy = 131+4 .: Maximum force on the piston, Fc = TD2 xp = Tr (104) 2 x 2.28 = 19368.29N Assuming the factor of Safety as Fis=6 Buckling load, MB = FL x F. = 19368.29 x6 = 16209.7N Radius of gysation, k = 1.78+ MB = (1+ 0x (1/2)2 $\frac{116209.7 = \frac{320 \times 114^{2}}{1 + \frac{1}{7500} \cdot \frac{314}{1.781}}^{2}}{1 + \frac{1}{7500} \cdot \frac{314}{1.781}}^{2}$ $33 = \frac{\pm 8}{1 + 4.15}$ t4-33+2-136.95 =0 t=6,000 "Will of the section, B=4t =4x6 =24mm Height of the section, H=5t = 5x6 = 30mm : Depth new the big and, H = 1.2 x 30 = 36mm Depth near the Small end, Ha = 0.85H = 0.85 ×30 = 25.5mm .. Dimensions of the section near the big end = 36x24mm Dimensions of the section near the small end = 25.5x24mm 2. Size of Botts for Securing the Big End Cap Let, det = Core diameter of the botts of = Allocable tensile stress

20

= 60 N/mm2 (Assume) nb - Number of botts.

: force on the botts,

.. Inertia force of the secipsocating pasts,

$$F_1 = M_R \cos^2 \left(\cos \theta + \frac{\cos 2\theta}{4/r}\right)$$
 (Taking $\theta = 0^\circ$)

$$= 1.855 \times \left(\frac{2 \times 11 \times 1500}{60}\right)^{2} \times 0.07 \left(\frac{1 + 0.07}{0.314}\right)$$

= 3917.98N

i. Equating inextia force to the force on the botts, 3917.98 = 94.26(gcP)3

: dcb = 6.44 & 7-mm

Nominal diameter of bott,

$$d_b = \frac{d_{cb}}{0.84} = \frac{7}{0.84} = 8.5$$

-3. Determine the dimensions of an I-section connecting rad for a petrol engine from the following data, Diameter of the piston = 110 mm, Mass of the reciporating parts = 2kg, Length of the Connecting rod from Centre to Centre=325mm Stroke length = 150mm, R.p.M = 500 with possible overspeed of \$500, Compression ratio = 4:1, Maximum explosion pressure = 2.5 N/min Green that,

Diameter of the piston, D=110mm Mass of the reciprocating parts, MR = 2kg Length of the connecting sod, 1=325mm Stroke = 150mm Spee d, N = 1500 8.pm

Possible over speed = 2500 r.p.m Compression ratio = 4:1 Maximum explosion pressure, p= 2.5 N/mm2 1. Dimensions of I-section of Connecting Rod Let flange and web thickness of the Section = t Width of the section, B=4t Depth or height of the section, H = 5t In order to have a connecting and equally strong about both the axis. $T_{xx} = 4T_{yy}$ Ixx is kept slightly less than 4744. Now, area of the section is Considered as, A = 2 (4t x-t) + 3t x t = 11t2 $\therefore T_{xx} = \frac{BH^3}{12}$ $T_{xx} = \frac{1}{12} \left[4t (5t)^3 - 3t (3t)^3 \right]$ = 419+4 $T_{yy} = 2 \times \frac{1}{12} \times t (4t)^3 + \frac{1}{12} \times 3t \times t^3$ $=\frac{1}{12}\left[2+x(4+)^3+3+x+3\right]$ $=\frac{131}{12}$ +4 $\frac{T_{xx}}{T_{yy}} = \frac{419t^4}{12} \times \frac{12}{131t^4} = 3.2$

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piston due to gas presence. (2) of the one of the month of the box for the on the one Alence the section is soft

N466-8916E= G. G. X (10) x 11 =

load about x-axis, accuming beth ends hinged. We know that, connecting took is designed for buckling

Buckling load, Mg = Fc x F.5

F=F=7=7=7

Fig - Factor of sodety = 5 (Assumed)

MOG-161811 = BX 4PE-830EB = BM

Radius of gyration of the section about X -exis is,

 $\frac{xx}{A}\sqrt{=xx}$

(6711=14 =) = 11+3 /=

Length of Goods, 78F-1=

1 = Stocke of poston

mmer = 001 =

mgL0.0 =

marget of the connecting tod, l= 305 m

is Equivalent length of the connecting sod for both ends

man 266 = 1 = 1

According to Rahkines togenula, we know that,

Buckling load,

$$H_{B} = \frac{2A}{1+\alpha \left\lfloor \frac{L}{K_{xx}} \right\rfloor^{2}}$$
(Assume, $\sigma_{c} = 330 \text{ N} \mid_{\text{rms}^{2}} \text{ and } \alpha = \frac{1}{7500}$)

$$118791.47 = \frac{330 \times 114^{2}}{1+\frac{1}{1500}}$$

$$359.974 = \frac{114^{2}}{1+\frac{14 \times 033}{1500}}$$

$$359.974(4^{2} + 4.44) = 114^{4}$$

$$359.9744^{2} + 1600.034 = 114^{4}$$

$$114^{4} - 359.9744^{2} - (600.034 = 0$$
By trail and esses method,

$$t = 6.0575 \text{ Say 7 mm}$$
Dimensions of the Connecting sod ase,

Thickness of the flarge and coeb of the section, temm width of the section, $B = 44 = 4 \times 7 = 28 \text{ mm}$
Depth for height of the section, $H = 54 = 5 \times 7 = 35 \text{ mm}$.

Width (B) is kept constant, but depth (H) vasies

Depth mass the big end or coank end is kept as lift to liast and the depth meas the Small end or piston end is kept as on 75H to 0.9H.

Depth meas the big end is,

 $H_{1} = 1.2 \times 35 = 42 \text{ mm}$
Depth meas the Small end is,

 $H_{2} = 0.85H = 0.85 \times 35 = 39.75 \text{ mm}$ Say 30 mm

Dimensions of the section near the big end is, =42mm x 28mm (: Wilth of section = 28m Dimensions of the section near the small and is, = 30mm x 28-mm. 2. Dimensions of Counkpin and Piston Pin il Big End Bearing: dc - Diameter of Crankpin Le-Length of the Grankpin = 1.1 de (Assume) Pbc-Bearing pressure = 8.5 N/mm? (Assume) Load on big end bearing or Crankpin. = projected areax Bessing pressure =dc lc x Pbc = dc x l·ldc x8.5 Crankpin or big and bearing is designed for the maximum gas force (F). 9.35 d = FL = 23758.294 . dc = \ \frac{23758.294}{9.35} = 50.408 mm cay 51mm. Lc=111dc=1.1x51=56.1mm say 57mm ii, Small End bearing or Piston Pin =let dp-Diameter of piston pin or small and bearing. lp-length of piston pin or small and bearing . - 1.8dp (Assur) Pbp - Bearing pressure = 14N/mme (Assume). Load on the piston pin or Small and bearing = dplp. Pop. = dp x1.8dp x 14 = 25.2dp. Piston pin or small end bearing is designed for the maximum gas fora (FL).

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26.2 (dp) = FL = 23758.294. dp = $\sqrt{\frac{23758 \cdot 294}{95.2}} = 30.705 mm Say 3 mm.$ lp=1.8 xdp And lp=1.8 ×31 = 55.8mm Say 56mm. iii) Size of Bolts for Securing the Big End Cap: Let db - Gove diameter of the bolts. of Allowable tensile stress for the material of the bolts. of = 100 N/mm & (Assume). Mb-Number of bolts (Generally two bolts are used) Force on boths = Th (dcb) 2 x of x nb. = 1 (deb) = 100 ×2 = 157.08 deb. Inertia force of the reciprocating parts, (: 0 = 311V) E = mems = [1+x] = 2 × [211×1500] × 0.15 × [1+ 0.395] = 4555 202 · 1 Equating the inertia force and force on the bolts, we get, 4555-202 = 157.08(deb). deb= 157.05 = 5.385. , Whit hominal diameter of bolt; db = dcb. db = 5-385 = 6.41mm. from standard of bolts, take do = 8mm.

```
Design a connecting and foor a petrol engine foram the following
 Diameter of the piston = 120mm.
 weight of the seciprocating Past = 2.0kg
 Length of the connecting and = 300 mm
Stroke length = 140 mm; speed = 2000 aipim
Maximum explosion Pressure = 2.25 Mpc
 Given that,
     Diameter of piston. D = 120mm
     Stroke length, L= 140mm
     Crank length = Stroke length = 140 = 70 mm
     weight of reciprocerting parts, MR = 2.0 kg
    Length of Connecting and Lc = 300 mm
    Speed, N = 2000 to Po m
    Maximum explosion pressure, P= 2.25 MPa
                                       = 2.25 N mm2
   Assume, Compression settio = 8:1
    factor of Safety FS = 5.
    Gross-section of connecting Rod
                           * 1. Let, the thickness of flange &
                                 web = t
                                  width of section, B=4t
                                  Depth, H = 5t
          54
                                AS TXX N 3.2, the Section
                                  chasen is satisfactory.
                               T Load on the piston = 1 xD2xp
                                            = \frac{\pi}{L} \times (120)^{8} \times 2.25
                                            = 25446.900
                                            ~ 25447-N
                             Buckling load, WB = 25447×F.S
```

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Radius of gyaration about x-axis,

$$k_{xx} = \sqrt{\frac{T_{xx}}{A}}$$

Area, A = 2(Ltxt) + 3txt = 11t2

Moment of inestia about x-oxis,

$$T_{xx} = \frac{1}{12} \left[(0+t) (5+t)^3 - a \times 1.5 + \times (3+t)^3 \right]$$

$$k_{xx} = \sqrt{\frac{419+4}{12x11+2}} = 1.782+$$

Rankine's formula for buckling load,

$$H_{\beta} = \frac{\sigma_{C} \times A}{1 + \alpha \left[\frac{L}{T_{xx}}\right]^{2}}$$

L - Equivalent Length, since both the ends are

hinged.

$$W_{B} = \frac{320 \times 11t^{8}}{1 + \frac{1}{7500} \left[\frac{300}{1.782} \right]}$$

$$W_{B} = \frac{320 \times 11t^{n}}{1 + \frac{1}{7500} \left[\frac{300}{1.782}\right]}$$

$$(-Assume \ \sigma_{c} = 320 \ N/mm^{8})$$

$$121235 = \frac{3520t^{8}}{1 + \frac{3.779}{t^{3}}}$$

$$1235 \left(1 + \frac{3.779}{t^{3}}\right) = 3520t^{8}$$

127235
$$\left(1 + \frac{3.779}{4^8}\right) = 35204^9$$

$$t^{2} + 3.779 = \frac{3520}{127925} t^{4}$$

$$t^{2} = \frac{1 + \sqrt{12 + 4 \times 0.028 \times 3.779}}{2 \times 0.028}$$

 $t^2 = 39.161$ t=6.258 mm The dimension of section are, t = 6.258 mm B=4t=4x6.858=25mm H=5t=5x6.258=31.290mm Depth now big end, H1 = 1.2 H = 1.2 x 31.290 = 37.548 mm <u>~ 38 mm</u> pepth near small end. the =0.7H to 0.9H Taking it as 0.85H Ha = 0-85 × 31.290 = 26.597 mm 2 27 mm Dimensions at big end = 38 mm x 25 mm Dimensions at small and = 27 mm x 25 mm a. Design of the Big End Bearing or coank pin Assuming the bearing Pressure, Po = 12 N/mm2 Let, d. - Dia meter of the by end bearing LI - Length of the big end bearing = 1.3 d₁ (Assume $\frac{L_1}{d_1} = 1.3$) wad on the big end bearing, = projected asea x Beasing Pressure = d1 x L1 x Pb = d, x1.3d, x12 = 15.6 dig Equating the load on the big end boosing or coank pin to the maximum gas pressure, 15-6 dia = 25,447

$$F_{I} = 2 \times \left(\frac{2 \times 11 \times 2000}{60}\right)^{2} \times 0.070 \times \left[1 + \frac{0.070}{0.3}\right]$$

9

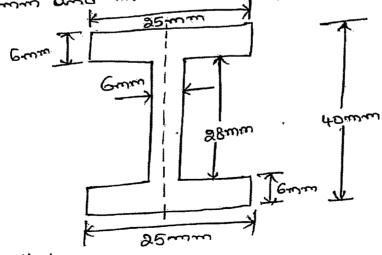
= 2 × 43864.908 × 0.070 × 1.233

= 7571.960 N

Equating the force on bolts to the meeting force,

The following section has been suggested for I-section connecting and of a petrol engine summing at door in the depth of the section is as mun, the width of the flange the depth of the section is as mun, the width of the flange and web 6 min. Is form and the thickness of the flange and web 6 min. If the stocke of the piston be 150 mm, determine the maximum of the stocke of the piston be 150 mm, determine the maximum of the stocke of the piston be length connecting stood be stress in the connecting rod if the length connecting stood be 300 mm. The density of the materials is 7800 kg/m³.

NOTE: In the given solution, the depth of the section is taken as 40mm and width of flange is 25mm.



Given that,

Length of connecting sod, l = 300 mmStroke length, 2r = 150 mmCrank viadius, $r = \frac{150}{R} = 75 \text{ mm}$

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```
da = 1631-218
         d = 40.388 mm 2 40.4mm
          L, = 1.3 d,
             = 1.3 × 40.4
             =52.520 mm
.3. Design of Small End of the connecting Rod/piston pin
    Let,
       Diameter of piston pin - da
  Length of the piston pin, Lo = 2da (Assume do = 2)
 Also, assuming that bearing pressure = 12 N/mm2
  road on the piston pin = dala x12
                          = d2 x2d2 x12 = 24d22
  Equating to the maximum force on the poston, we get,
             24 dg = 25447
                 da = 1060.292
                 d<sub>2</sub> = 32.562 ≈ 33~~~
                 L2 = 2d2
                    = 2 × 33 = 66 mm
4. Size of the Botts for securing Big End Cap
            dch - Core diameter of the bolts
             of - Allocophie tensile stress
                - 60 N/mrod (ASSume)
             My - rember of botts
    Force on bolts = I (Och) x & xnb
                     = II (dcb) 2x60x2 (Assume ng=2)
   Inestia force of the reciprocating parts,
           FI = MR COB XX [COS 0+ COS 20]
     Taking 0 = 0, We get,
```

Speed, N = 2000 8.p.m Depth of Section, h = 40 mm width of flonge, b = 25mm

Thickness of flange and web, t = 6mm

Angulan velocity,

$$\omega = \frac{311}{60} \text{ rod/s}$$

$$= 3 \times 11 \times 2000$$

Co = 209, 64 Food 5

Area of cross section,

$$= 2 \times (25 \times 6) + (28 \times 6)$$

= 468 mma

Modulus of Section,

$$Z = \frac{1}{12} \frac{(25 \times 40^3 - 19 \times 28^3)}{(\frac{40}{2})}$$

Z = 4928.8 mm3

Inertia load, fi on the connecting rad is given as

= 7800 x 468 x 10 6 x (209. 44) x 0.3 x 0.075

= 1801.409 N

Maximum bending moment,

$$= \frac{8\sqrt{3}}{87} \times 1.801 \times 10^{3} \times 300$$

Mmax = 69320.522 N-mm

These-fore, Maximum bending stress induced due to inestia bending force or Whipping 9tress, $\sigma_{\text{max}} = \frac{M_{\text{max}}}{7} = \frac{69320.522}{4928.8} = 14.064 \text{ N/mm}^{3}$

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-> The Connecting and of a slow speed diesel engine is 3 meters long and is made up of St 55 Determine the Scitable dimensions for the cioncular section of a good. The bone and Stroke of the Cylinder asie 900 mm and 1200 mm siespectively. The maximum Combustion pressure 4.8 Mpg. Determine the Whipping stresses if the engine oruns at 150 v.p.m Given that Length of Connecting sed, 1=3m

Glinder bore, D=900mm =0.9m Stroke length, 20 = 1200 mm = 1.2m Crank radius, r= 1200=600mm =0.6m Maximum Combustion Pressure, P= 4.8 MPa Emm/(1814) =

speed, -1=150 x.p.m Angulan velocity, co = atm rad sec

= 2.11 × 150

co = 15.708 and sec The load due to gos pressure,

Since the engine is summing at slow speed, the inextra force due to execiprocating parts may be neglected According to Euler's theory, the buckling load is given P = CITSEI

where C= I for both ends hinged

E - Youngs modulus for steel (Π) =0.21 ×106 for Steel (St-55) I - moment of Inestia = Tr xq 4mm 4 But, buckling load (Fb) = Fg x Fos Assume, factor of safety (F.9)=20. Wb = 3053628.059 x20 Wh = 61072561.18N Substituting (WB) value in Equation (1), We get, 61072561.18 = 1.172 x0.21 x106 x 9 d4 5.4965 × 1014 = TT2 × 0.21 × 106 × Ft d4 d4 = 5402531322 d = 271-112 = 280 mm .. The diameter of the Connecting and, d= 280mm check for whipping stress Area of Connecting sod, 0=1093 = 11 (280)2 = 61575.216 rom? Inertia load on the Connecting rood is given as, Fi = 6.0.2.000 (... T=1) Assume density, 0=7800 kg/m3 =7800 x 61575-216 x 10 x 0.6 x (15.708) 3x3

F: = 10655.889 N

:, Maximum bending moment, Mmax = 253 x F, x1 $=\frac{2\sqrt{3}}{27}$ × 106655.889 ×3000 Www = 41.025 × 106 10-mm Modulus of Section, $Z = \frac{11}{32} \times d^3$ $= \frac{1}{10} \times (380)^3$ 1 2 = 2155132.56 mm3 Whipping Stress, = Mmax =41.052×106 2155132.56 = 19.048 N/mm? .: whipping stress = 19:048 MB. The following data sietes to a 4-, Stroke cylinder vertical ഭഹ്വീറ piston diametes = 125mm Stroke = 150mm Speed of the engine = 1200 ripim speed of the seceprocating parts = 45N Design stress four the material used = 80 N/mm? Design a suitable connecting and, and check for stresses due to inestia. Given that, Piston diameter, D= 125mm stroke, 5 = 150 mm Speed, N = 1200 8.p.m Weight of reciprocating Parts = 45N Mass, m = 45/9.81 = 4.587 kg

Taking length of connecting and, L = 300 mm = 0.3 m Design of connecting Rod Area of cross-section, A =11t2

force on connecting rod due to gas pressure,

$$F = \frac{\prod_{4} (D)^{2} \cdot p}{4}$$
$$= \frac{\prod_{4} (J \approx 5)^{2} \times 5}{4}$$

(: Assume maximum gas pressure, P=5 N/mm3)

f=61359,232N

Radius of gyration of the Section,

$$k_{xx} = \sqrt{\frac{I_{xx}}{A}} = \sqrt{\frac{419t^4}{12}} \times \frac{1}{11t^2}$$

Length of Crank,

from Rankine's formula,

Buckling bad,

$$H_{B} = \frac{\sigma_{c} \cdot A}{1 + a \left(\frac{L}{k_{xx}}\right)^{2}}$$

Buckling load,

$$306796 \cdot 16 = \frac{600 \times 114^{2}}{1 + 1}$$

$$306796.16 = \frac{6600t^{2}}{1+3.779}$$

```
306796;16 (t2+3,779) = 6600t4
 16600+4-306796.16+2-1159382.689=0
  By solving the above equation, we get,
          t2 = 49.998
           t= 9.071 mm 2 7mm (Egy)
  The Flage and web thickness of Section is 7mm
   Width of Section,
            B=4t=4x7
            B = 28 mm
   Depth or height of section,
             H = 5t = 5x7
             H = 35~~
   Depth near the big end of CR;
              H_1 = 1.2 + 1.2 \times 35
              H1 = 42 mm
   Depth mean the Small end of GR,
           Ha = 0.85H = 0.85 x 35
   The dimensions means small end = 42 mm x29.75mm
Design of Crankpin & piston pin at Big End of GR
  Load on comprim = projected Area x Boaring Pressure
                    = dc · lc · Pbc ( · Take lc = 1.5dc)
                    = dc . (1.5dc) . Pbc =1.5dc2. Pbc
    Force on piston = Load on crankpin
               : F = 1.5dc2 x 12 (: Take Poc = 12N/mm)
        61359·232 = 18dc2
             dc = 58.385 & 59 mm (say)
              lc=1.5dc = 1.5 ×59 = 88.5 mm
        At big-end of CR,
```

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```
1 29 Diameter of Grankpin, dc = 59mm
Length of Grankpin, lc = 88.5mm
      Design of Croonkpin and piston pin at Small End of C.R
                  F=dc, xlc, xPbp
                61359.232 = dc, xadc, x15 (: Take Pop = 15 N/mm?)
                 61359-232 =30dc2
                    da = 45.225 & 46 mm (say)
                     lc, = 2x46 = 92 mm
                 At Small-end of C.P
               Diameter of Coankpin, da = 46mm
               length of Crankpin, la = 92 mm
      Design of Bolts
             Let, do = Dia-meter of bolts
                  q = permissible tensile stress too bott material.
     force on botts = Area of botts x Tensile stress x Number of
                      bolts.
                 Fb = 1 (d) x of xm (: Take of = 80 N/mma)
                    = 1 (9) x80 x 3
                Th = 125.664 dha
        Inertia force of succiprocating Parts,
                   F1 = M. Co28 [Co0+ Co020]
             Angular velocity,
                   \omega = \frac{2\pi N}{60} = \frac{2x\pi x 1200}{60}
                 60 = 125.664 rad/sec
      Fg = 4.587 x (25.664) x0.075 (cos o + (05.2x0))
```

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 $=5432.65\left[1+\frac{1}{4}\right]$ Fr = 5432.65 x1.25 For = 6790.813N Equating inestia force and force on bolts, ice, For = FL 6790.813 = 125.664 dr db = \(\frac{6196.813}{125.001} dr = 7.351 28mm Nomical diameter of both, do = db = 8 = 9.586 mm & lomm (say) Design of Cap at Big End of CoR Let, to = Thickness of big-end cap by = Width of big - and cap Maximum bending moments Mc = FIXX x = distance between the botts. X = dc+(2xthickness of booming lines) + Norming I diameter of both + clear ornce = 59 *(2×3)+10*3 X = 78 mm .. Mc = FT xx = 6790.813 x78 = 88280.569 N-mm section modelus for cap, $7c = \frac{bc \cdot (t_c)^2}{88.5 + t_c^2} = 88.5 + t_c^2$ (: bc = 1c) Zc = 14.75 tc8 .: Bending stress,

$$G_b = \frac{Mc}{7c} = \frac{88280.569}{14.75tc^2}$$
 (** Take $G_b = 120 \text{ N/mm}^3$)

$$tc^2 = \frac{88280.569}{(4.75 \times 120)}$$

Now, checking for induced bending stresses due to inester bending forces on CoR,

.. Mass of Connecting per meter length

Maximum bending moment,

Section modulus of Cop,

$$L_{xx} = \frac{T_{xx}}{\frac{5t}{2}} = \frac{419t^{4}}{12} \times \frac{2}{5t} = 13.967t^{3}$$

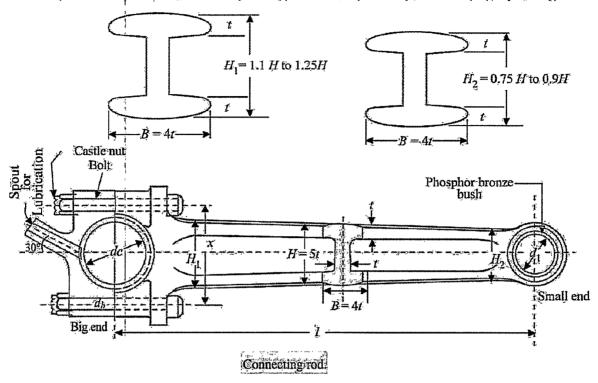
.. Maximum induced bending stress,

$$O_{b} = \frac{M}{Z_{xx}} = \frac{8602.585}{4790.681} = 1.8 \text{ N/mm}^{2}$$

Since, the induced bending stress is less than the Permissible value, therefore, the design is Safe.

Connecting Rod

The connecting rod is the intermediate member between the piston and the crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crankpin and thus convert the reciprocating motion of the piston into the rotary motion of the crank. The usual form of the connecting rod in internal combustion engines is shown in Fig. 32.9. It consists of a long shank, a small end and a big end. The cross-section of the shank may be rectangular, circular, tubular, *I*-section or *H*-section. Generally circular section is used for low speed engines while *I*-section is preferred for high speed engines.



The *length of the connecting rod (l) depends upon the ratio of l/r, where r is the radius of crank. It may be noted that the smaller length will decrease the ratio l/r. This increases the angularity of the connecting rod which increases the side thrust of the piston against the cylinder liner which in turn increases the wear of the liner. The larger length of the connecting rod will increase the ratio l/r. This decreases the angularity of the connecting rod and thus decreases the side thrust and the resulting wear of the cylinder. But the larger length of the connecting rod increases the overall height of the engine. Hence, a compromise is made and the ratio l/r is generally kept as 4 to 5.

The small end of the connecting rod is usually made in the form of an eye and is provided with a bush of phosphor bronze. It is connected to the piston by means of a piston pin.

The big end of the connecting rod is usually made split (in two **halves) so that it can be mounted easily on the crankpin bearing shells. The split cap is fastened to the big end with two cap bolts. The bearing shells of the big end are made of steel, brass or bronze with a thin lining (about 0.75 mm) of white metal or babbit metal. The wear of the big end bearing is allowed for by inserting thin metallic strips (known as *shims*) about 0.04 mm thick between the cap and the fixed half of the connecting rod. As the wear takes place, one or more strips are removed and the bearing is trued up.

The connecting rods are usually manufactured by drop forging process and it should have adequate strength, stiffness and minimum weight. The material mostly used for connecting rods varies from mild carbon steels (having 0.35 to 0.45 percent carbon) to alloy steels (chrome-nickel or chrome-molybdenum steels). The carbon steel having 0.35 percent carbon has an ultimate tensile strength of about 650 MPa when properly heat treated and a carbon steel with 0.45 percent carbon has a ultimate tensile strength of 750 MPa. These steels are used for connecting rods of industrial engines. The alloy steels have an ultimate tensile strength of about 1050 MPa and are used for connecting rods of aeroengines and automobile engines.

The bearings at the two ends of the connecting rod are either splash lubricated or pressure lubricated. The big end bearing is usually splash lubricated while the small end bearing is pressure lubricated. In the splash lubrication system, the cap at the big end is provided with a dipper or spout and set at an angle in such a way that when the connecting rod moves downward, the spout will dip into the lubricating oil contained in the sump. The oil is forced up the spout and then to the big end bearing. Now when the connecting rod moves upward, a splash of oil is produced by the spout. This splashed up lubricant find its way into the small end bearing through the widely chamfered holes provided on the upper surface of the small end.

In the **pressure lubricating** system, the lubricating oil is fed under pressure to the big end bearing through the holes drilled in crankshaft, crankwebs and crank pin. From the big end bearing, the oil is fed to small end bearing through a fine hole drilled in the shank of the connecting rod. In some cases, the small end bearing is lubricated by the oil scrapped from the walls of the cyinder liner by the oil scraper rings.

Forces Acting on the Connecting Rod

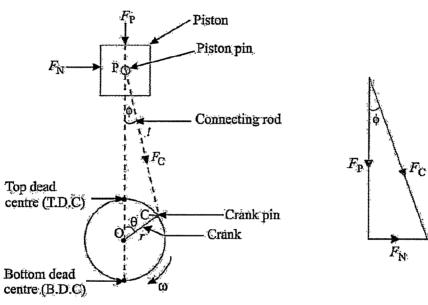
The various forces acting on the connecting rod are as follows:

- 1. Force on the piston due to gas pressure and inertia of the reciprocating parts,
- 2. Force due to inertia of the connecting rod or inertia bending forces,
- 3. Force due to friction of the piston rings and of the piston, and
- 4. Force due to friction of the piston pin bearing and the crankpin bearing.

We shall now derive the expressions for the forces acting on a vertical engine, as discussed below.

1. Force on the piston due to gas pressure and inertia of reciprocating parts

Consider a connecting rod PC as shown in Fig. 32.10.



Forces on the connecting rod.

Let p = Maximum pressure of gas,

D = Diameter of piston,

$$A = \text{Cross-section area of piston} = \frac{\pi D^2}{4}$$

 $m_{\rm R}$ = Mass of reciprocating parts,

$$\Rightarrow \text{ Acceleration of reciprocating}$$

$$\text{parts} = \omega^2 \cdot r \left(\cos \theta + \frac{\cos 2\theta}{n} \right)$$

= Mass of piston, gudgeon pin etc. $+\frac{1}{3}$ rd mass of connecting rod,

 ω = Angular speed of crank,

 ϕ = Angle of inclination of the connecting rod with the line of stroke,

 θ = Angle of inclination of the crank from top dead centre,

r = Radius of crank,

I =Length of connecting rod, and

n = Ratio of length of connecting rod to radius of crank = 1/r.

We know that the force on the piston due to pressure of gas,

$$F_{\rm L} = \text{Pressure} \times \text{Area} = p \cdot A = p \times \pi D^2/4$$

and inertia force of reciprocating parts,

$$F_{\rm L} = {
m Mass} \times {
m *Acceleration} = m_{\rm R} \cdot \omega^2 \cdot r \left(\cos \theta + \frac{\cos 2\theta}{n} \right)$$

It may be noted that the inertia force of reciprocating parts opposes the force on the piston when it moves during its downward stroke (i. e. when the piston moves from the top dead centre to bottom dead centre). On the other hand, the inertia force of the reciprocating parts helps the force on the piston when it moves from the bottom dead centre to top dead centre.

.. Net force acting on the piston or piston pin (or gudgeon pin or wrist pin),

$$F_{\rm p}$$
 = Force due to gas pressure \mp Inertia force = $F_{\rm L} \mp F_{\rm I}$

The -ve sign is used when piston moves from TDC to BDC and +ve sign is used when piston moves from BDC to TDC.

When weight of the reciprocating parts $(W_R = m_R \cdot g)$ is to be taken into consideration, then $F_P = F_L \mp F_1 \pm W_R$

The force $F_{\rm p}$ gives rise to a force $F_{\rm C}$ in the connecting rod and a thrust $F_{\rm N}$ on the sides of the cylinder walls. From Fig. 32.10, we see that force in the connecting rod at any instant,

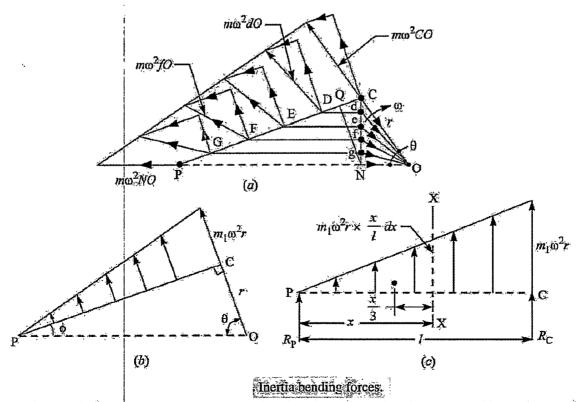
$$F_{\rm C} = \frac{F_{\rm P}}{\cos \phi} = \frac{F_{\rm P}}{\sqrt{1 - \frac{\sin^2 \theta}{n^2}}}$$

The force in the connecting rod will be maximum when the crank and the connecting rod are perpendicular to each other (i.e. when $\theta = 90^{\circ}$). But at this position, the gas pressure would be decreased considerably. Thus, for all practical purposes, the force in the connecting rod $(F_{\rm C})$ is taken equal to the maximum force on the piston due to pressure of gas $(F_{\rm L})$, neglecting piston inertia effects.

2. Force due to inertia of the connecting rod or inertia bending forces

Consider a connecting rod PC and a crank OC rotating with uniform angular velocity w rad f s. In order to find the accleration of various points on the connecting rod, draw the Klien's acceleration diagram CQNO as shown in Fig. 32.11 (a). CO represents the acceleration of C towards O and NO represents the accleration of P towards O. The acceleration of other points such as D, E, F and G etc., on the connecting rod PC may be found by drawing horizontal lines from these points to intresect CN at G, G, and G respectively. Now G, G and G respectively. Now G, G and G respectively. The inertial force acting on each point will be as follows:

Inertia force at $C = m \times \omega^2 \times CO$ Inertia force at $D = m \times \omega^2 \times dO$ Inertia force at $E = m \times \omega^2 \times eO$, and so on.



The inertia forces will be opposite to the direction of acceleration or centrifugal forces. The inertia forces can be resolved into two components, one parallel to the connecting rod and the other perpendicular to rod. The parallel (or longitudinal) components adds up algebraically to the force

acting on the connecting rod (F_C) and produces thrust on the pins. The perpendicular (or transverse) components produces bending action (also called whipping action) and the stress induced in the connecting rod is called whipping stress.

It may be noted that the perpendicular components will be maximum, when the crank and connecting rod are at right angles to each other.

The variation of the inertia force on the connecting rod is linear and is like a simply supported beam of variable loading as shown in Fig. 32.11 (b) and (c). Assuming that the connecting rod is of uniform cross-section and has mass m_1 kg per unit length, therefore,

Inertia force per unit length at the crankpin

$$= m_1 \times \omega^2 r$$

and inertia force per unit length at the piston pin

Inertia force due to small element of length dx at a distance x from the piston pin P,

$$dF_1 = m_1 \times \omega^2 r \times \frac{x}{L} \times dx$$

.. Resultant inertia force,

$$F_{J} = \int_{0}^{l} m_{l} \times \omega^{2} r \times \frac{x}{l} \times dx = \frac{m_{l} \times \omega^{2} r}{l} \left[\frac{x^{2}}{2} \right]_{0}^{l}$$

$$= \frac{m_{l} l}{2} \times \omega^{2} r = \frac{m}{2} \times \omega^{2} r \qquad \qquad \text{...(Substituting } m_{l} \cdot l = m)$$

This resultant inertia force acts at a distance of 2l/3 from the piston pin P.

Since it has been assumed that $\frac{1}{3}$ rd mass of the connecting rod is concentrated at piston pin P

(i.e. small end of connecting rod) and $\frac{2}{3}$ rd at the crankpin (i.e. big end of connecting rod), therefore, the reaction at these two ends will be in the same proportion. i.e.

$$R_{\rm p} = \frac{1}{3} F_{\rm I}$$
, and $R_{\rm C} = \frac{2}{3} F_{\rm I}$

Now the bending moment acting on the rod at section X-X at a distance x from P_i

$$M_{X} = R_{P} \times x - {}^{*}m_{1} \times \omega^{2}r \times \frac{x}{I} \stackrel{?}{\otimes} \frac{1}{2}x \times \frac{x}{3}$$
$$= \frac{1}{3} F_{I} \times x - \frac{m_{1}J}{2} \times \omega^{2}r \times \frac{x^{3}}{3I^{2}}$$

...(Multiplying and dividing the latter expression by I)

$$= \frac{F_1 \times x}{3} - F_1 \times \frac{x^3}{3t^2} = \frac{F_1}{3} \left(x - \frac{x^3}{t^2} \right) \qquad ...(1)$$

For maximum bending moment, differentiate M_{χ} with respect to x and equate to zero, i.e.

$$\frac{d_{\text{MX}}}{dx} = 0 \text{ or } \frac{F_{\text{I}}}{3} \left[1 - \frac{3x^2}{l^2} \right] = 0$$

$$\therefore 1 - \frac{3x^2}{l^2} = 0 \text{ or } 3x^2 = l^2 \text{ or } x = \frac{l}{\sqrt{3}}$$

Maximum bending moment,

$$M_{max} = \frac{F_{\rm I}}{3} \left[\frac{l}{\sqrt{3}} - \frac{\left(\frac{l}{\sqrt{3}}\right)^3}{l^2} \right] \qquad ... [From equation (i)]$$

$$= \frac{F_{\rm I}}{3} \left[\frac{l}{\sqrt{3}} - \frac{l}{3\sqrt{3}} \right] = \frac{F_{\rm I} \times l}{3\sqrt{3}} \times \frac{2}{3} = \frac{2F_{\rm I} \times l}{9\sqrt{3}}$$

$$= 2 \times \frac{m}{2} \times \omega^2 r \times \frac{l}{9\sqrt{3}} = m \times \omega^2 r \times \frac{l}{9\sqrt{3}}$$

and the maximum bending stress, due to inertia of the connecting rod,

$$\sigma_{max} = \frac{M_{max}}{Z}$$
* B.M. due to variable force from $\left(0 \text{ to } m_1 \omega^2 r \times \frac{x}{l}\right)$ is equal to the area of triangle multiplied by the distance of C.G. from $X - X\left(i.e.\frac{x}{3}\right)$.

From above we see that the maximum bending moment varies as the square of speed, therefore, the bending stress due to high speed will be dangerous. It may be noted that the maximum axial force and the maximum bending stress do not occur simultaneously. In an I.C. engine, the maximum gas load occurs close to top dead centre whereas the maximum bending stress occurs when the crank angle $\theta = 65^{\circ}$ to 70° from top dead centre. The pressure of gas falls suddenly as the piston moves from dead centre. Thus the general practice is to design a connecting rod by assuming the force in the connecting rod (F_{r}) equal to the maximum force due to pressure (F_{τ}) , neglecting piston inertia effects and then checked for bending stress due to inertia force (i.e. whipping stress).

3. Force due to friction of piston rings and of the piston

The frictional force (F) of the piston rings may be determined by using the following expression

 $F = \pi D \cdot t_{p} \cdot n_{p} \cdot p_{p} \cdot \mu$

D = Cylinder bore.

 $t_{\rm p} = Axial$ width of rings,

 $n_{\rm R}$ = Number of rings,

 $p_{\rm p}$ = Pressure of rings (0.025 to 0.04 N/mm²), and

 μ = Coefficient of friction (about 0.1).

Since the frictional force of the piston rings is usually very small, therefore, if may be neglected.

The friction of the piston is produced by the normal component of the piston pressure which varies from 3 to 10 percent of the piston pressure. If the coefficient of friction is about 0.05 to 0.06, then the frictional force due to piston will be about 0.5 to 0.6 of the piston pressure, which is very low. Thus, the frictional force due to piston is also neglected.

4. Force due to friction of the piston pin bearing and crankpin bearing

The force due to friction of the piston pin bearing and crankpin bearing, is to bend the connecting rod and to increase the compressive stress on the connecting rod due to the direct load. Thus, the maximum compressive stress in the connecting rod will be

 $\sigma_{c'(max)}$ = Direct compressive stress + Maximum bending or whipping stress due to inertia bending stress

Design of Connecting Rod

In designing a connecting rod, the following dimensions are required to be determined:

- 1. Dimensions of cross-section of the connecting rod,
- 2. Dimensions of the crankpin at the big end and the piston pin at the small end,
- 3. Size of bolts for securing the big end cap, and
- 4. Thickness of the big end cap.

Let

The procedure adopted in determining the above mentioned dimensions is discussed as below:

1. Dimensions of cross-section of the connecting rod

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile forces, therefore, the cross-section of the connecting rod is designed as a strut and the Rankine's formula is used.

A connecting rod, as shown in Fig. 32.12, subjected to an axial load W may buckle with X-axis as neutral axis (i.e. in the plane of motion of the connecting rod) or Y-axis as neutral axis (i.e. in the plane perpendicular to the plane of motion). The connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis.

A connecting rod should be equally strong in buckling about both the axes.

A =Cross-sectional area of the connecting rod.

l =Length of the connecting rod,

 $\sigma_c = \text{Compressive yield stress}$,

 $W_{\rm R} = \text{Buckling load},$

 I_{yy} and I_{yy} = Moment of inertia of the section about X-axis and Y-axis respectively, and

 k_{xx} and k_{yx} = Radius of gyration of the section about X-axis and Y-axis respectively.

where

ř

According to Rankine's formula.

$$W_{\rm B} \text{ about } X - \text{axis} = \frac{\sigma_c.A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{\sigma_c.A}{1 + a\left(\frac{l}{k_{xx}}\right)^2} \qquad \dots (\because \text{ For both ends hinged, } L = l)$$

$$W_{\rm B} \text{ about } Y - \text{axis} = \frac{\sigma_c.A}{1 + a\left(\frac{L}{k_{yy}}\right)^2} = \frac{\sigma_c.A}{1 + a\left(\frac{l}{2k_{yy}}\right)^2} \qquad \dots [\because \text{ For both ends fixed, } L = \frac{l}{2}]$$

where

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and

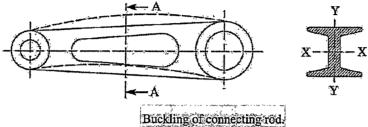
L = Equivalent length of the connecting rod, and

a = Constant

= 1 / 7500, for mild steel

= 1/9000, for wrought iron

= 1/1600, for cast iron



In order to thave a connecting rod equally strong in buckling about both the axes, the buckling loads must be equal, i.e.

$$\frac{\sigma_c \cdot A}{1 + a \left(\frac{L}{k_{xx}}\right)^2} = \frac{\sigma_c \cdot A}{1 + a \left(\frac{I}{2k_{yy}}\right)^2} \quad \text{or} \quad \left(\frac{I}{k_{xx}}\right)^2 = \left(\frac{I}{2k_{yy}}\right)^2$$

$$k_{xx}^2 = 4k_{yy}^2 \quad \text{or} \quad I_{xx} = 4I_{yy} \qquad \dots (\because I = A \cdot k^2)$$

This shows that the connecting rod is four times strong in buckling about Y-axis than about X-axis. If $I_{xx} > 4I_{yy}$ then buckling will occur about Y-axis and if $I_{xx} < 4 I_{yy}$ buckling will occur about X-axis. In actual practice, $I_{\rm co}$ is kept slightly less than $4I_{\rm co}$. It is usually taken between 3 and 3.5 and the connecting rod is designed for bucking about X-axis. The design will always be satisfactory for buckling about Y-axis.

The most suitable section for the connecting rod is I-section with the proportions as shown in Fig. 32.13 (a).

Let thickness of the flange and web of the section = t

Width of the section, $B=4\tau$ and depth or height of the section,

$$H = 5t$$

From Fig. 32.13 (a), we find that area of the section,

$$A = 2(4t \times t) + 3t \times t = 11t^2$$

Moment of inertia of the section about X-axis,

$$I_{xt} = \frac{1}{12} \left[4 t' (5t)^3 - 3t (3t)^3 \right] = \frac{419}{12} t^4$$

and moment of inertia of the section about Y-axis,

$$I_{yy} = \left[2 \times \frac{1}{12} t \times (4t)^3 + \frac{1}{12} (3t) t^3\right] = \frac{131}{12} t^4$$

$$\frac{I_{xx}}{I_{yy}} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

-X/3t H=5t

(b)

I-section of connecting rod.

-B=4t

(a)

(d) Design of right hand crank web

The right hand crank web is subjected to the following stresses:

- (i) Bending stresses in two planes normal to each other, due to the radial and tangential components of F_0 ,
- (ii) Direct compressive stress due to $F_{\rm R}$, and
- (iii) Torsional stress.

The bending moment due to the radial component of F_0 is given by

$$M_{\rm R} = H_{\rm R2} \left(b_{\rm I} - \frac{l_c}{2} - \frac{t}{2} \right)$$
 ... (2)

We also know that
$$M_R = \sigma_{bR} \times Z = \sigma_{bR} \times \frac{1}{6} \times w.t^2$$
 ... (ii)

where

 σ_{bR} = Bending stress in the radial direction, and

$$Z = Section modulus = \frac{1}{6} \times w \cdot t^2$$

From equations (i) and (ii), the value of bending stress σ_{bR} is determined.

The bending moment due to the tangential component of F_{O} is maximum at the juncture of crank and shaft. It is given by

$$M_{\rm T} = F_{\rm T} \left[r - \frac{d_{s1}}{2} \right] \qquad \dots (iii)$$

where

 d_{s1} = Shaft diameter at juncture of right hand crank arm, i.e. at bearing 2.

We also know that $M_T = \sigma_{bT} \times Z = \sigma_{bT} \times \frac{1}{6} \times t \cdot w^2$... (iv) σ_{kT} = Bending stress in tangential direction. where

From equations (iii) and (iv), the value of bending stress σ_{kT} is determined.

The direct compressive stress is given by,

$$\sigma_d = \frac{F_R}{2w.t}$$

The maximum compressive stress (σ) will occur at the upper left corner of the cross-section of the crank.

$$\sigma_{c} = \sigma_{bR} + \sigma_{bT} + \sigma_{il}$$
Now, the twisting moment on the arm,

$$T = H_{T1} \left(b_2 + \frac{l_c}{2} \right) - F_T \times \frac{l_c}{2} = H_{T2} \left(b_1 - \frac{l_c}{2} \right)$$

We know that shear stress on the arm

$$\tau = \frac{T}{Z_{\rm R}} = \frac{4.5 \, T}{w.t^2}$$

where

$$Z_{\rm P} = \text{Polar section modulus} = \frac{w.t^2}{4.5}$$

.. Maximum or total combined stress,

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

The value of $(\sigma)_{max}$ should be within safe limits. If it exceeds the safe value, then the dimension w may be increased because it does not affect other dimensions.

(e) Design of left hand crank web

Since the left hand crank web is not stressed to the extent as the right hand crank web, therefore, the dimensions for the left hand crank web may be made same as for right hand crank web.

(f) Design of crankshaft bearings

The bearing 2 is the most heavily loaded and should be checked for the safe bearing pressure.

We know that the total reaction at the bearing 2,

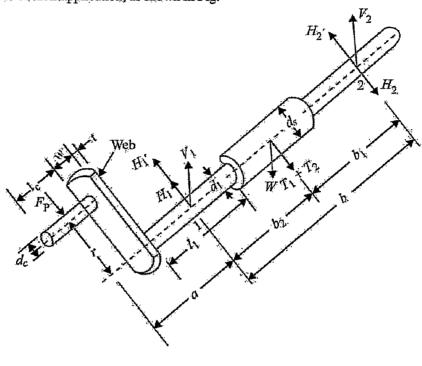
$$R_2 = \frac{F_{\rm P}}{2} + \frac{W}{2} + \frac{T_1 + T_2}{2}$$

 \therefore Total bearing pressure = $\frac{R_2}{l_1 d_2}$ where l_2 = Length of bearing 2.

Side or Overhung Crankshaff

The side or overhung crankshafts are used for medium size and large horizontal engines. Its main advantage is that it requires only two bearings in either the single or two crank construction. The design procedure for the side or overhung crankshaft is same as that for centre crankshaft. Let us now design the side crankshaft by considering the two crank positions, *i.e.* when the crank is at dead centre (or when the crankshaft is subjected to maximum bending moment) and when the crank is at an angle at which the twisting moment is maximum. These two cases are discussed in detail as below:

1. When the crank is at dead centre. Consider a side crankshaft at dead centre with its loads and distances of their application, as shown in Fig.



Side crankshaft at dead centre

Let

D = Piston diameter or cylinder bore in mm.

p = Maximum intensity of pressure on the piston in N/mm²,

W =Weight of the flywheel acting downwards in N, and

 $T_1 + T_2 =$ Resultant belt tension or pull acting horizontally in N.

We know that gas load on the piston,

$$F_{\mathbf{p}} = \frac{\pi}{4} \times D^2 \times p$$

Due to this piston gas load (F_p) acting horizontally, there will be two horizontal reactions H_1 and H_2 at bearings 1 and 2 respectively, such that

$$H_1 = \frac{F_P(a+b)}{b}$$
; and $H_2 = \frac{F_P \times a}{b}$

Due to the weight of the flywheel (W) acting downwards, there will be two vertical reactions V_1 and V_2 at bearings 1 and 2 respectively, such that

$$V_1 = \frac{W.b_1}{h}$$
; and $V_2 = \frac{W.b_2}{h}$

Now due to the resultant belt tension (T_1+T_2) acting horizontally, there will be two horizontal reactions H_1 and H_2 at bearings 1 and 2 respectively, such that

$$H_1'' = \frac{(T_1 + T_2)b_1}{b}$$
; and $H_2' = \frac{(T_1 + T_2)b_2}{b}$

The various parts of the side crankshaft, when the crank is at dead centre, are now designed as discussed below:

(a) Design of crankpin. The dimensions of the crankpin are obtained by considering the crankpin in bearing and then checked for bending stress.

Let

 $d_c = \text{Diameter of the crankpin in mm},$

 $l_c = \text{Length of the crankpin in mm, and}$

 $p_b = \text{Safe bearing pressure on the pin in N/mm}^2$. It may be between 9.8 to 12.6 N/mm².

We know that $F_{\rm p} = d_{\rm c} \cdot l_{\rm c} \cdot p_{\rm b}$

From this expression, the values of d_c and l_c may be obtained. The length of crankpin is usually from 0.6 to 1.5 times the diameter of pin. The crankpin is now checked for bending stress. If it is assumed that the crankpin acts as a cantilever and the load on the crankpin is uniformly distributed, then maximum bending moment will be $\frac{F_{\rm p} \times l_{\rm c}}{2}$. But in actual practice, the bearing pressure on the crankpin is not uniformly distributed and may, therefore, give a greater value of bending moment ranging between $\frac{F_P \times l_c}{2}$ and $F_P \times l_c$. So, a mean value of bending moment, i.e. $\frac{3}{4} F_P \times l_a$ may be assumed.

... Maximum bending moment at the crankpin,

$$M = \frac{3}{4} F_{\rm P} \times I_{\rm c} \qquad \dots$$

... (Neglecting pin collar thickness)

Section modulus for the crankpin,

$$Z = \frac{\pi}{32} (d_c)^3$$

.. Bending stress induced.

$$\sigma_h = M/Z$$

This induced bending stress should be within the permissible limits.

(b) Design of bearings. The bending moment at the centre of the bearing 1 is given by

$$M = F_p (0.75 L_c + t + 0.5 L_1)$$
 ...(1)

where

 $l_{x} =$ Length of the crankpin,

 $t = \text{Thickness of the crank web} = 0.45 d_c \text{ to } 0.75 d_{cd}$ and

$$l_1 = \text{Length of the bearing} = 1.5 d_c \text{ to } 2 d_c$$

We also know that

$$M = \frac{\pi}{32} (d_1)^3 \sigma_b \qquad ...(ii)$$

From equations (i) and (ii), the diameter of the bearing 1 may be determined.

Note: The bearing 2 is also made of the same diameter. The length of the bearings are found on the basis of allowable bearing pressures and the maximum reactions at the bearings.

(c) Design of crank web. When the crank is at dead centre, the crank web is subjected to a bending moment and to a direct compressive stress.

We know that bending moment on the crank web,

$$M = F_{\rm p} (0.75 l_{\rm e} + 0.5 t)$$

and

section modulus,
$$Z = \frac{1}{6} \times w \cdot t^2$$

... Bending stress, $\sigma_h = \frac{M}{Z}$

$$\sigma_b = \frac{M}{Z}$$

We also know that direct compressive stress,

$$\sigma_d = \frac{F_p}{w \cdot t}$$

 \therefore Total stress on the crank web $\sigma_{\rm m} = \sigma_{\rm c}$

$$\sigma_{T} = \sigma_{b} + \sigma_{d}$$

This total stress should be less than the permissible limits.

(d) Design of shaft under the flywheel. The total bending moment at the flywheel location will be the resultant of horizontal bending moment due to the gas load and belt pull and the vertical bending moment due to the flywheel weight.

 $d_{\rm x}$ = Diameter of shaft under the flywheel.

We know that horizontal bending moment at the flywheel location due to piston gas load,

$$M_1 = F_p (a + b_2) - H_1 \cdot b_2 = H_2 \cdot b_1$$

and horizontal bending moment at the flywheel location due to belt pull.

$$M_2 = H_1' b_2 = H_2' b_1 = \frac{(T_1' + T_2) b_1 b_2}{h}$$

.. Total horizontal bending moment,

$$M_{\rm H}=M_1+M_2$$

We know that vertical bending moment due to flywheel weight.

$$M_{V} = V_{1}.b_{2} = V_{2}.b_{1} = \frac{Wb_{1}b_{2}}{b}$$

.. Resultant bending moment,

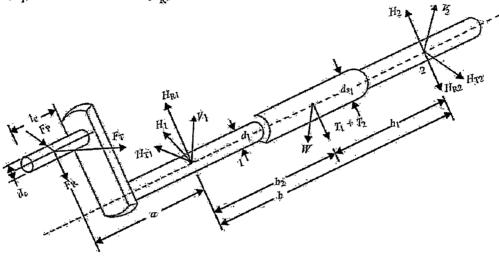
$$M_{\rm R} = \sqrt{(M_{\rm H})^2 + (M_{\rm V})^2}$$
 ...(i)

We also know that

$$M_R = \frac{\pi}{32} (d_s)^3 \sigma_b \qquad ...(ii)$$

From equations (i) and (ii), the diameter of shaft (d_i) may determined.

2. When the crank is at an angle of maximum twisting moment. Consider a position of the crank at an angle of maximum twisting moment as shown in Fig. 32.19. We have already discussed in the design of a centre crankshaft that the thrust in the connecting rod (F_O) gives rise to the tangential force (F_p) and the radial force (F_p) .



Crank at an angle of maximum twisting moment

Due to the tangential force (F_T) , there will be two reactions at the bearings 1 and 2, such that

$$H_{\mathrm{Tl}} = \frac{F_{\mathrm{T}}(a+b)}{b}$$
; and $H_{\mathrm{T2}} = \frac{F_{\mathrm{T}} \times a}{b}$

Due to the radial force (F_p) , there will be two reactions at the bearings 1 and 2, such that

$$H_{R1} = \frac{F_R(a+b)}{b}$$
; and $H_{R2} = \frac{F_R \times a}{b}$

The reactions at the bearings 1 and 2 due to the flywheel weight (W) and resultant belt pull $(T_1 + T_2)$ will be same as discussed earlier.

Now the various parts of the crankshaft are designed as discussed below:

- (a) Design of crank web. The most critical section is where the web joins the shaft. This section is subjected to the following stresses:
 - (i) Bending stress due to the tangential force F_{τ} ;
 - (ii) Bending stress due to the radial force $F_{\rm R}$;
 - (iii) Direct compressive stress due to the radial force $F_{\rm R}$; and
 - (iv) Shear stress due to the twisting moment of $F_{\rm T}$.

We know that bending moment due to the tangential force,

$$M_{\rm BT} = F_{\rm T} \bigg(r - \frac{d_1}{2} \bigg)$$
 where $d_1 \doteq {\rm Diameter}$ of the bearing 1.

... Bending stress due to the tangential force,
$$\sigma_{BT} = \frac{M_{BT}}{Z} = \frac{6M_{BT}}{t.w^2} \qquad ...(\because Z = \frac{1}{6} \times t \cdot w^2) ...(i)$$
We know that handing respect the tangential force

We know that bending moment due to the radial force,

$$M_{bR} = F_R (0.75 l_c + 0.5 t)$$

 $M_{\rm bR} = F_{\rm R} (0.75 \, l_{\rm c} + 0.5 \, t)$:. Bending stress due to the radial force,

$$\sigma_{bR} = \frac{M_{bR}}{Z} = \frac{6M_{bR}}{Wt^2}$$
 ... (Here $Z = \frac{1}{6} \times w \cdot t^2$) ... (ii) We know that direct compressive stress,

$$\sigma_{d} = \frac{F_{R}}{w.t} \qquad ...(iii)$$

:. Total compressive stress,

$$\sigma_c = \sigma_{bT} + \sigma_{bR} + \sigma_d \qquad ...(iv)$$

We know that twisting moment due to the tangential force,

$$T = F_{\rm T}(0.75 \ l_c + 0.5 \ t)$$
 $T = 4.5T$

$$\therefore \text{ Shear stress,} \qquad \tau = \frac{T}{Z_P} = \frac{4.5T}{w.t^2}$$

$$Z_{\rm p}$$
 = Polar section modulus = $\frac{w.t^2}{4.5}$

Now the total or maximum stress is given by

$$\sigma_{max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2} \qquad \dots (v)$$

This total maximum stress should be less than the maximum allowable stress.

(b) Design of shaft at the junction of crank

Let

 d_{v1} = Diameter of the shaft at the junction of the crank.

We know that bending moment at the junction of the crank,

$$M = F_{Q} (0.75l_{c} + t)$$

and twisting moment on the shaft

$$\hat{T} = \vec{F}_{rr} \times r$$

.. Equivalent twisting moment,

$$T_{e} = \sqrt{M^2 + T^2} \tag{1}$$

We also know that equivalent twisting moment,

$$T_e = \frac{\pi}{16} (d_{s1})^3 \tau$$
 ...(ii)

From equations (i) and (ii), the diameter of the shaft at the junction of the crank (d_{ct}) may be determined.

(c) Design of shaft under the flywheel

 d_r = Diameter of shaft under the flywheel.

The resultant bending moment (M_R) acting on the shaft is obtained in the similar way as discussed for dead centre position.

We know that horizontal bending moment acting on the shaft due to piston gas load,

$$M_1 = F_{\rm P}(a+b_2) - \left[\sqrt{(H_{\rm R1})^2 + (H_{\rm T1})^2}\right]b_2$$

and horizontal bending moment at the flywheel location due to belt pull,

$$M_2 = H_1' \cdot b_2 = H_2' \cdot b_1 = \frac{(T_1 + T_2) \cdot b_1 \cdot b_2}{b}$$

... Total horizontal bending moment,

$$M_{\rm rr} = M_1 + M_2$$

 $M_{\rm H} = M_1 + M_2$ Vertical bending moment due to the flywheel weight,

$$M_{V} = V_{1} \cdot b_{2} = V_{2} \cdot b_{1} = \frac{\hat{W} \hat{b}_{1} \hat{b}_{2}}{h}$$

.. Resultant bending moment,

$$M_{\rm R} = \sqrt{(M_{\rm H})^2 + (M_{\rm V})^2}$$

We know that twisting moment on the shaft,

$$T = F_{\tau} \times r$$

.. Equivalent twisting moment,

$$T_e = \sqrt{(M_{\rm R})^{2} + T^2} \qquad ...(i)$$

We also know that equivalent twisting moment,

$$T_e = \frac{\pi}{16} (d_s)^3 \tau \qquad ...(ii)$$

From equations (i) and (ii), the diameter of shaft under the flywheel (d_i) may be determined.

Example 32.4. Design a plain carbon steel centre crankshaft for a single acting four stroke single cylinder engine for the following data:

Bore = 400 nm; Stroke = 600 mm; Engine speed = 200 rp.m.; Mean effective pressure = 0.5 N/mm^2 ; Maximum combustion pressure = 2.5 N/mm^2 ; Weight of flywheel used as a pulley = 50 kN; Total belt pull = 6.5 kN.

When the crank has turned through 35° from the top dead centre, the pressure on the piston is IN/min² and the torque on the crank is maximum. The ratio of the connecting rad length to the crank radius is 5. Assume any other data required for the design.

Solution. Given: D = 400 mm; L = 600 mm or r = 300 mm; $p_m = 0.5 \text{ N/mm}^2$; $p = 2.5 \text{ N/mm}^2$; W = 50 kN; $T_1 + T_2 = 6.5 \text{ kN}$; $\theta = 35^\circ$; $p' = 1 \text{ N/mm}^2$; $l \cdot l \cdot r = 5$

We shall design the crankshaft for the two positions of the crank, i.e. firstly when the crank is at the dead centre; and secondly when the crank is at an angle of maximum twisting moment.

1. Design of the crankshaft when the crank is at the dead centre

We know that the piston gas load,

$$F_{\rm P} = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (400)^2 2.5 = 314200 \,\text{N} = 314.2 \,\text{kN}$$

Assume that the distance (b) between the bearings 1 and 2 is equal to twice the piston diameter (D).

and

$$b_1 = b_2 = \frac{b}{2} = \frac{800}{2} = 400 \text{ mm}$$

We know that due to the piston gas load, there will be two horizontal reactions H_1 and H_2 at bearings 1 and 2 respectively, such that

$$H_1 = \frac{F_P \times b_1}{b} = \frac{314.2 \times 400}{800} = 157.1 \text{ kN}$$

$$H_2 = \frac{F_P \times b_2}{b} = \frac{314.2 \times 400}{800} = 157.1 \text{ kN}$$

and

Assume that the length of the main bearings to be equal, i.e., $c_1 = c_2 = c/2$. We know that due to the weight of the flywheel acting downwards, there will be two vertical reactions V_2 and V_3 at bearings 2 and 3 respectively, such that

$$V_2 = \frac{W \times c_1}{c} = \frac{W \times c/2}{c} = \frac{W}{2} = \frac{50}{2} = 25 \text{ kN}$$

$$V_3 = \frac{W \times c_2}{c} = \frac{W \times c/2}{c} = \frac{W}{2} = \frac{50}{2} = 25 \text{ kN}$$

and

Due to the resultant belt tension (T_1+T_2) acting horizontally, there will be two horizontal reactions H_2 and H_3 respectively, such that

$$H_{2}' = \frac{(T_{1} + T_{2}) c_{1}}{c} = \frac{(T_{1} + T_{2}) c/2}{c} = \frac{T_{1} + T_{2}}{2} = \frac{6.5}{2} = 3.25 \text{kN}$$

$$H_{3}' = \frac{(T_{1} + T_{2}) c_{2}}{c} = \frac{(T_{1} + T_{2}) c/2}{c} = \frac{T_{1} + T_{2}}{2} = \frac{6.5}{2} = 3.25 \text{kN}$$

and

Now the various parts of the crankshaft are designed as discussed below:

(a) Design of crankpin

Let

 $d_v = \text{Diameter of the crankpin in mm};$

 $I_c =$ Length of the crankpin in mm; and

 σ_p = Allowable bending stress for the crankpin. It may be assumed as 75 MPa of N/mm².

We know that the bending moment at the centre of the crankpin,

$$M_C = H_1 \cdot b_2 = 157.1 \times 400 = 62.840 \text{ kN-mm}$$
 ...(i)

We also know that

$$M_{\rm C} = \frac{\pi}{32} (d_c)^3 \sigma_b = \frac{\pi}{32} (d_c)^3 75 = 7.364 (d_c)^3 \text{ N-mm}$$

= 7.364 × 10⁻³ (d_c)³ kN-mm ...(ii)

Equating equations (i) and (ii), we have

$$(d_c)^3 = 62\,840 / 7.364 \times 10^{-3} = 8.53 \times 10^6$$

 $d_c = 204.35 \text{ say } 205 \text{ mm Ans.}$

or

We know that length of the crankpin.

$$I_c = \frac{F_{\rm p}}{d_{\rm g} \cdot p_b} = \frac{314.2 \times 10^3}{205 \times 10} = 153.3 \text{ say } 155 \text{ mm Ans.}$$
...(Taking $p_b = 10 \text{ N/mm}^2$)

(b) Design of left hand crank web

We know that thickness of the crank web,

$$t = 0.65 d_c + 6.35 \text{ mm} = 0.65 \times 205 + 6.35 = 139.6 \text{ say } 140 \text{ mm Ans.}$$

and width of the crank web, $w = 1.125 d_c + 12.7 \text{ mm}$

$$= 1.125 \times 205 + 12.7 = 243.3$$
 say 245 mm Ans.

We know that maximum bending moment on the crank web,

$$M = H_1 \left(b_2 - \frac{l_c}{2} - \frac{t}{2} \right)$$
= 157.1 \left(400 - \frac{155}{2} - \frac{140}{2} \right) = 39 668 kN-mm

Section modulus, $Z = \frac{1}{6} \times w.t^2 = \frac{1}{6} \times 245 (140)^2 = 800 \times 10^3 \text{ mm}^3$

$$\therefore$$
 Bending stress, $\sigma_b = \frac{M}{Z} = \frac{39.668}{800 \times 10^3} = 49.6 \times 10^{-3} \text{ kN/mm}^2 = 49.6 \text{ N/mm}^2$

We know that direct compressive stress on the crank web.

$$\sigma_c = \frac{H_1}{w.t} = \frac{157.1}{245 \times 140} = 4.58 \times 10^{-3} \text{ kN/mm}^2 = 4.58 \text{ N/mm}^2$$

... Total stress on the crank wel

$$= \sigma_h + \sigma_c = 49.6 + 4.58 = 54.18 \text{ N/mm}^2 \text{ or MPa}$$

Since the total stress on the crank web is less than the allowable bending stress of 75 MPa, therefore, the design of the left hand crank web is safe.

(c) Design of right hand crank web

From the balancing point of view, the dimensions of the right hand crank web (i.e. thickness and width) are made equal to the dimensions of the left hand crank web.

(d) Design of shaft under the flywheel

Let

 $d_s =$ Diameter of the shaft in mm.

Since the lengths of the main bearings are equal, therefore

$$l_1 = l_2 = l_3 = 2\left(\frac{b}{2} - \frac{l_c}{2} - t\right) = 2\left(400 - \frac{155}{2} - 140\right) = 365 \text{ mm}$$

Assuming width of the flywheel as 300 mm, we have

$$c = 365 + 300 = 665 \text{ mm}$$

Allowing space for gearing and clearance, let us take c= 800 mm.

$$c_1 = c_2 = \frac{c}{2} = \frac{800}{2} = 400 \,\text{mm}$$

We know that bending moment due to the weight of flywheel,

$$M_{\rm W} = V_3 - c_1 = 25 \times 400 = 10000 \text{ kN-mm} = 10 \times 10^6 \text{ N-mm}$$

and bending moment due to the belt pull,

$$M_{\rm T} = H_3' - c_1 = 3.25 \times 400 = 1300 \text{ kN-mm} = 1.3 \times 10^6 \text{ N-mm}$$

... Resultant bending moment on the shaft,

$$M_{\rm S} = \sqrt{(M_{\rm W})^2 + (M_{\rm T})^2} = \sqrt{(10 \times 10^6)^2 + (1.3 \times 10^6)^2}$$

= 10.08 × 10⁶ N-mm

We also know that bending moment on the shaft (M_s) .

$$10.08 \times 10^{6} = \frac{\pi}{32} (d_{s})^{3} \sigma_{b} = \frac{\pi}{32} (d_{s})^{3} 42 = 4.12 (d_{s})^{3}$$
$$(d_{s})^{3} = 10.08 \times 10^{6} / 4.12 = 2.45 \times 10^{6} \text{ or } d_{s} = 134.7 \text{ say } 135 \text{ mm Ans.}$$

2. Design of the crankshaft when the crank is at an angle of maximum twisting moment

We know that piston gas load,

. .

$$F_{\rm p} = \frac{\pi}{4} \times D^2 \times p' = \frac{\pi}{4} (400)^2 1 = 125.68 \,\mathrm{kN}$$

In order to find the thrust in the connecting rod (F_Q) , we should first find out the angle of inclination of the connecting rod with the line of stroke (i.e. angle ϕ). We know that

$$\sin \phi = \frac{\sin \theta}{l/r} = \frac{\sin 35^{\circ}}{5} = 0.1147$$

$$\phi = \sin^{-1}(0.1147) = 6.58^{\circ}$$

We know that thrust in the connecting rod,

$$F_{\rm Q} = \frac{F_{\rm P}}{\cos \phi} = \frac{125.68}{\cos 6.58^{\circ}} = \frac{125.68}{0.9934} = 126.5 \text{ kN}$$

Tangential force acting on the cranks

$$F_{\rm T} = F_{\rm O} \sin (\theta + \phi) = 126.5 \sin (35^{\circ} + 6.58^{\circ}) = 84 \text{ kN}$$

and radial force,

$$F_{\rm R} = F_{\rm O} \cos (\theta + \phi) = 126.5 \cos (35^{\circ} + 6.58^{\circ}) = 94.6 \text{ kN}$$

Due to the tangential force (F_{τ}) , there will be two reactions at bearings 1 and 2, such that

$$H_{\text{T1}} = \frac{F_{\text{T}} \times b_1}{b} = \frac{84 \times 400}{800} = 42 \text{ kN}$$
 $F_{\text{T}} \times b_2 = 84 \times 400$

and

$$H_{\rm T2} = \frac{F_{\rm T} \times h_2}{b} = \frac{84 \times 400}{800} = 42 \,\rm kN$$

Due to the radial force (F_n) , there will be two reactions at bearings 1 and 2, such that

$$H_{R1} = \frac{F_R \times b_I}{b} = \frac{94.6 \times 400}{800} = 47.3 \text{ kN}$$

$$H_{\rm R2}=\frac{F_{\rm R}\times b_2}{b}=\frac{94.6\times400}{800}=47.3~\rm kN$$
 Now the various parts of the crankshaft are designed as discussed below:

(a) Design of crankpin

Let

$$d_a = Diameter of crankpin in mm.$$

We know that the bending moment at the centre of the crankpin,

$$M_{\rm C} = H_{\rm R1} \times b_2 = 47.3 \times 400 = 18920 \text{ kN-mm}$$

and twisting moment on the crankpin,

$$T_{\rm c} = H_{\rm TL} \times r = 42 \times 300 = 12\,600$$
 kN-mm

.. Equivalent twisting moment on the crankpin,

$$T_c = \sqrt{(M_C)^2 + (T_C)^2} = \sqrt{(18.920)^2 + (12.600)^2}$$

= 22.740 kN-mm = 22.74 × 10⁶ N-mm

We know that equivalent twisting moment (T_n) ,

$$22.74 \times 10^6 = \frac{\pi}{16} (d_a)^3 \tau = \frac{\pi}{16} (d_a)^3 35 = 6.873 (d_a)^3$$

...(Taking $\tau = 35$ MPa or N/mm²)

$$(d_c)^3 = 22.74 \times 10^6 / 6.873 = 3.3 \times 10^6 \text{ or } d_c = 149 \text{ mm}$$

Since this value of crankpin diameter (i.e. $d_c = 149$ mm) is less than the already calculated value of $d_c = 205$ mm, therefore, we shall take $d_c = 205$ mm. Ans.

(b) Design of shaft under the flywheel

Let

42

Ξ.

$$d_s$$
 = Diameter of the shaft in mm.

The resulting bending moment on the shaft will be same as calculated earlier, i.e.

$$M_{\rm s} = 10.08 \times 10^6 \, \text{N-mm}$$

and twisting moment on the shaft,

$$T_{\rm S} = F_{\rm T} \times r = 84 \times 300 = 25\ 200\ {\rm kN-mm} = 25.2 \times 10^6\ {\rm N-mm}$$

. Equivalent twisting moment on shaft.

$$T_e = \sqrt{(M_S)^2 + (T_S)^2}$$

= $\sqrt{(10.08 \times 10^6)^2 + (25.2 \times 10^6)^2} = 27.14 \times 10^6 \text{ N-mm}$

We know that equivalent twisting moment (T_a) ,

$$27.14 \times 10^{6} = \frac{\pi}{16} (d_{s})^{3} \tau = \frac{\pi}{16} (135)^{3} \tau = 483156 \tau$$
$$\tau = 27.14 \times 10^{6} / 483156 = 56.17 \text{ N/mm}^{2}$$

From above, we see that by taking the already calculated value of ds = 135 mm, the induced shear stress is more than the allowable shear stress of 31 to 42 MPa. Hence, the value of d_s is calculated by taking $\tau = 35$ MPa or N/mm² in the above equation, i.e.

$$27.14 \times 10^6 = \frac{\pi}{16} (d_x)^3 35 = 6.873 (d_x)^3$$

 $(d_x)^3 = 27.14 \times 10^6 / 6.873 = 3.95 \times 10^6 \text{ or } d_x = 158 \text{ say } 160 \text{ mm. Airs.}$

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(c) Design of shaft at the functure of right hand crank arm
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 $d_{st} = D$ is motien of the shaft at the juncture of the right hand erank arm.

We know that the resultant force at the bearing 1,

$$K_{\perp} = 63.3 \text{ k/V} = \frac{2(4.74)^{2}}{\sqrt{181}} = \frac{2}{\sqrt{181}} + \frac{2}{\sqrt{181}} + \frac{2}{\sqrt{181}} = 63.3 \text{ k/V}$$

.. Bending moment at the juncture of the right hand crank arm,

 $\left(\frac{041}{5} + \frac{221}{5}\right)$ 2.321 $-\left(\frac{041}{5} + \frac{221}{5} + 004\right)$ 5.53 = $M_{SI} = R_1 \left(b_2 + \frac{l_2}{2} + \frac{l}{2} \right) - R_0 \left(\frac{l_2}{l_2} + \frac{l}{2} \right)$

 $=34.7 \times 10^3 - 18.7 \times 10^3 = 16 \times 10^3 \text{ kW-mm} = 16 \times 10^6 \text{ M-mm}$

and twisting moment at the juncture of the right hand crank arm,

 $T_{SV} = F_T \times \tau = 84 \times 300 = 25 200 \text{ kW-imm} = 25.2 \times 10^6 \text{ W-imm}$

.. Equivalent iwisting moment at the juncture of the right hand crank arm,

$$\overline{L_{(12T)} + \frac{r}{2}(R_{(21)})^2 + \frac{r}{2}(R_{(21)})^2} = \frac{r}{2}$$
min-N ⁵01×58.92 = $\overline{L_{(15T)}^5 + L_{(15T)}^5 + L_{(15T$

We know that equivalent twisting moment (T),

 $e^{(t_1 b)} \cdot 22.8 = 24^{\circ} (t_1 b) \frac{34}{31} = e^{(t_1 b)} \frac{34}{31} = 0.01 \times 28.92$

(*mm/N 10 a MM Sh = T gableT)...

*suA mm 251 yes 2.521 = $\frac{1}{18}$ 10.01 × 23.6 = 22.8 \ 01 × 28.92 = $\frac{6}{18}$

(b) Design of right hand crank web

 $\sigma_{hT} = \text{Bending stress in the tangential direction.}$ $\sigma_{bR} = Bending$ stress in the radial direction; and

We also know that bending moment due to the radial component of $V_{\mathbf{0}^*}$

 $M_{\rm R} = H_{\rm R2} \left(b_{\rm I} - \frac{l_{\rm A}}{2} - \frac{1}{2} \right) = 47.3 \left(400 - \frac{155}{2} - \frac{140}{2} \right) \, \text{kM-mm}$

= 11.94 × 103 kW-mm = 11.94 × 106 N-mm

We also know that bending moment,

 $(z^{2} \times \frac{1}{2} = Z \times)$

(2)...

$$M_R = \sigma_{bR} \times Z = \sigma_{bR} \times \frac{1}{8} \times M_A^2$$

 $11.94 \times 10^6 = \sigma_{bR} \times \frac{1}{6} \times 245 (140)^2 = 800 \times 10^3 \sigma_{bR}$

 $\sigma_{MA} = 11.94 \% \ 10^6 \ / 800 \times 10^3 = 14.9 \ \text{M/mm}^2 \text{ or Mag}$

We know that bending moment due to the tangential component of Γ_{Q^i}

 $M_T = F_T \left(r - \frac{\dot{d}_{sL}}{2} \right) = 84 \left(300 - \frac{1.55}{2} \right) = 18 690 \text{ kW-mm}$

We also know that bending moment,

 $(S_{W1} \times \frac{1}{2} = S_{Y1})$...

$$M_{\rm T} = \sigma_{\rm bT} \times \Sigma = \sigma_{\rm bT} \times \frac{1}{6} \sigma = T_{\rm bT}$$

 $_{Td}\mathbf{D}^{0}01 \otimes 4.1 = ^{12}(245)041 \times \frac{1}{3} \times_{Td}\mathbf{D} = ^{3}01 \otimes 90.81$

 $64M \text{ for } ^{2}\text{mm/N-35.E1} = ^{6}\text{11.4.7.1} \setminus ^{6}\text{10.5.E1} = ^{2}\text{13.35.N/mm} = ^{2}\text{14.6}$

Direct compressive suess,

 $\sigma_b = \frac{P_R}{2 \text{ mm/M}^2} = \frac{94.6}{1.38 \times 10^6} = 1.38 \times 10^{-3} \text{ kW/mm}^2 = 1.38 \text{ AV/mm}^2$

and total compressive stress,

$$\Phi_{\hat{c}} = \Phi_{BR} + \Phi_{\hat{b}\Gamma} + \Phi_{\hat{d}}$$

= 14.9 + 13.35 + 1.38 = 29.63 W/mm² or MPa

We know that twisting moment on the arm, $T = H_{T2} \left(b_1 - \frac{L_2}{2} \right) = 42 \left(400 - \frac{155}{2} \right) = 13.545 \text{ kN-mm}$

and shear stress on the arm.

$$\tau = \frac{T}{Z_P} = \frac{4.5T}{w_i t^2} = \frac{4.5 \times 13.545 \times 10^6}{245 (140)^2} = 12.7 \text{ N/mm}^2 \text{ or MPa}$$

We know that total or maximum combined stress,

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

= $\frac{29.63}{2} + \frac{1}{2}\sqrt{(29.63)^2 + 4(12.7)^2} = 14.815 + 19.5 = 34.315 \text{ MPa}$

Since the maximum combined stress is within the safe limits, therefore, the dimension w = 245 mm is accepted.

(e) Design of left hand crank web

The dimensions for the left hand crank web may be made same as for right hand crank web.

(f) Design of crankshaft bearings

Since the bearing 2 is the most heavily loaded, therefore, only this bearing should be checked for bearing pressure.

We know that the total reaction at bearing 2,

$$R_2 = \frac{F_P}{2} + \frac{W}{2} + \frac{T_1 + T_2}{2} = \frac{314.2}{2} + \frac{50}{2} + \frac{6.5}{2} = 185.35 \text{ kN} = 185.350 \text{ N}$$

.. Total bearing pressure

$$= \frac{R_2}{l_2 \cdot d_{s1}} = \frac{185350}{365 \times 155} = 3.276 \text{ N/mm}^2$$

Since this bearing pressure is less than the safe limit of 5 to 8 N/mm², therefore, the design is safe.

Example 32.5. Design a side or overhung crankshaft for a 250 mm × 300 mm gas engine. The weight of the flywheel is 30 kN and the explosion pressure is 2.1 N/mm². The gas pressure at the maximum torque is 0.9 N/mm², when the crank angle is 35° from I. D. C. The connecting rod is 4.5 times the crank radius.

Solution. Given: D = 250 mm; L = 300 mm or r = L/2 = 300/2 = 150 mm; W = 30 kN= $30 \times 10^3 \text{ N}$; $p = 2.1 \text{ N/mm}^2$, $p' = 0.9 \text{ N/mm}^2$; l = 4.5 r or l/r = 4.5

We shall design the crankshaft for the two positions of the crank, i.e. firstly when the crank is at the dead centre and secondly when the crank is at an angle of maximum twisting moment.

. Design of crankshaft when the crank is at the dead centre

We know that piston gas load,

$$F_{\rm p} = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (250)^2 2.1 = 103 \times 10^3 \,\mathrm{N}$$

Now the various parts of the crankshaft are designed as discussed below:

(a) Design of crankpin

Let

$$d_c$$
 = Diameter of the crankpin in mm, and

$$I_c$$
 = Length of the crankpin = 0.8 d_c

...(Assume)

Considering the crankpin in bearing, we have

$$F_{\mathbf{p}} = d_{c} l_{c} p_{b'}$$

$$103 \times 10^{3} = d_{c} \times 0.8 \ d_{c} \times 10 = 8 \ (d_{c})^{2} \qquad \qquad \text{(Taking } p_{g} = 10 \ \text{N/mm}^{2}\text{)}$$

$$(d_{c})^{2} = 103 \times 10^{3} / 8 = 12.875 \text{ or } d_{c} = 113.4 \text{ say } 115 \text{ mm}$$

$$l_{c} = 0.8 \ d_{c} = 0.8 \times 115 = 92 \text{ mm}$$

and

Let us now check the induced bending stress in the crankpin.

We know that bending moment at the crankpin,

$$M = \frac{3}{4} F_P \times I_c = \frac{3}{4} \times 103 \times 10^3 \times 92 = 7107 \times 10^3 \text{ N-num}$$

and section modulus of the crankpin,

$$Z = \frac{\pi}{32} (d_c)^3 = \frac{\pi}{32} (115)^3 = 149 \times 10^3 \text{ mm}^3$$

.. Bending stress induced

$$= \frac{M}{Z} = \frac{7107 \times 10^3}{149 \times 10^3} = 47.7 \text{ N/mm}^2 \text{ or MPa}$$

Since the induced bending stress is within the permissible limits of 60 MPa, therefore, design of crankpin is safe.

(b) Design of bearings

Let

 d_1 = Diameter of the bearing 1.

Let us take thickness of the crank web;

$$t = 0.6 d = 0.6 \times 115 = 69 \text{ or } 70 \text{ mm}$$

and length of the bearing,
$$l_{\tau} = 1.7 d_{\rho} = 1.7 \times 115 = 195.5$$
 say 200 mm

We know that bending moment at the centre of the bearing 1,

$$M = F_{\rm p} (0.75 l_{\rm c} + t + 0.5 l_{\rm i})$$

= 103 \times 10³ (0.75 \times 92 + 70 + 0.5 \times 200) = 24.6 \times 10⁶ N-mm

We also know that bending moment (M),

$$24.6 \times 10^6 = \frac{\pi}{32} (d_1)^3 \sigma_b = \frac{\pi}{32} (d_1)^3 60 = 5.9 (d_1)^3$$

...(Taking $\sigma_b = 60 \text{ MPa or N/mm}^2$)

 $(d_1)^3 = 24.6 \times 10^6 / 5.9 = 4.2 \times 10^6$ or $d_1 = 161.3$ mm say 162 mm Ans.

(c) Design of crank web

w =Width of the crank web in mm.

We know that bending moment on the crank web.

$$M = F_{\rm p} (0.75 l_c + 0.5 t)$$

= 103 \times 10³ (0.75 \times 92 + 0.5 \times 70) = 10.7 \times 10⁶ N-mm

and section modulus,

$$Z = \frac{1}{6} \times w \cdot t^2 = \frac{1}{6} \times w (70)^2 = 817 \text{ w/mm}^3$$

A Bending stress,

$$\sigma_b = \frac{\dot{M}}{Z} = \frac{10.7 \times 10^6}{817 w} = \frac{13 \times 10^3}{w} \text{ N/mm}^2$$

and direct compressive stress.

$$\sigma_b = \frac{F_p}{w.t} = \frac{103 \times 10^3}{w \times 70} = \frac{1.47 \times 10^3}{w} \text{ N/mm}^2$$

We know that total stress on the crank web

$$\sigma_{\rm T} = \sigma_{p} + \sigma_{q} = \frac{13 \times 10^{3}}{w} + \frac{1.47 \times 10^{3}}{w} = \frac{14.47 \times 10^{3}}{w} \text{ N/mm}^{2}$$

The total stress should not exceed the permissible limit of 60 MPa or N/mm².

$$60 = \frac{14.47 \times 10^3}{w} \text{ or } w = \frac{14.47 \times 10^3}{60} = 241 \text{ say } 245 \text{ mm Ans.}$$

(d) Design of shaft under the flywheel.

 d_{r} = Diameter of shaft under the flywheel.

First of all, let us find the horizontal and vertical reactions at bearings 1 and 2. Assume that the width of flywheel is 250 mm and $l_1 = l_2 = 200$ mm.

Allowing for certain clearance, the distance

$$b = 250 + \frac{l_1}{2} + \frac{l_2}{2} + \text{clearance}$$

$$= 250 + \frac{200}{2} + \frac{200}{2} + 20 = 470 \text{ mm}$$

$$a = 0.75 l_2 + t + 0.5 l_1$$

$$= 0.75 \times 92 + 70 + 0.5 \times 200 = 239 \text{ mm}$$

and

We know that the horizontal reactions H_1 and H_2 at bearings 1 and 2, due to the piston gas load $(F_{\rm p})$ are

$$H_1 = \frac{F_R(a+b)}{b} = \frac{103 \times 10^3 (239 + 470)}{470} = 155.4 \times 10^3 \text{ N}$$

$$H_2 = \frac{F_P \times a}{b} = \frac{103 \times 10^3 \times 239}{470} = 52.4 \times 10^3 \text{ N}$$
Assuming $b_1 = b_2 = b/2$, the vertical reactions V_1 and V_2 at bearings 1 and 2 due to the weight

and

of the flywheel are

$$V_1 = \frac{W \cdot b_1}{b} = \frac{W \times b/2}{b} = \frac{W}{2} = \frac{30 \times 10^3}{2} = 15 \times 10^3 \text{ N}$$

$$V_2 = \frac{W \cdot b_2}{b} = \frac{W \times b/2}{b} = \frac{W}{2} = \frac{30 \times 10^3}{2} = 15 \times 10^3 \text{ N}$$

and

Since there is no belt tension, therefore the horizontal reactions due to the belt tension are neglected.

We know that horizontal bending moment at the flywheel location due to piston gas load.

$$\begin{split} M_1 &= F_{\rm P} (a + b_2) - H_1 \cdot b_2 \\ &= 103 \times 10^3 \left(239 + \frac{470}{2} \right) - 155.4 \times 10^3 \times \frac{470}{2} \qquad \dots \left(\because b_2 = \frac{b}{2} \right) \\ &= 48.8 \times 10^6 - 36.5 \times 10^6 = 12.3 \times 10^6 \,\text{N-mm} \end{split}$$

Since there is no belt pull, therefore, there will be no horizontal bending moment due to the belt pull, *i.e.* $M_2 = 0$.

.. Total horizontal bending moment,

$$M_{\rm H} = M_1 + M_2 = M_1 = 12.3 \times 10^6 \,\text{N-mm}$$

We know that vertical bending moment due to the flywheel weight,
$$M_{V} = \frac{W.\dot{b}_{1}.\dot{b}_{2}}{b} = \frac{W \times b \times b}{2 \times 2 \times b} = \frac{W \times b}{4}$$

$$= \frac{30 \times 10^{3} \times 470}{4} = 3.525 \times 10^{6} \text{ N-mm}$$

.. Resultant bending moment,

$$M_{\rm R} = \sqrt{(M_{\rm H})^2 + (M_{\rm V})^2} = \sqrt{(12.3 \times 10^6)^2 + (3.525 \times 10^6)^2}$$

= 12.8 × 10⁶ N-mm

We know that bending moment (M_p) ,

$$12.8 \times 10^6 = \frac{\pi}{32} (d_x)^3 \sigma_b = \frac{\pi}{32} (d_x)^3 60 = 5.9 (d_x)^3$$

$$(d_x)^3 = 12.8 \times 10^6 / 5.9 = 2.17 \times 10^6 \text{ or } d_x = 129 \text{ mm}$$

Actually d_s should be more than d_1 . Therefore let us take

$$d_{\rm x} = 200$$
 mm Ans.

2. Design of crankshaft when the crank is at an angtle of maximum twisting moment

We know that piston gas load,

$$F_{\rm p} = \frac{\pi}{4} \times D^2 \times p' = \frac{\pi}{4} (250)^2 \ 0.9 = 44 \ 200 \ \text{N}$$

In order to find the thrust in the connecting rod (F_0) , we should first find out the angle of inclination of the connecting rod with the line of storke (i.e. angle o). We know that

$$\sin \phi = \frac{\sin \theta}{l/r} = \frac{\sin 35^{\circ}}{4.5} = 0.1275$$

$$\phi = \sin^{-1}(0.1275) = 7.32^{\circ}$$

We know that thrust in the connecting rod,

$$F_{Q} = \frac{F_{P}}{\cos 6} = \frac{44\ 200}{\cos 7.32^{\circ 2}} = \frac{44\ 200}{0.9918} = 44\ 565\ N$$

Tangential force acting on the crankshaft

$$F_{\rm T} = F_{\rm O} \sin^2(\theta + \phi) = 44.565 \sin(35^{\circ} + 7.32^{\circ}) = 30 \times 10^3 \text{ N}$$

and radial force,

$$F_{\rm R} = F_{\rm O} \cos{(\theta + \phi)} = 44\,565 \cos{(35^{\circ} + 7.32^{\circ})} = 33 \times 10^3 \,\text{N}$$

Due to the tangential force (F_T) , there will be two reactions at the bearings 1 and 2, such that

$$H_{\text{T1}} = \frac{F_{\text{T}}(a+b)}{b} = \frac{30 \times 10^3 (239 + 470)}{470} = 45 \times 10^3 \text{ N}$$

$$H_{\text{T2}} = \frac{F_{\text{T}} \times a}{b} = \frac{30 \times 10^3 \times 239}{470} = 15.3 \times 10^3 \text{ N}$$

and

Due to the radial force (Fp), there will be two reactions at the bearings 1 and 2, such that

$$H_{R1} = \frac{F_R (a+b)}{b} = \frac{33 \times 10^3 \times (239 + 470)}{470} = 49.8 \times 10^3 \text{ N}$$

and

$$H_{R2} = \frac{\dot{F}_{R} \times a}{b} = \frac{33 \times 10^{3} \times 239}{470} = 16.8 \times 10^{3} \text{ N}$$

Now the various parts of the crankshaft are designed as discussed below:

(a) Design of crank web

We know that bending moment due to the tangential force,

$$M_{\rm bT} = F_{\rm T} \left(r - \frac{d_1}{2} \right) = 30 \times 10^3 \left(150 - \frac{180}{2} \right) = 1.8 \times 10^6 \text{ N-mm}$$

: Bending stress due to the tangential force,

$$\sigma_{bT} = \frac{M_{bT}}{Z} = \frac{6M_{bT}}{t.w^2} = \frac{6 \times 1.8 \times 10^6}{70 (245)^2} \qquad \dots (: Z = \frac{1}{6} \times t.w^2)$$

$$= 2.6 \text{ N/mm}^2 \text{ or MPa}$$

Bending moment due to the radial force,

$$M_{BR}^{\parallel} = F_{R} (0.75 l_{c} + 0.5 t)$$

= 33 × 10³ (0.75 × 92 + 0.5 × 70) = 3.43 × 10⁶ N-mm

... Bending stress due to the radial force,

$$\sigma_{bR}^{1} = \frac{M_{bR}}{Z} = \frac{6 M_{bR}}{wt^{2}} \qquad ... (: Z = \frac{1}{6} \times wt^{2})$$

$$= \frac{6 \times 3.43 \times 10^{6}}{245 (70)^{2}} = 17.1 \text{ N/mm}^{2} \text{ or MPa}$$

We know that direct compressive stress,

$$\sigma_{d} = \frac{F_R}{w \cdot t} = \frac{33 \times 10^3}{245 \times 70} = 1.9 \text{ N/mm}^2 \text{ or MPa}$$

:. Total compressive stress,

$$\sigma_c = \sigma_{bT} + \sigma_{bR} + \sigma_{d} = 2.6 + 17.1 + 1.9 = 21.6 \text{ MPa}$$

We know that twisting moment due to the tangential force,

$$T = F_{\rm T} (0.75 l_e + 0.5 t)$$

= 30 × 10³ (0.75 × 92 + 0.5 × 70) = 3.12 × 10⁶ N-mm

Shear stress,
$$\tau = \frac{T}{Z_P} = \frac{4.5 T}{w \cdot t^2} = \frac{4.5 \times 3.12 \times 10^6}{245 (70)^2}$$
 ... $\left[\because Z_P = \frac{w \cdot t^2}{4.5} \right]$
= 11.7 N/mm² or MPa

We know that total or maximum stress,

$$\sigma_{max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2} = \frac{21.6}{2} + \frac{1}{2}\sqrt{(21.6)^2 + 4(11.7)^2}$$
$$= 10.8 + 15.9 = 26.7 \text{ MPa}$$

Since this stress is less than the permissible value of 60 MPa, therefore, the design is safe.

(b) Design of shaft at the junction of crank

Let $d_{s1} = \text{Diameter of shaft at the junction of crank.}$

We know that bending moment at the junction of crank,

$$M = F_{Q}(0.75l_c + t) = 44\,565\,(0.75 \times 92 + 70) = 6.2 \times 10^6\,\text{N-mm}$$

 $T = F_{T} \times r = 30 \times 10^3 \times 150 = 4.5 \times 10^6\,\text{N-mm}$

... Equivalent twisting moment,

and twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(6.2 \times 10^6)^2 + (4.5 \times 10^6)^2} = 7.66 \times 10^6 \text{ N-mm}$$

We also know that equivalent twisting moment (T_p) ,

$$7.66 \times 10^{6} = \frac{\pi}{16} (d_{s1})^{3} \tau = \frac{\pi}{16} (180)^{3} \tau = 1.14 \times 10^{6} \tau \qquad ... (Taking d_{s1} = d_{1})$$

$$\tau = 7.66 \times 10^{6} / 1.14 \times 10^{6} = 6.72 \text{ N/mm}^{2} \text{ or MPa}$$

Since the induced shear stress is less than the permissible limit of 30 to 40 MPa, therefore, the design is safe.

(c) Design of shaft under the flywheel

Let $d_{s} = \text{Diameter of shaft under the flywheel.}$

We know that horizontal bending moment acting on the shaft due to piston gas load,

$$\begin{split} M_{\rm H} &= F_{\rm P} \; (a+b_2) - \left[\sqrt{\left(H_{\rm R1}\right)^2 + \left(H_{\rm TI}\right)^2} \right] b_2 \\ &= 44 \; 200 \left(239 + \frac{470}{2} \right) - \left[\sqrt{\left(49.8 \times 10^3\right)^2 + \left(45 \times 10^3\right)^2} \right] \frac{470}{2} \\ &= 20.95 \times 10^6 - 15.77 \times 10^6 = 5.18 \times 10^6 \; \text{N-mm} \end{split}$$

and bending moment due to the flywheel weight

$$M_{\rm V} = \frac{W.b_1.b_2}{b} = \frac{30 \times 10^3 \times 235 \times 235}{470} = 3.53 \times 10^6 \text{ N-mm}$$

... $(b_1 = b_2 = b_1/2 = 470/2 = 235 \text{ mm})$

:. Resultant bending moment,

$$M_{\rm R} = \sqrt{(M_{\rm H})^2 + (M_{\rm V})^2} = \sqrt{(5.18 \times 10^6)^2 + (3.53 \times 10^6)^2}$$

= 6.27 × 10⁶ N-mm

We know that twisting moment on the shaft,

$$T = F_T \times r = 30 \times 10^3 \times 150 = 4.5 \times 10^6 \text{ N-mm}$$

.. Equivalent twisting moment,

$$T_{e} = \sqrt{(M_{\rm R})^2 + T^2} = \sqrt{(6.27 \times 10^6)^2 + (4.5 \times 10^6)^2}$$

= 7.72 × 10⁶ N-mm

We also know that equivalent twisting moment (T_e) ,

$$7.72 \times 10^6 = \frac{\pi}{16} (d_s)^3 \tau = \frac{\pi}{16} (d_s)^3 30 = 5.9 (d_s)^3$$
 ...(Taking $\tau = 30$ MPa)
 $(d_s)^3 = 7.72 \times 10^6 / 5.9 = 1.31 \times 10^6$ or $d_s = 109$ mm

Actually, d_s should be more than d_1 . Therefore let us take

$$d_s = 200 \text{ mm Ans.}$$

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hostidantal four stroke diesel engine has the following specifications, (1)
  Speed = 1200 r.p.m
 Imep = 0.35 N/mmp
 Mechanical efficiency = 80%
 Calculate the demensions of the cylender and the cylender head,
 Given that
    Brake power, B.P = 5kW = 5000 watts
     Speed, N = 1200 8. pom
    Indicated man effective pressure
          Pro =0.35 N/mma
      Mechanical efficiency, Mmech = 80% = 608
(a) Bore and Length of the Cylindes
    Let, 1 - Length of stroke in m
        Di - Bore of the Cylinder in mm
        A - cross-sectional area of the cylender
             =\frac{1}{10} \times D_g
    we know that,
   Indicated passes,
             I.p = Brake power (B.p).

Mechanical officiency (Imech)
            1.p = 5000 = 6050 W
      But we know that,
       indicated passes,
            Pop = PmxlxAxN
                  (: For 4-stroke, m = N = 1200 = 600 8.p.m)
           6950 = \frac{0.35 \times 1 \times A \times 1200}{60 \times 2}
            1xA = 6250x60 x2
0.35 x1200
      Let us assume,
             1 = 1.5 Dimm = 1.5 Di
  \frac{1.5}{1000} D_i \times \frac{11}{4} D_i^2 = \frac{6250 \times 60 \times 2}{0.35 \times 1200}
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 $D_{1}^{3} = \frac{6250 \times 60 \times 2 \times 1000 \times 4}{0.35 \times 1000 \times 1.5 \times 1000 \times 4}$ $D_{i} = 3 \frac{6250 \times 60 \times 2 \times 1000 \times 4}{0.35 \times 1200 \times 1.5 \times 17}$ =114.871mm 2115mm .. D. = 115mm ·· 人=1.5 D,=1.5×115=172.5mm Taking a clearance on both Sides of the Cylinder equal to 15% of the stroke, therefore length of the Glander, L = 15% of length of stake + Length of (:· l = 1-5D1) =0.151+1 L = 1.151 = 1.15 x 172.5 =198.375 mm 2200 mm (b) Thickness of cylinder Head we know that the thickness of the Glander head, $t_h = D_f \times \sqrt{\frac{C.P}{\sigma_i}}$ since the maximum pressure(p) in the engine cylinder is taken as 9 to 10 times the mean effective pressure(pm) therefore, P=9×Pm=9×0-35=3.15 N/mm3. Let us assume (=0.1 (constant) And permissible tensile stress, Qt = 48 N/mmg -th= 115 x | 0.1 x 3.15 = 9.959 mm 20mm :th = lo mm (c) Size of Stude for Cylinder head Preserve force acting on stude, = Areax Total maximum pressure $=\frac{10}{4} \times 0^{2} \times P = \frac{1}{4} \times 115^{2} \times 3.15$ = 32718.706 N

The number of studs (ng) "is taken between, ns =0.010,+4 and ns =0.020,+4 ng = 0.01 x 115+4 and ng = 0.02 x 115+4 ms = 5.15 to 6.3 Let us take, no = 6 The total resisting force offered by all the studs = n3 * [[GG] x of where, of = core diameter of Ital of = Allowable tensile stress of stud material which is usually mickel steel. of = 65 N/mmg (assuming) Resistance force offered by the Studs, = 42 x (4) x of =6 x 1 6.84d 2 x 65 usually de=0.84x Nominal diameter) = 316.12942 Equating equations (0 & 6), we get 216.129 dn = 32718.706 dn2 = 32718-706 dn = \ 32718.706 iodn = 12:304 mml say 14mm Pitch circle diameter of the study, Dp = D1 +3dm Dp =115+3x14=157mm Pitch of the stude $=\frac{\P \times O_{P}}{\gamma_{0}} = \frac{\P \times 157}{6} = 82.205 \text{ mm}$ for a leak-proofjoint, the pitch of the stude should lie between 19/dn to 28.5/dn.

 $9375 = \frac{0.35 \times \frac{1.40}{1000} \times \frac{11}{4} (D^2) \times 500}{2}$ D = 3/2923254.057 . D = 14a.98 .. Bore of Cylinder, D= 143 mm Length of Stroke, L=1.4×143=200mm Length of Cylinder, l = 1.15 x L = 1.15 x 200 =230 (b) Wall thickness (t) Hoop stress, 0 = 35 MPa (given) using the following relation, $t = \frac{\partial \times \sigma}{\partial \times \sigma} + c$ For D=200mm, reboring Constant, C=6.3 $t = \frac{3.5 \times 143}{8 \times 35} + 6.3$.. t = 13.45 mm (C) Thickness of Cylinder head(th) 1s calculated as, th = 0/ Cp Permissible stress, oc =45MA =45N/ (given) Let, C=0.1 th= 143 \ \ \frac{0.1x3.5}{45} : th = 12.61mm Size of stude stress, of = 65MPa =65 Nmm (gren) Number of Stude, ng = (0.010+4) to (0.02D+4) =(0.01x143+4)to (0.02 ×143+4) Gas bad on stude = $\frac{\pi}{4} \times 0^{3} \times p$ $=\frac{11}{4}\times(143)^{2}\times3.5$ = 56219.12N - (1) Resisting force by all bolts $= \int_{S} \times \frac{1}{t} (d_{c})^{2} \times \sigma_{t}$ = $7 \times \frac{11}{4} (d_c)^2 \times 65$

=357.36 dc2 -----@ Equating equations (and (), de = 56212.12 dc = 157.298 = 12.54mm Core diameter of Stude, d_= 12.54mm Nominal diameter of stude, dn = dc 12.54 0.84 = 14-92mm Pitch of stude lies between, Ps=19/dn to 28.5/dn · Minimum pitch of stude = 19 Jan =19/14-92 = 73.39 :. Maximum pitch of stude = 28.5 Jdn =28.5 × 14,9 = 110-08mm -> Design a Cylindestora four strake petrol engine developing 15kW at 2000 rpm. Assume indicated mean effective Pressure ato be 1.2 MPa. A Given that, Brake power, B.p=15kW Speed, N = 2000 s.p.m Mean effective pressure, Pm = 1.2MPa Assume mechanical efficiency, Mmech = 80% = 0.8 (a) Bore and Length of the cylinder Indicated power, I.p = Brake power (B.p)

Mechanical efficiency (nmech) = 15x103 = 18750 W. P= 18.75×103 W Indicated power, Pop = PmxlxAxm $\frac{1}{2} = \log \sqrt{2}$ for 4-stroke n= N

: 18.75×103 = 1.2×1×A×1000 2xA = 937.5 mm3 1.0 Let us assume, l=1.5 x D, mm =1.5 x 10 , D, m from eq. (1) 1.5 × 103. Dr × 1 (Dr) = 937.5 0.001178 Di = 937.5 .. Dp = 92.671 mm № 93 mm ·· l = 1.5D; = 1.5x93 = 139.5mm ~140mm (b) Thickness of Cylinder head The thickness of Cylinder head, th = Di x Cp Assume, C=0.1, P= 9. Pm=9x1.2 =10.8 N/mm ong of = 43 M/my th = 93 x 0.1x10.8 =14.913 (C) size of stude for Cylenders head The force acting on the glirder head, F= 11 0,2 xp = (43) 2 × 10.8 =73363.414 N. The number of stude (ng) is taken between, ns =0.010;+4 and ns =0.020;+4 =0.01(93)+4 = 0.02(93)+4 = 4.93 = 5.8€ Assume, ng =5 The resisting force offered by all the Stude, = 10 * (4 (9) & & =5x11 (0.84 dn)2 x65 (.: Assume, de = 0.84 dy and = 65 N/mm2) = 180.108 dn 3 Edragid @'E D'

73363.414 = 180.108 da dn = 20, 182 22 21 mm Pitch circle diameter of stude(Dp) is taken es Dottedn Dp = 93 +4/(21) = 177mm Pitch of Stude _ TIXDP = TIXDD = 92.677mm In order to have leak proof joint, the pitch of the stud should lie between 19/da to 28.5/di Minimum potch of stud = 19 Jan = 19 Jai Maximum pitch = 28.5/dn = 28.5/21 =130.603 ~~ since, the pitch of the stud, P= 92.677 lies between 87.069 to 130.603/m therefore, the design is satisfactory The cylinder of a four stroke diesel Engine has the following Specifications Brake pocces = 3kW. speed = 800 x.p.m Indicated mean effective pressure =0.3 MA Mechanical efficiency = 80 % Determine the bose & length of the allendes line. Given that, Brake passer, B.P=3kW=3000W Speed, N = 800 r.p.m for 4-stroke, n = 1 = 800 = 400 5.pm Mean effective pressure, Pm = 0.3 MR Mechanical efficiency, Mmech=80%=0.8 =0.3 N mm3 Bore and Length of the Glinder Line I - length of stocke Indicated power (I.p) = Broke power | Mechanical efficient I.p=B.p=3000 =3750W But, I.p = Pmx (xAx400

 $3750 = 0.3 \times 1 \times 1.400$ (3) where, A-Area of cross-section of Glinder = II D, 2 1=1.50 mm = 1000 in m $\frac{1.5D_1}{1000} \times \frac{1}{4} \times D_1^2 = 1875$ $D_p^3 = \frac{1000 \times 1875}{1.5 \times 11}$ D; = 3/159/549.431 D, =116.754 Bore of cylindes, D, = 116.754mm length of Cylinder = 15% of length of Stroke + Length of stroke L =1.15 x length of 9boke =1.15 X1.5 xDp =1.15 ×1.5 ×116.754 ·· L = 201.40/mm The cylindes of a fow stooke diesel Engine has the following specifications: Brake Posses = 3.75 kW; Speed = 10000 Pm Indicated man effective presence = 0.35 MPa , Mechanical efficiency = 80% Determine the bose and length of the Glinder line.

A: Given-that,

Brake power, B.p = 3.75kW=3750W

Speed, N = 1000 8.p.m

For four Stroke, n= N = 1000 = 500 8.p.m

Mean effective pressure, Pm = 0.35MPa

= 0.35 N/mm²

Mechanical effeciency, 1/mech = 80.6

Bore and length of the cylinder

Let, L-Length of stroke

Indicated passes (I.p) = Brakepower

Mechanical

efficiency

 $P.p = \frac{8.p}{\eta_{\text{mech}}} = \frac{3750}{0.8} = 4687.5 \text{ N}$ But, $I.p = \frac{Pm \times 1 \times A \times 500}{60}$ $4687.5 = \frac{0.35 \times 1 \times A \times 500}{60}$ $1 \times A = \frac{4687.5 \times 60}{0.35 \times 500} = 1607.143$

where, A = Area of cross-section of Glinden.

 $A = \frac{11}{4} \times D_1^2$ $l = 1.5D_1 \text{ in min} = \frac{1.5D_1^2}{1000} \text{ in m}$

 $\frac{1.50!}{1000} \times \frac{11}{4} \times D_1^2 = 1607.143$ $D_1^3 = \frac{4 \times 1000 \times 1607.143}{1.5 \times 11}$

De =3/1364185.227

D; = 110.907~111mm

.. Bore of Cylender, Dralling

Length of Cylinder = 15% of length of stocker Length of

=1015 x Length of stroke

= 1.15x1.5 xD;

Length of cylinder = 191. 475mm

A four stoke diesel engine has the following specifications Brake power=5km, speed = 1200 pm, indicated mean effective pressure = 0.35 N/mm², Mechanical efficiency = 80%. Determine.

i Bore and length of the cylinder

ii, Thickness of the liner.

iii, Thickness of the cylinder head.

iv Size, number and pitch of stude.

Refer Q(I) in Cylinders.

Permissible limit of tensile stess:

for glinder liver 10 ranges from

35 Mpa to 100 Mpa.

Assume, the tensile stess, of =50 N/m².

Maximum gas pressure Pmax = 10 × Pm.

Pmax = 10 × 0.35 = 3.5 N/m².

For Di = 115 mm, reboing Constant.

C = 3.88 mm.

Thickness of the cylinder liner is givenous.

Thickness of the cylinder liner is givenous.

1 = \frac{Pmax}{200} \times \frac{1}{200} + 2.88 = 6.905

> The piston of an automobile engine " is held to the connecting sed by a civist pin having an external diameter of somm and internal diameter of limm. The tength of the basing in the rod end is 25mm and the bearing in each boss of the piston's 15mm. The maximum load transmi-tted is 14.50kM. a) Determine the bearing pressure between the rod end and the wrist pin. b) Compare the maximum bending stress in the pin, osciming that it is a simple beam with aunitorin (c) Compute the maximum deflection of the pin. Given that, outer diameter of pin, do = 20mm Immer diameter of pin, di=11mm Length of the bearing, 1 = 25mm Maximum load transmitted, W=14.5KN=14.5x03N (a) Determine the Bearing pressure between the Red End and consist pin, Load on pin due to bearing Pressure, W = Booking pressure x Bearing area. W=Poxdoxl, 14.5×103=P6×30×25 Pb = 29 Nmm2 (b) Maximum Bending in the centre of pin $M = \frac{P \cdot D}{8}$ where, D - Diameter of Cylinder 1. L =0.45D $D = \frac{35}{0.45} = 55.556$ D ≥ 55 mm .. Maximum bending moment, M=14.5x103x55

M = 99687.5 N-mm (2).. Maximum bending Stress is given as, $G = \frac{M}{2}$ where, Z-Section modulus ofpin •• $Z = \frac{11}{32} \left[\frac{d_0^4 - d_0^4}{d_0} \right] = \frac{11}{2} \left[\frac{(20)^4 - (1)^4}{20} \right]$ Z = 713.529~~~~ 3 $\therefore G = \frac{M}{Z} = \frac{99687.5}{713.589}$ \$ = 139-711 N mmg since, the induced stress is less than the permissible value, i.e., 140 Mmm, therefore, the design is safe. (C) Maximum Deflection of the pin Maximum deflection of a simply Supported beam with uniform distributed load is given as, where, I - Moment of inertia E - Young's modulus for the pin section, I= 1 Cdo4-d;4) I =7135.293 mm4 :. 8 = (4.5×103×(25)3 48×65×18×135+293 (Assume, for aluminium E = 65 × 103 N/mm3) 6 = 0 = 0 mm The deflection of Pin is 0.01mm. Design an aluminium alloy picton for a Single acting four stocke engine for the following data, Cylinder bore = 400 mm Stroke, 1=375mm =0.375m Moximum gas pressure, P=9MPa=9N/mm?

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Break mean effective pressure, Pm=2MPa

= 2 M/mmg Speed, N = 50 r.p.m fuel consumption, m=0-22 Eg/KW/hr 1. Dosign of piston hand (a) Strength consideration Thickness of piston head, $t_{H} = \sqrt{\frac{3PD^2}{160L}}$ (for aluminium, of =20to 25 MB) $= \sqrt{\frac{3 \times 9 \times (400)^{2}}{16 \times 34}} = 106.06 mm$ (b) Thermal Consideration Thickness of piston, tH = H 12.56k(TC-TE) Number of working Strokes per minute n= 2 = 50 = 25 (For 4- stocke engine) Cross-sectional area, H = 1 × (4 ∞) = 125663 - الم سيم I.p = PmLAn _2×375×125663·70×25 60 × 100 8 = 39269.90 watts = 39.269kW Assuming mechanical efficiency, n=80 % .. B.p=P.px7m =39.26 x0.80 = 31.41 kW H=Cxc.Vxm xB.p =0.02×45×103×30.93×31.4) (Assuming C=0.05, C.V=42×103kJ/kg) =4.0317 KW = 4031.7W.

Throkness of piston, $-\frac{1}{4} = \frac{1}{12.56(T_C - T_E) \times k}$ = 4031.7 12.56x111x17475 [: k=174.75,4/m/°c] (For aluminium, TC-TF=11°C) · + + = 16.54mm Paking the larger of the team values 1.e, 107 mm 2. Radial Ribs Radial ribs may be 4 in number Thickness of the orbs vanies from the total $-t_r = \frac{107}{3} + \frac{107}{3} = 35.66 + 053.5$.. Average thickness of the sibs, tr=45mm 3. <u>Piston Rings</u> Let, ti-Radial thickness 41=D/3A $= 400 \times \sqrt{\frac{30003}{100}} = 12000$ Axial thickness of Piston rings, ta = 0.7+, to+, =0.7 ×12 to 12 = 8.4 to 12 ta ~ lomm Minimum axial thickness of the Piston Mings, $t_0 = \frac{D}{lon} = \frac{400}{lox4}$ (: Assuming number of preton sings, nr=4) .. ta = 10mm Width of the top land, p1 = 44 to 1.344 = 101 to 101 x1.3

```
= lot to 128:4
  p12 115 mm
width of the lands between the sing
grooves may be,
      ba = 0.75to to to
          = 0.75 x10 to 10
          = 7.5 to lo
 Gap between the free end of the
 ring,
     G1 = 3.5t, to 4t,
       =3.5 ×12 10 4×12
      = 42 to 48
  +. G, ≈ 45 mm
 Gap when the ring is in the
  Cylinder,
     62 =0.002D to 0.004D
        בסי שם אנים לם . סיסטי צ ניסט
        =0.8 to 1.6
    -1 G2 ~ 1 mm
4. Piston Barrel
Depth of the piston sing,
          b = + 40 < 4
            =12+0.4=12.4 mm
Maximum thickness of the basisel,
        t3 = (0.03 D+b+4.5) mm
           =(0.63 x400 +12.4+4.5)
           = 28.9-7
Piston wall thickness towards the
open end,
    ty = 0.25 ts to 0.35t3
        = 0.25 x 28.9 to 0.35 x 28.9
        =7.225 to 10.115
  : ty~8.5mm
5. Piston skirt
 Maximum Side thrust,
     R= \u x \frac{10}{4} D 2 x P
```

=0.1x TX(400)2 × 9 (ASSume = 113097.33Side thrust due to bearing pressure, R=R XDXL =0.7×400 xl. (: Bearing pressure Po is limited to 0.7 N/mm3) : 113097.33 =0.7x400xl : Length of Skirt, 1=403.91 6. piston pin Pb = Bearing preserve at bushing =35 N mon for aluminium bushing Load on the pin = Pb, xdox1, =35 x do x0-45 x400 (taking & as 0.45D) =6300 de N Maximum load on the piston due to Jas pressure, $b = \frac{r}{U} D_3 \times b$ p× €00+3× 1]== = 1130973-35 N towating maximum load on the Piston to the load on the piston Pin, 1130973.35 =6300 do do = 179.51 = 180mm Inner diameter of the pin, di = 0.600 = 180×0.6 = 108mm Checking the induced bending Stress in pin Maximum bending moments $M = \frac{P.D}{8}$ _(130973-35 ×400

-56548667.5 N-mm But, $M = \frac{\Omega}{3a} \left[\frac{d_0 + -d_1 + 1}{d_0} \right] O_b$ 56548667.5 = 1 (180) 4 (108) 4 7

: # = 113-47 N/mm2 Since, the induced bending Stress is below the permissible stress of 40 N/mm? (for heat treated alloy steel), the design is safe

> Design a piston, piston rings and gudgeon pin for a single acting four Stroke engine for the following data, cylinder bosie = toomm

Stroke = 125 mm Maximum gas pressure = 5MPa Breake mean effective pressure

Fuel consumption = 0.15 kg Brake Power in kulh

Speed = 2000 r.p.m

Given that

Bore, D = 100 mm

Stroke, L= 125mm

Gos pressure, P=5MPa=5N/mm

B.M.E., Pm = 0.6MPa = 0.6 N/mm

Fuel consumption, m = 0-15kg/B.plh

=41.7×10 kg/Bp/s

Speed'11 = 5000 a.b.m 1. piston head or Crown

(a) strength Consideration The throkness of piston head is given by,

 $t_{H} = \sqrt{\frac{3PD^{2}}{16 \cdot 67}} = \sqrt{\frac{3 \times 5 \times 100}{16 \times 38}}$

(: For cost fron, of = 38 N/m)

: + + = 15.707mm= 16-mm

(b) Thermal Consideration

The thickness of Piston head,

Since the engine is four stroke, the number of Cooking Strokes per minute,

 $M = \frac{N}{N} = \frac{2000}{2} = 1000$

Area of cross section of the Cylinder,

A=\(\frac{1}{4}\)\(\rap{2}=\frac{1}{4}\)\(\rap{1}\)\(\rap{2}=\frac{1}{4}\)\(\rap{1}\)\(\rap{2}=\frac{1}{4}\)\(\rap{1}\)\(\rap{2}=\frac{1}{4}\)\(\rap{2}=\frac{1}

Indicated horse power,

11 P. P = Pm-LAn

= 6-6x125x7853-982 x 1000

=9817477.5 N-mm/s

= 9817477.5 x103 N-m/s

=9817- 478 W

1. 1. P = 9.817 KW

: Assuming mechanical efficiency as

1 mech =0.8

: Brake power 1 B.P. I. Primech.

=9.817x0.8

BP =7-854 kW

Heat from through piston head is given by,

H= CxHCVxmxB.P

(Assuming, C=0.05 and HCV=42×103 KJ/kg)

H = 0.05 x42 x 103 x 41.7 x 10 87.854

H = 0-688 KW

= 688 W.

.. Thickness of piston head,

: +b=11.4 -Piston wall thickness towards the open end, tu = 0.25 tb to 0.35 x tb =0.25×11.4 to 0.35 ×11.4 = 2.85 to 3.99~~~ Considering an average value, i.e, tu = 3mm 5. piston skirt The maximum side thrust on the cylindes, R= N#T XDXXP $=0.1\times\frac{\pi}{4}\times(100)^{8}\times5$ (: Assume, R= 3926.991N -Also, the side thrust on the piston basissel (A). R=Pb x Dx 1 = 0.45 × 100 ×1 (: Bearing pressure p) is, limited to 0.45 N/mm?) R = 45lEquating equations(1) and (2), we geti 3926.991 = 451 2 = 87-266 ≈ 90mm Skirt length, 1 = 90 mm : Total length of the piston L = Length of the skint + Length of the oiling Section + Top land L = l+(4.ta+3.ba)+b1 = 90 + (4×3+ 3×2.5) +18 L = 127.5 mm L 02/28mm 6. piston pin (or) Gudgeon pin

Let, Pb, - Beasing pressure at bushing =25 N/mm? for bronze bush do-outer diameter of pin in l, - Length of pin in mm Maximum Load on the piston due to Jas pressure, $= \mathcal{F} \mathcal{D}_{S} \times P$ = II x(100) x5 = 39269.908N The load on the piri due to bearing pressure, = Bearing pressure x Bearing axea = Pb, x do xl, = 25 x do x (0.45 x 100) (:1,=0.45D) = 1125 × do Equating equations (1) and (2), we get, 39269.908 = 1125do do = 34-907-40 a 35 mm Inside diameter of the pin, d: = 0.6 x do = 0.6 x 35 = 2 mm Checking the Induced Bending Moximum bending moment at the Centre of pin, Mmax = P-D = 39269-908 x 100 = 490873.85 N-mm Also, the maximum bending moment is given as Mmax = 1 [dot-d; 4] x 9 $\sigma_{b} = \frac{32M}{n} \left[\frac{d_{0}}{d_{0}^{4} - d_{0}^{4}} \right]$ $= \frac{32 \times 490873.85 \left| \frac{35}{35^4} \right| \frac{35}{35^4}}{35^4}$

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 $t_{H} = \frac{H}{12.56 \cdot k(T_{C} - T_{E})}$ For Castiron, (TC-TE) = 220°C and K = 46.6 W/m/°C] £H = 13.56 x46.6x32 = 5.343mm Consider larger of two values, i.e, ty = 16 mm. a. Radial Ribs The thickness of the piston wall or ribs, tp = TH to TH in Thickness of ribs, th= 16 to 2 25.333 to Average trickness of ribs, to = 7 mm 3. Piston Rings The radial thickness of Piston ring is given by, $tr = D \cdot \sqrt{\frac{3Pr}{\sigma_L}}$ (Assuming radial wall pressure Pr = 0.035 N/mm2 Emer 350bb tensile stress, of = 90 N/mm3) :. tr = 100 x 3x0.035 tr = 3.416 mm i tr = 3.5 mm Axial thickness of the piston rings ta = 0.7tr to tr = 0.7 x 3.5 to 3.5 = 2.45 to 3.5 mm Axial thickness of the piston rings

ta = 0.7 tr totr = 0.7 × 3.5 to 3.5 = 2.45 to 3.5 mm Considering an average value, i.e. ta = 3 mm The minimum axial thickness of the ringe, ta = D $= \frac{100}{10 \times 4} = 2.5 (Assuming number of rings, n_r = 4)$.. to = 2.5 mm Thus, the axial thickness of 3mm is Satisfactory (i.e., ta=3 mm) width of the top land, | p| = tH +p 1.2+H =16 to 1.3 x 1c p = 16-10 19.2mm ~ 18mm Width of other sing loads, b2 = 0.75 ta to ta = 0.75 × 3 to 3 b2 = 2.25 to 3mm ≥ 2.5mm Grap between the free end of the ച്ചുവ G, = 3.5tr to 4tr $=3.5 \times 3.5 + 64 \times 3.5$ G1 = 12.25 to 14mm ≥ 13mm Also the gap when the ring in the Cylindes is given by, Ga = 0.002D to 0.004D = 0.002 x 100 to 0.004 x 100 ** 65 = 0.5 to 0.4 mm 503 mm 4. pistol Barrel Depth of Piston ring growves, b = tr +0.4 = 3.5 +0.4 = 3.9mm Maximum thickness of basissel th = (0.03D + b + 4.5) mm =(0.03 × 100 +3.9+4.5)

- CH = 13.56×46.6×30 (For Gost from, 1/2 = 46.6 H/m/2 and (五) N 20.51 HT intrapisation of basis searysisting : H = 3600 M Or 3.6 KM. 23561.945 x 3 61 x 5 0 1 x 4 4 x 5 0 1 x 4 4 5 x 10 = KIKEJ EASSERING C= 0.05 and HCV=44X18 H = Cx HCN x w x Bp theat Acousty Answell prison head .. B.b = 93.26 KM m576-19588 = 00EXT&P. 31415 x 36.0 x 3.00 Bake power, B.P = Rox100 Fmm759-31415-927mm= + Area of the cylinder, ₩= 1300 = 300 Number of working strokes/min, mm 98E.18= #4 : BMOT = pxm284} (000)2 [ASSUME [-4- (180) = H-Strength, to sized and no assorabilities booth noteing .1 Since, the induced bending stress-nts of the piston are determined as is well below the permissible follows. -anagma 2 200 isov rat 200 iznamib 2011

opg.1=7: 5.1= JP / PWY KME 4016 RA OIX G4 - PO = m : 4481 gy = 3600 = AHER CONSUMMED 25 PARTH Parail 4= Moximum gas pressure, P=4MPa Pm=0.6 Mg = 0.6 12 mm Break mean effective presente, speed, N = 600 rpm 2410kg, L = 250mm, = 0.25m Cylindes bosse D= 200mm Gilven data, " , shormostymuses ant state broo Assume suitable data, P required Gol = bor gribacroa to bos Illame in hand has other alt (948) Firel Consumption = 0.05 kg per = 4MB: =0.6 Mps Maximum gas pressure Broke man effective presence (wds 009 = boods immogra syons go Cylindes borse = 200 mm; Length engine with the following dato; Single acting four stroke dissel Design a west from piston for a Strees of 1404/mm2. Therefore

.. of = 133.989 Whoma

ty =0.02795m or 27.95mm Take larger of two values, i.e., ty = 27.95 mm = 28 mm (b) Radial Ribs Number of ribs =4 Thickness of sibs, tp = 38 to 38 = 9.33 to 14 mm Taking, tR=12-mm. (c) Piston Rings Assume number of sings, n=4ings oil ring. Radial thickness of piston rings . $t_1 = D \sqrt{\frac{3\rho_U}{\sigma_1}} = 200 \sqrt{\frac{3x0.04}{100}}$ Take Pu = 0.04 whomas and of=100 =6.93 ~ 7~mm Axial thickness of piston sings, to = 0.7+1 to +, =0.7(7) to 7 = 4.9 to 7 Take to = 6 mm Minimum axial thickness of the Piston ring, to = D [Taking mg = 5] $= \frac{800}{10x5} = 4$. Axial thickness is satisfactory. width of the top land, pl=+H to loth = 38 to 1.3 x28 = 28 to 33.6 mm Taking, b, = 30mm width of the other ring lands,

ba = 0.75 to 6 by =0.75(6) to 6 = 4.5 to 6mm Take, by =5mm Gap between free ends, G1 = 3.5 + to 4+, = 3.5(7) to 4(7)= 24.5 to 28mm Take, G1 = 26mm Gap when ring is in the cylinder, Ga = 0.002 D to 0.004 D = 0.002 (200) to 0.004(20) = 0.4 to 0.8 mm Q5 = 0. 6 mm (H) Piston Borrel Radial depth of the piston sing Peces grooves, b = + +0.4 = 7 to 4 = 7.4 Maximum thickness of the bossel, t3 = 0.03D+b+4.9~~~ = · 0 · 63 × 200 + 7 · 4 + 4 · 9 = 17.9 43:00 18mm Piston wall thickness towards the i openiend, ty = 0.25t3 to 0.35t3 = 0.25 x 18 to 0.35 x 18 = 4.5-to 6.3 mm ty = 6mm Say (e) piston skirt Let 'l' is the length in mm. Maximum side trust due to gas pressure (p),

 $R = \mu \times \frac{110^{2}}{4} \times P$ $= 0.1 \times \frac{11 \times (200)^{2}}{4} \times 4 \left[\text{Take } \mu = 0.1 \right]$ = 12566-371 N Side thrust due to bearing Pressure on the piston basisely-treated allow steel and, Frake B=0.4 R = Pb x Dx & = 0.4 x200 x1 = 80 1 N :. \$ = \frac{12566 \cdot 371}{80} = 157.08 \cdot 6 Length of the piston 15, L= 1+(5t2+4b2)+b1 = 160+(5x6+4x5)+30 = 240 (f) piston pin do = outside diameter of the Pin, mm l, = Length of the pin is bush of the small end of the connecting rod in min and Pb, =30 N/mm2 (For bronze bushing) Load on pin due to bearing Pressure, = Pb, xdo xl = 30 x do x 1.5 x do [:1,=1.5do] = 45do N. Maximum load on piston due to gas pressure, = 103 x p = 11 x (200) x 4

6) 45 do = 125663.706 do = 52.84 ~ 70 mm Inside deameter, di = 0.6 do ..di = 0.6×70 = 42 mm Assuming pin is made of heat 0 = 140 MPa Maximum bending moment at the Centre of pin is given by, $m = \frac{P \cdot D}{8} = \frac{125663 \cdot 706 \times 200}{9}$.. m= 3141.59 × 103 N-mm Maximum bending moment, m= 17 \ do4-di4 05 $3141459\times10^{3} = \frac{17}{32} \left[\frac{(70)^{4} - (42)^{4}}{70} \right] q_{2}^{2}$ $Q_{b} = \frac{3141.59 \times 10^{3}}{293.69.202}$ = 107.185 N/mm 00 MPa % = 107.185 MPa is less than induced bending stress i.e., I 40 MPa 1 do = 70mm and di = 42 mm are satisfactory. > Design a cast iron trunktype piston for a single acting four Stocke empire developing 75kW per Cylinder when running at 600 s.p.m. The other available data is as follows, Maximum gas pressure = 4.8 Nami, Indicated mean effective Pressure = 0.65 N/mm3, Mechanical efficiency = 95%, Radius of Counk = 110mm, Fuel consumption = 0.3 kg/Bp/kg calorific value of fuel (higher) = 44 x 103 KJ/kg, Difference of temperature at the centre and edges of the piston Equating equations (1) and (2), we get, head = 200°c, Alberable Stress for

the material of the piston rings & Judgeon pin = 80 MPa, Allowable beasing pressure on the piston passel = 0.4 N/mmg and allocable bearing pressure on the gudgeon bil = 17 N/wwg. AM Given that, Power, B.P = 75KW = 75x10 W Speed, N = 600 r.p.m Maximum gas pressure, P=4.84mm Indicated mean effective pressure P: = 0.65 Nmma Crank radius, r= 110 mm Fuel consumption, m=0.3 kg/B-p/hr=0.3 kg/B.p/sec Higher Colorific value of fuel (H.cv) = 44 × 103 kg/kg, Temperature difference at the Centre and edges of the piston head, Tc-TE = 200°C Allowable stress for Piston material, 9, = 33.5MPa Allowable stress for piston ring & gudgeon pin, $Q_2 = 80 MRa$ Allowable bearing pressure on Piston baxell, Pb1 = 0.4 N/mm2 Allocable bearing pressure on Judgeon Pin, Pb2 = 17 N/mm3. For a 4-stroke engine, Number of working strokes per. minute, M= = = = 300 g. b.m Let, D- cylinder bore in mm 1 - Length of Stroke=1.50 (Assume)

Indicated power, I.p = Brake power Mechanical CPAciency I.p= 75×103 = 78947.3684 or 78.947KW But, I.p= PilxAxm 78947.368 = 0.65 × 1.50 × 40 0 × 300 $D = \sqrt{\frac{78947.368 \times 60 \times 4000}{0.65 \times 1.5 \times 11 \times 300}}$ D = 274.215 mm Say 275 mm · Bore of Cylhoder, D=275-1919 Length of Stroke, 1=1.5x275 = 412.5mm Length of Cylindes, L= 1.15 × 412.5 = 474.375 mm ≈ 475 mm. 1. Design of piston head (a) strength consideration Thickness of piston head, $-t_{H} = \sqrt{\frac{3PD^{2}}{16q_{2}}} = \sqrt{\frac{3\times4.8\times275^{2}}{16\times33.5}}$ · + ty = 45 mm (b) Thermal consideration Thickness of piston, +H = H 12.56k(TC-TE) Cross-Sectional area, $A = \frac{\pi}{4} D^2 = \frac{\pi}{4} x (275)^2 = 59395.736 mm^2$ Heat flowing through piston head is given by, H = CxHcvxm xB.P =0.05 × 44 × 103 × 0.3 ×75 Assuming, C=0.05, given, HCV = higher Calorific value for petrol=44x10 kJ/kg : H = 13.75 KW

= 13.75 x 103 W Thickness of piston, $t_{H} = \frac{H}{12.56 \times (T_{C} - T_{E}) \times k}$ $t_{\rm H} = \frac{13.75 \times 10^3}{12.56 \times 200 \times 1.6.6}$ (for cost from k=46.6 W/~/°c) # = 0.11/m = 117mm Taking larger of two values, 1.e., 120 mm. a. Radial Ribs Radial wibs may be 4 in number .. Thickness of the pribs Vanies 学のサー Thickness of ribs, tr= 120 to 120 = 40mm to 60mm Let us take, tr = 50mm 3. piston Rings Let, t, = Radial thickness. $-t_1 = D\sqrt{\frac{3\rho_M}{\sigma_2}}$ (Assume, PW = Wall Pressure for piston ring =0.04N/mm2 for petrol engine) Giveni at = 80 W/wwg $t_1 = 275 \sqrt{\frac{3x0.04}{80}}$ = 10.65mm = 11mm Axial thickness of piston rings, to =0.7+, to +1 =0.7×111011 =7.7 to 11mm

Taking, ta = 10mm. Also, minimum axial thickness of the Aston sing, $t_2 = \frac{D}{10 \, \text{km}}$ (Assuming mainter of sings, to = 275 = 6.875 mm Thus, the exial thickness of the piston ring as albeady calculated (le, to=10mm) is Satisfactory. width of the top Land, PI = # 40 1.5 # = 120 to 1.2 x 120 = 120mm to 144mm Taking, b = 130 mm width of the other sing lands (i.e., land between the grooves), ba =0.75ta to ta =0.75 × 10-6 10 = 75mm to 10mm Taking ba = 8mm Gap between the free ends of the G,=3.5+, to 4+, =3.5×11 to 4×11 = 3805 min to 44mm Taking, 61 = 40 mm Gap when the ring in the Cylinder is given by, Ga = 0.002 D to 0.004 D =0.000 x 275 to 0.004 to x 275 = 0.55mm to 1.1mm Taking, Ga = 0.8 mm 4. piston Barrel Depth of the Aston ring, b=+1+0.4

```
=11+0.4=11.4mm
Maximum thickness of the boosel,
     t3 = (0.03 D+b+4.05)~~
         =(0.03×275+11.4+4.5)
         = 24.15mm
piston wall thickness towards the
 open end,
     ty = 0.25 to 0.35 ts
        = 0.25 x24.15 to 0.35 x24.15
        = 6.0375 mm to 8:4525mm
 Taking, ty =5~~
5. piston Skirt
   Maximum Side, thoust,
         R = U \times \frac{\Pi}{L} \times D^2 \times P
        =0.1 x 1 x 275 x 5
                   (ASSume µ=0·1)
          = 29697.868 N
 Side thoust due to bearing
                ' pressure,
         R=Pb xDx1
           = 0.4× 275×l
   (: Booring pressure Pb is
        (formited 0.4 Nomma)
     . R=1101.
29697.868 = 1101.
         1 = 29697.868
         1 = 269.981 mm
:. skirt length,
        1= 269.981 mm
         270mm
.. Total length of the piston,
 L=Length of the Skirt+Length of
   the ring section + Topland.
   L= 1+(4+2+3b2)+b1
    =270+(4×10+3×8)+130
    = 464mm
  L = 470 mm (504)
```

6. piston pin Let, Pb - Bearing pressure at bushing = 17 N/mm2 for bronde buch do-auter diameter of the pin I, - Leigth of pin in the bush of the small end of the Connecting God in mm. .. Load on the pin due to bearing pressure, - Bearing pressure X Bearing area. $=Pb_1 \times do \times l_1$ = 17 x do x l, (Assume, l,=0.45D) = 17 x do x 0.45 x 275 = 2103.75 do N Maximum load on the piston due to gas pressure F= TD2×P = (275) x4.8 = 285099.533~ wow, equating maximum load on the piston to the load on the piston pin, 2103.75do = 285099.53] $d_0 = \frac{885099.533}{2103.75} = 135.519$: do = 136 ---Inside diameter of the pin, di =0.6xdo = 136x81.6=82mm checking the induced bending stress in blu $M = \frac{F \times D}{8} = \frac{885099.533 \times 375}{8}$ = 9800296-447 N/mm But, M= 1 [do4-di4] xq OP = TW [90 4] since, the induced bending stress is well below the permissible stress of 80 N/mm? Therefore

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given by,

th = H 12.56k (TC-TE) since the engine is fourstroke, the number of working strokes per minute, $m = \frac{N}{2} = \frac{2000}{2} = 1000$ Area of cross-section of the cylinder, $A = \frac{11}{4} D^2 = \frac{11}{4} \times (100)^2 = 7853.981$ Indicated horse power, I.p=Pm.LAn = 0.75 x 125 x 7853.981 x 1000 = 12271.845 N-m/s (64) W = 12.271 KW. Brake power, B.P=I.P × Momech =12.271 x0.8 = 9.816 KW Heat, flow through piston head is given by, H = C x HCV xm x B-P =0.05×42×103×41.7×10×9.86 Assuming C=0.05 = 0,859KW = 859.587W Therefore, thickness of piston had, $-t_{H} = \frac{H}{12.56 \, k(T_{C} - T_{E})}$ [For Cost from. K=46.6 W/m/2 and (TC-TE) = 220°C] = 6.675 mm

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.. Taking larger value of ty

Wall thickness or Radial Ribs

:. tH = 16 mm.

The thickness of the piston wall is given by, 型的型 : Thickness of wall the 15 to =5.333 mm Average thickness of wall, th = 6.666mm ~ 7. Piston Rings The radial thickness of piston ring is given by, $t_r = D \sqrt{\frac{3p_r}{\sigma_i}}$ $t_r = 100 \times \sqrt{\frac{3 \times 0.035}{90}}$ [: Assuming oradial wall pressure Pr=0.035 N/mm? & Tensile stress, of =90 N/min2]. tr=3.415~~ ~ 3.5 mm Axial thickness of the piston rings, ta = 0.7 tr totr =0.7x3.5 to3.5 = 2.45 to 3.5-mm Considering on average value, 1.e., =ta = 2.975 -m 23 -mm The minimum axial thickness of the piston ing, $t_a = \frac{D}{loxn_x}$ = loo [. Assuming number of rings, n= 4 = 9.2mm

Thus, the exial thickness of 3 mm is safe, width of the top land, b,=tu to 1.24 = 16 to 1.2 x 16 = 16-mm to 19.2-mm width of other sing boods, ba = 0.75 ta to ta = 0.75 x3 to 3 = 2.25 mm to 3 mm Consider, mm 81 2 mm 9.71=19 by = 2.625mm = 2.5mm. Gap between the free end of the sing, G1 = 3.5ty to 4tr $=3.5 \times 3.5 \pm 0.4 \times 3.5$ = 12:25 mm to 14 mm And the gap when the ring in the cylinder is given by, G2 = 0.002D to 0.004D =0.002 x100 to 0.004 x100 = 0.2 mm to 0.4 mm Consider, Gy = 13.125-mm 213-mm G2=0.3~~ Piston Barrel Depth of piston ring grooves, b=tr +0.4 = 3.5+0.4 =3.9mm Maximum thickness of bootel, ty = (0.030+b+4.5) =(0.03×100+3.9+4.5)

=11.4mm

```
Piston wall thickness towards the
, open end,
    tu = 0.25tb to 0.35tb
        = 0.25 x |1.4 to 0.35 x 11.4
        = 2.85mm to 3.99mm
 Considering am average value,
     tu = 3.42 mm ~ 3 mm
  Piston skirt
 The maximum Side thrust on
 the cylimder,
     6= mx # xDxxb
     = 0.1 x 1 x (100) 2 x 5 [: Assuming
                         H= 0.1
     = 3926.990 N
 Also, the side thrust on the
 piston barrel,
     R = Pb x D x l
       = 0.45 ×100 × Q
 ( Bearing Pressure Ph is limited
  to 0:45 N -mm?)
  Equating equations (1) 8(18),
        3926.990 = 0.45×100×1
             1 = 3926.990
0.65×100
                 =87.266 ~ 90 mm
      : skirt tegth, 1=90mm
  : Total length of the piston,
     L= length of the Skirt+
       length of the ring section+
              Top land.
     L= 1+(4xta+3xb2)+b1
       =90 +(4×3+3×2.5)+18
       =127.5mm ~ 128mm
```

biston bin let, Phy = Bearing pressure at bushing for bronde bush = 25 N mm2 do = aiter diameter of pin I = length of pin in mm Maximum load on the Piston Que to gas pressure, $=\frac{a}{4L}D_{S}\times b$ $=\frac{11}{20} \times (100)^{3} \times 5 = 39269.908 \text{ M}$ The load on the pin due to bearing Pressure, -Bearing pressure x Bearing $= P_b, \times do \times l,$ = 25 x dox (0.45 x loo) [: 1,=0.45D] =1125% Equating equations 3 & 6 39269.908=11256 do = 39269,908 = 34, 906mm ~ 35mm Inside diameter of the pin, , d; =0.6x do =0.6x35=2/mm checking the induced bonding Stress in Pin, M = P.D = 39269.908 x 1000 = 490873.85 N-mm Also, the maximum bending moment is given as, $M = \frac{\eta}{3a} \left[\frac{d_0 q - d_1^2 q}{d} \right] \times q$ 490873.85= 1 / (35)4-(21)4) of

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490873.85 = 3663.725 % $G = \frac{490873.85}{3663.725} = 133.982$ $G = 133.982 \text{ N/mm}^2$ Since, the induced bending stress is below the permissible stress of 140 N/mm², the design is

Design a piston for a four Stroke diesel engine Consuming 0.3kg of fuel per kw of power her hour and produces a brake mean effective pressure of the 20 munison of som NT.0 pressure inside the cylinder is 5 Name at a speed of 3500 r.p.m The cylinder diameter is required to the 300 mm with stroke 1.5times the diameter. The piston may have 4 compression rings and an oil ring. The following data can be used for design: Higher Calorific Value of fact = 46x103 kJ/kg; Temperature at the piston centre = 700°k; temperature at the piston edge = 475°k; Heat conductivity factor= 46.6 W/m/K; Heat conducted though top = 5% of heat produced; permissi ble tensile strength for the material of Piston = 27N mm2; pressure between rings and piston = 0.04 Mmm permissible tensile stress in rings = 80 M/mmg) posmissible pressure on piston basisel = 0-4 N/mm; Permissible pressure on piston pin = 15N mm3 permissible stress in Piston pin= 85 Name Any other data required for the design may be assumed. Govern that, Fuel consemption, m= 0.3 kg/kW/hr $=\frac{0.3}{3600}$ kg/kW/Sec Brake mean effective pressure,

Pm = 0.7N/mm²

Moximum gas pressure, P=5N/mm²

Speed, N=3500 rp.m

Cylinder diameter, D=300mm

Cylinder diameter, D=300mm

Stroke, L=1.5D=1.5x300=450mm

=0.45m

The piston has 4 compression rings

and an oil ring

Speed, N = 3500 r. p.m

Higher calorific value (H·Cv) = 46×10

KJ/kg

Temperature at the piston centre,

Tc = 700 k

Temperature at the piston end,

TE = 475 k

Tc-TE=700-475=225k
Permissible tensile strength for
piston material,

of = 27N/mm²

Pressure between rings and piston

Pu = 0.04 N/mm²

Permissible tensile stress for

piston rings,

(at) = 80 N mmg

permissible pressure on piston

Permissible pressure on piston

Pin, Pb = 15 N/mm²

Permissible stress in piston pin

= 85 N/mm².

1. Design of piston Head

(a) Strength consideration

Thickness of piston head, $t_{H} = \sqrt{\frac{3pD^{2}}{160}}$

 $-t_{H} = \sqrt{\frac{3x5x300^{2}}{16x27}} = 55.902$

chi Thermal consideration
Thickness of piston, the 12.56k(tc-TE)

Ans

HIE1 07 HI= 19 width of the top land, (i.e., ta = 10mm) ? 2 satisfactory. Pieton ring as already columbted Thus, the exical thickness off ends $uu = \frac{G \times 01}{G \times 0} = 61$ G= PW BBUZE to = 10xnt (Given neumber of the piston ring, to seambirt laixa muninimaelA mm01= 67 18013/10T monel of 4.8 = 61 of 61x F.o= けかけたの= み Axial thickness of piston sing, mm Pla + 11 = 40.0x8 / 00 = 15 Fmm 1008 = 70 army 40.0= (gah notely Assume, Pu = wall pressure for H= D/384 Let, t, - Radial thickness. ३% ग्रह्मां विकास wmos = 1 solat su tos mos at mose of the it = it adin to assorbird 前中時 sained adt adt to esenabilit : Radial vibes may be 5 in numbers. 3. Rodial Ribes Taking larger of two values, i.e., 60m ansidesation. prochedly wrong. Thus the design of piston is bosed on strength f) of conster of piston, in (9) The thickness of piston is greater

mungth= mgth.0= Hz. K = 46.6 W (2) 2°= 62 = 3T - 5T most \$200 voj ::) 13.56 x 23 5 x 46.6 ह्वार हाम महा = भुम inotated to assembling M = 01× 8 L + + + + = M> Ert・tel = =46×10 47/48 Given HCV = Higher colorific value (20.0=0, galon 22A) 364.843× E.O × 61× 84 × 20.0= H=C×Hev×m×B.p the wante of Heat Alocalns through piston head Broke power = 649.426 KM M POI - 264942 = = 1000. 10-w | ecc(09)M 5 men - CA 2601984949 = OGLIX 568-8890L XOE # X L.00 = B.p = Axxxxn = q.B 1 हिन क्राजिस हिल्ला है। भूग क्रिया क्रिया क्रिया है। Francis = 3000 = 50000 x 10 = 50 10 = A Losse Landhas-San (عماله (·· 耳后 a 4-5troke w.d. 1091= = 0000 = N=u "Number of working strokes per imfinite,

```
= 60 to 1.2×60
      = 60mm to 72mm
Taking it as b_1 = 55mm
width of the other ring lands
Ci.e., land between the grooves)
       ba =0.75ta to to
          =0.75 x 10 to 10
         .= 7.5 mm to 10 mm.
 Taking it as ba = 9 mm
Gap blu the free ends of the
sing,
   G1 = 3.5t, to 4t,
     =3.5 x35 to 4x12
     = 42mm to 68mm
Taking It as G1 = 45 mm
Gap when the sing in the glinder
is given by,
   Ga =0.002 D to, 0.004 D
       = 0.002 x300 to 0.04 to
                      × 300
       = 0.6mm to 1.2mm
 Taking it as 0.8 mm.
4. Piston Bossel
Depth of the piston sing,
      b=+1+0.4
        = 12+0.4
        = 12. 4mm
 Maximum theckness of the
  boorsel,
   t3=(0.03D+b+4.5)~m
      = (0.03×300+12.4+4.5)
   tz = 25.9 mm
Aston wall thickness towards
 the open end,
   t4 = 0.25t3 to 0.35t3
      = 0.25x 25.9 to 0.35x25.9
       = 6.475mm to 9.065mm
Taking it as ty=7-mm
```

```
5. Piston Skirt
  Maximum side thoust,
    R=HX TXD2xP
      =0.1x 1 x3002 x5 (ASSume
      = 35342,917N
side thrust due to bearing pressure,
        R=PoxDX1
         =0.4 ×300 XL
 (: Boxing pressure Pb is limited
          (Emm/N 4.0
    : R = 1201
  35342-917=1201
         1 = 35362.917 = 294.524mm
  : Skirt length,
        1 = 294.524 mm
          = 295~~~
  .. rotal length of the piston
        L = Length of the skirt +
 Length of the sight section + TOP
 land 1
  L=1+(4t2+3b2)+b1
  L = 295+(4×10+3×9)+55=417mm
  L= 417 mm(say)
6. piston pin
Let, Pb - Bearing pressure at bushing
  =15N mm3
   do - outer diameter of the pin
    l, - Leigth of pin in the bush
of the small end of the connecting
sod in mm.
: Load on the pin due to bearing
   pressure,
    =Bearing pressurex Bearing
    = Pb1 x do x l,
    = 15 x do x l, (Assuming, l=0.45)
```

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=15 x do x 0 45 x 300 : = 2025d. N Maximum load on the piston due to gas pressure, = 1 02 MP $=\frac{11}{12}300^{2}\times5=353429.174N$ now, equating maximum, load on the piston to the load on the piston pin, we get, 2025do = 353429.174 do = 353429.174 =174-532-mm = 175mm : do = 175 --Inside diameter of the pin, d: =0.6xd0 =0.6x175=105~~ thecking the induced bending Stress in Pin, $M = \frac{PXD}{8} = \frac{353429.174 \times 300}{8}$ = 13253594.03 N-mm But, M = 17 [de 4-di 4] x 02 $O_b = \frac{32m}{n} \left[\frac{do}{do^4 - di^4} \right]$ = 135000000 2 x0 000000014 Emin (4 4 P. 86 = P : Since, the induced bending

Stress is well below the permissi--ble stress of 85 N/mm? . Therefore the design is Safe

> Design a cost iron piston for a Simple acting four stroke diesel (10) engine with the following data ajtinder bosse = 300mm Length of Stroke = 250mm speed = 600 spm Brake mean effective pressure= Maximum gas pressure = 4Mpa. fuel consumption = 0.25kg Pex Bp I/d ratio for bush in small end of Connecting rod = 1.5. Assume Suitable data if required and state the assumptions made. 世 Given that, Cylinder bore, D=300 mm Stroke, L= 250mm=0.25 m speed, N = 600 pm Brake mean effective pressure, Pm = 0.6 MPa = 0.6 N/mm? Waximum gas bressise, p=4MPa =40/2002 fuel consumption m = 0.25 kg | Bplb= = 0.25 = 69.45×10 kg/Bp. Per Second. 1 = 1.5 ⇒ l = 1.5do .. The dimensions for vasious components of the piston are determined as follows. 1. piston Head (1) Thickness on the bosis of Strength, th = \(\frac{3PD^2}{16.00}\)

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1. ty = 41.079 = 42 --number of working strokes per minute, $n = \frac{N}{2} = \frac{600}{2} = 300$ Area of the colimder, $A = \frac{11}{4} \times D^2 = \frac{11}{6} \times (300)^2 = 70685-834$ Booke power, Bop = Pin LAm = 0.6 x 0.25 x 70685.834 × 300 = 53.014 KW Heat flowing through piston head, H= CxHCVxmxBxP =0.05 x 44 x 103 x 69. 45 x 106 x 53.014 [Assume, C= 0.05 and Hov= Gexx10 K] 4 H = 8100.009 W or 8.1KW ii Thickness based on dissipation, [for cost from / K = 46. 6 W/m/2 and TC-TE = 220'C] : +H = 8.1 × 103 th = 0.063 m 08 63 mm Take larger of two values, i.e., t# = 63 mm 2. Radial Ribs Number of sibs = 4 thickness of ribs, to = 63 to 63 = 21 to 31.5 mm Taking to = 26mm 3. Piston Rings Assume number of sings, n=51.4, three compression and two are oil

vings radial thickness of piston sivas, $t_1 = D \sqrt{\frac{2bn}{3bn}} = 300 \times \sqrt{\frac{3 \times 0.000}{3 \times 0.000}} = 10.3$ ≈11mm [Assume, radius pressure, PN = 0:04 N/mm2 and Permissible tensile stress of Piston sings, of = 100 Wmm2 Axial thickness of the piston to = 0.74 to to = 0.7(11) to 11 = 7.7 to 11mm Take, to = 10mm Minimum axial thickness of the biston sing, (f) win = 10xxx = 300 = 6mm (Assume, Nx=5) Since ita>(ta)min, axial thickness is satisfactory. width of the top land, b = tH to liatH = 63 to 1.2 x 63 = 63 to 75.6mm 4 p1=68 width of the other sing lands, ba = 0.75 to to to =0.75(10) to 10 = 0.7.5 to lomm .. by = 8 mm Gap between free ends, G1 = 3.5+1 to 4+1

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=3.5(11) to 4(11)

```
= 38.5 to 44mm
1: G' = ctom
 Gap, when ting is in the
                    Cyli-moles,
        62 =0,002 D to 0.004D
          =0,002 (300) to 0,004
                       (300)
          = 0.6 to 1.2 mm
     ·: 62 = 1 mm
 4. Piston Bossel.
  Radial depth of the piston
  ting grooves,
      b= 4 +0.4
        =11+0.4
   : b= 11.4 mm
maximum thickness of the boore
  to = 0.03 (D) +b+49mm
      = 0.03×300+11.4+4.9
      = 9+11.4+4.9=25.3~~
: 412 26mm
Piston wall thickness towards
the open end,
    ty=0.25t3 to 0.35t3
       = 0.95 x26 to 0.35 x26
       = 6.5 to 9.1mm
   : + = 8 mm
5 piston skirt
Let, I be the length of the
 Skirt in mm.
Maximum Side thoust due to
gas bressure (b)
       R=LLX ID2 XP
```

= 28274.334 N Side thoust due to bearing pressure on the piston boorel, P. R = POXDXL = 0.4 x 300 x l (Assume, Pb= 0.4 MPa] 28274.334 = 1201 N 120 = 28274.334 = 235.619 2 syomm length of thepiston, eight of the Skirt-Leight of the ring Section + Top land L= 1+(3+4b)+61 = 240+(5x10+4x8)+68 - = 390 mm According to empirical formula, L = D to 1.5D =300 to 1.5 x 300 = 300 to 450 mm * D<L<1.5D 6. piston Pin Let, . do - outside diameter of the bird in man li= length of the pininthebush of the small and of the Connecting rod in mm Pb - Bearing pressure at bushing (for bronze bushing) =30N/ma, Load on the pin due to bearing Pressure, =Pxdox1, = 30 xdo x1.5 xdo [:1=1.5do] $=45\times d_0^2 N$ =0.1 x 11x600) 4 (Assume H=0.1) maximum load on the Piston

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Proside diameter = 70 mm. .. out side diameter = 100 mm いいのでからこうか ED-19119866= 484862117 -4(001)=4(6月) 4(16)-4(00) = (B.84884)1FF 1 (16) -100) = (CX001X LO1X + (8.600) $\frac{1}{2} \left(\frac{1}{2} \right) \left(\frac{1$ Phib-Pob II = dM tramon for basdminixon ted mmcr Eax 258-60201 = 00CXCE. 54 FEBG = 0x7 = 24 the centre of the pin, totremen bending menical of = Thomba treated allog Steel toanto abom is mig primuest my 00] = 00 ... more de los mon 2: "B wolled rig satt sylem of mm 08 - T36.PT = 06 1. 45 do = 282743.33 (Dbno (Driothorpo Britanga) = 288743:339 pp. 1 4 x (000) x 11 = 9 x 6(11) = due to gas preseure,

DESIGN OF CURVED BEAMS

Introduction

For a straight beam, the bending equation $\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$ has been derived and seen in

Mechanics of solids. This equation can be applied, with sufficient accuracy, to the beams or bars having small initial curvature. However the machine members such as crane hooks, chain links and rings etc. which are having large initial curvature, the simple bending equation cannot be used. This chapter deals with the bending of curved bars, which are having large initial curvature. Actually curved bars means the bars of large initial curvature.

Generally, if the radius of curvature is more than 5 times the depth of the beam, the beam is said to be having small initial curvature. But if radius of curvature is less than 5 times the depth, the beam is said to be having large initial curvature. Hence for large initial curvature, the radius of curvature is small. Also for curved beams, the neutral and centroidal axes do not coincide.

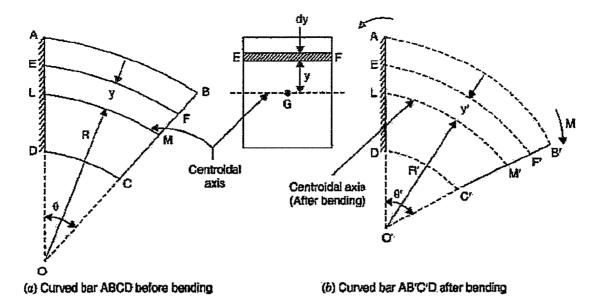
Assumptions made in the derivation of stresses in a curved beam:

Before deriving the expression for the stresses developed in a curved bar when it is subjected to some bending moments, the following assumptions are made:

- 1. Transverse sections which are plane before bending remain plane after bending.
- 2. Hook's law is applicable. This means the working stresses are below the limit of proportionality.
- 3. The longitudinal fibres of the bar, parallel to centroidal axis exert no pressure on each other. This means the distance between any longitudinal fibre from centroidal axis is same before and after bending.
- 4. Each layer of the beam is free to expand or contract, independently of the layer, above or below it.

Expression for stresses in a curbed beam:

The following Fig. shows the two positions of a curved bar, one position [i.e., Fig. (a)] is before bending whereas the second position [i.e., Fig. (b)] is after bending when some moment M is applied at the ends of the bar. The centroidal axis is shown by line LM and LM' in the two positions respectively.



Consider any fibre EF at a distance 'y' from the centroidal axis LM. This fibre takes the position of EF' when the moment M is applied. Then the fibre EF' will be at a distance .of 'y' from the centroidal axis LM'.

Refer to above Fig. (a). Here:

R = radius of curvature of centroidal axis LM,

 θ = angle subtended by centroidal axis LM at the centre of curvature O,

y =distance of fibre EF from centroidal axis LM.

Refer to Fig. (b). Here:

M = uniform bending moment applied to the bar.

R' = radius of curvature of centroidal axis LM'.

 θ' = angle subtended by centroidal axis LM' at the centre of curvature θ' .

y' = distance of fibre EF' from centroidal axis LM'.

 σ = stress in the strained fibre EF' due to bending moment M, and Let

e = strain in the same fibre EF'.

 e_0 = strain in centroidal layer LM.

Now, from Figs. (a) and (b), we have
$$EF = (R + y) \times \theta \quad \text{and} \quad EF' = (R' + y') \times \theta'$$
Also
$$LM = R \times \theta \quad \text{and} \quad LM' = R' \times \theta'$$

$$\therefore \quad \text{Strain,} \quad e = \frac{EF' - EF}{EF} = \frac{(R' + y') \times \theta' - (R + y) \times \theta}{(R + y) \times \theta}$$

$$= \frac{(R' + y') \times \theta'}{(R + y) \times \theta} - \frac{(R + y) \times \theta}{(R + y) \times \theta} = \frac{(R' + y')}{(R + y)} \times \frac{\theta'}{\theta} - 1$$
or
$$1 + e = \frac{(R' + y')}{(R + y)} \times \frac{\theta'}{\theta} \qquad ...(i)$$
Also, strain,
$$e_0 = \frac{LM' - LM}{LM} = \frac{R' \times \theta' - R \times \theta}{R \times \theta} = \frac{R'}{R} \times \frac{\theta'}{\theta} - 1$$
or
$$1 + e_0 = \frac{R'}{R} \times \frac{\theta'}{\theta} \qquad ...(ii)$$

...(ii)

or

Dividing equation (i) by equation (ii), we get:

$$\frac{1+e}{1+e_0} = \frac{(R'+y')}{(R+y)} \times \frac{\theta'}{\theta} / \left(\frac{R'}{R} \times \frac{\theta'}{\theta}\right)$$

$$= \frac{(R'+y')}{(R+y)} \times \frac{R}{R'} \qquad \left(\text{Cancelling } \frac{\theta'}{\theta} \text{ from numerator and denominator}\right)$$

$$= \frac{\left(\frac{R'+y'}{R'}\right)}{\left(\frac{R+y}{R'}\right)} = \frac{1+\frac{y'}{R'}}{1+\frac{y}{R}}$$
or $e = (1+e_0) \left[\frac{1+\frac{y'}{R'}}{1+\frac{y}{R}}\right] - 1$

$$= \frac{(1+e_0)\left(1+\frac{y'}{R'}\right) - \left(1+\frac{y}{R}\right)}{\left(1+\frac{y}{R}\right)} = \frac{1+\frac{y'}{R'} + e_0 + e_0 \times \frac{y'}{R'} - 1 - \frac{y}{R}}{\left(1+\frac{y}{R}\right)}$$

$$= \frac{\frac{y'}{R'} + e_0 + e_0 \frac{y'}{R'} - \frac{y}{R}}{\left(1+\frac{y}{R}\right)} = \frac{\frac{y'}{R'} + e_0 \frac{y'}{R'} + e_0 - \frac{y}{R}}{\left(1+\frac{y}{R}\right)}$$

But according to assumption (3), y = y'

4,

$$e = \frac{\frac{y}{R'} + e_0 \frac{y}{R'} + e_0 - \frac{y}{R}}{\left(1 + \frac{y}{R}\right)}$$

$$= \frac{\frac{y}{R'} + e_0 \frac{y}{R'} + e_0 - \frac{y}{R} + e_0 \frac{y}{R} - e_0 \frac{y}{R}}{\left(1 + \frac{y}{R}\right)}$$

$$= \frac{e_0 + e_0 \frac{y}{R} + \frac{y}{R'} - \frac{y}{R} + e_0 \frac{y}{R'} - e_0 \frac{y}{R}}{\left(1 + \frac{y}{R}\right)}$$

$$= \frac{e_0 \left(1 + \frac{y}{R}\right) + y\left(\frac{1}{R'} - \frac{1}{R}\right) + e_0 y\left(\frac{1}{R'} - \frac{1}{R}\right)}{\left(1 + \frac{y}{R}\right)}$$

$$= \frac{e_0 \left(1 + \frac{y}{R}\right) + \left(\frac{1}{R'} - \frac{1}{R}\right) (y + e_0 y)}{\left(1 + \frac{y}{R}\right)}$$

$$= \frac{e_0 \left(1 + \frac{y}{R}\right) + \left(\frac{1}{R'} - \frac{1}{R}\right) (y + e_0 y)}{\left(1 + \frac{y}{R}\right)}$$

$$= e_0 + \frac{\left(\frac{1}{R'} - \frac{1}{R}\right) (y + e_0 y)}{\left(1 + \frac{y}{R}\right)} = e_0 + \frac{y(1 + e_0)\left(\frac{1}{R'} - \frac{1}{R}\right)}{\left(1 + \frac{y}{R}\right)} \dots (iii)$$

The equation (iii) gives the expression for the strain produced in fibre EF under the action of moment M.

Now the expression for stress produced in the fibre EF can be obtained.

We know that
$$\frac{\text{Stress}}{\text{Strain}} = E$$

 $\text{stress} = E \times \text{strain}$
 $\sigma = E \times e$

Or

or

$$=E \times \left[e_0 + \frac{y(1+e_0)\left(\frac{1}{R'} - \frac{1}{R}\right)}{\left(1 + \frac{y}{R}\right)}\right] \qquad \dots (iv)$$

The above stress is produced due to bending moment M. Hence this stress is known as bending stress.

Consider a small strip of area dA at a distance y from centroidal axis LM.

The force on the strip = stress \times area of strip = $a \times dA$

Moment of this force about centroidal axis = Force x y

$$=(\sigma \times dA) \times y$$

The above expression gives the resisting moment offered by the strip. Total resisting moment is obtained by integrating the above equation.

 $\therefore \text{ Total resisting moment} = \int (\sigma \times dA \times y)$

For equilibrium, the total resisting moment must be equal to applied moment M.

$$M = \int G \times dA \times y \quad \text{or} \quad \int G \times y \times dA$$

$$= \int E \times \left[e_0 + \frac{y(1 + e_0) \left(\frac{1}{R'} - \frac{1}{R} \right)}{\left(1 + \frac{y}{R} \right)} \right] \times y \times dA$$

$$\left[\text{From equation (iv)}, \quad G = E \times \left[e_0 + \frac{y(1 + e_0) \left(\frac{1}{R'} - \frac{1}{R} \right)}{\left(1 + \frac{y}{R} \right)} \right] \right]$$

$$Ab \frac{\left(\frac{1}{H} - \frac{1}{H}\right)_{(0} + L)_{V}}{\left(\frac{1}{H} + \frac{1}{H}\right)} \int \underline{A} + Ab_{0} \cdot \underline{A} = \frac{1}{H} \times e_{0} \cdot \underline{A} + \underline{E}(\underline{1} + e_{0}) \cdot \underline{A} + \underline{A} \times e_{0} \times \underline{A} = \frac{1}{H} \cdot \underline{A} \cdot \underline{A} + \underline{A} \cdot \underline{A} \cdot \underline{A} \cdot \underline{A} + \underline{A} \cdot \underline{A} \cdot \underline{A} \cdot \underline{A} \cdot \underline{A} \cdot \underline{A} \cdot \underline{A} \cdot \underline{A} + \underline{A} \cdot$$

where A = Area of cross-section of the beam.

But for pure bending, there is no force on the section of the beam.

10

$$Ab\frac{\chi}{\left(\frac{\chi}{H}+L\right)}\left(\left(\frac{1}{H}-\frac{1}{\sqrt{H}}\right)(_0^3+L)\overline{A}+\Lambda\times_0^3\times\overline{A}=0$$

. Ab $\frac{\chi}{\left(\frac{\chi}{\pi} + 1\right)}$ lo sules of the won au tol

We know that $\int \frac{v^{\alpha}}{\left(\frac{Y}{X} + L\right)} dA = Ah^{\frac{\alpha}{2}}$ from equation (v). Hence let us use this value to find

$$Ab \frac{v}{\left(\frac{v}{A} + 1\right)}$$

$$VP \frac{\left(\frac{x}{K} + 1\right)}{\left(\frac{x}{K} + 1\right)} \int -VP \frac{\left(\frac{x}{K} + 1\right)}{K} \int$$

Totanemun sait of $\frac{x_C}{3}$ gainstaidus bas gaibhA

$$Ab \left[\frac{s_{\chi}}{\left(\frac{\kappa}{4} + L\right)} \times \frac{1}{4} - \kappa \right] = Ab \frac{\left[\frac{s_{\chi}}{4} - \left(\frac{\kappa}{4} + L\right)\kappa\right]}{\left(\frac{\kappa}{4} + L\right)} =$$

$$Ab \frac{c_{\zeta}}{\left(\frac{c_{\zeta}}{H} + L\right)} \times \frac{1}{H} \left[-Ab \, c \right] =$$

$$\left[0-VP^{\mathcal{L}}\int \phi\right]$$

$$\Delta \Delta \frac{2\sqrt{c}}{\left(\frac{C}{M}+L\right)} \int \frac{1}{M} - 0 = 0$$

$$\begin{bmatrix} s_{t}A = Ab \frac{s_{t}}{\left(\frac{c}{A} + 1\right)} \end{bmatrix} :$$

$$a_{\overline{A}} \sim \frac{1}{\overline{A}} - =$$

$$\left[{}^{2}AA = Ab \frac{{}^{2}V}{\left(\frac{Y}{A} + I \right)} \right] :$$

Substituting the value of
$$\int \frac{Nb \cdot V}{\left(\frac{N}{R} + L\right)}$$
 in equation (vii), we get

$$\mathbb{E}(1+e_0)\left(\frac{1}{R}-\frac{1}{R}\right)\times\left(\frac{1}{R}-\frac{1}{R}\right)\times\left(\frac{1}{R}-\frac{1}{R}\right)\times\left(\frac{1}{R}-\frac{1}{R}\right)\times\left(\frac{1}{R}-\frac{1}{R}\right)$$

or

$$(1+e_0)\left(\frac{1}{R'}-\frac{1}{R}\right)\times\frac{h^2}{R}=e_0 \qquad \qquad [\text{Cancelling $E\times A$ to both sides}]$$

$$(1+e_0)\left(\frac{1}{R'}-\frac{1}{R}\right)=\frac{e_0\times R}{h^2} \qquad \qquad ...(viii)$$

Substituting the value of $(1 + e_0) \left(\frac{1}{R'} - \frac{1}{R} \right)$ in equation (vi),

We get

**

$$\begin{split} M &= E \times \frac{e_0 \times R}{h^2} \times A \times h^2 \\ &= E \times e_0 \times R \times A \\ e_0 &= \frac{M}{E \times R \times A} \end{split} \qquad ...(ix)$$

OF

From equation (iv) it is clear that e_0 is constant for a given value of M, A and R.

Now the expression for stress (o) can be obtained by substituting the value of $(1 + e_0)$ $\left(\frac{1}{R'} - \frac{1}{R}\right)$ from (viii) in equation (iv).

$$\sigma = E \times \left[e_0 + \frac{y \times \left(\frac{e_0 \times R}{h^2} \right)}{\left(1 + \frac{y}{R} \right)} \right]$$

$$\left[\because (1 + e_0) \left(\frac{1}{R'} - \frac{1}{R} \right) = \frac{e_0 \times R}{h^2} \text{ from equation (viii)} \right]$$

$$= E \times e_0 \left[1 + \frac{y \times R}{h^2} \right]$$

$$= E \times \frac{M}{RRA} \left[1 + \frac{y \times R}{h^2} \right]$$

$$= E \times \frac{M}{RRA} \left[1 + \frac{y \times R}{RRA} \right]$$

$$\therefore \text{ From equation (ix) } e_0 = \frac{M}{RRA}$$

$$= E \times \frac{M}{ERA} \left[1 + \frac{\left(\frac{y \times R}{h^2} \right)}{\left(\frac{R+y}{R} \right)} \right] \qquad \left[: \text{ From equation (ix) } e_0 = \frac{M}{ERA} \right]$$

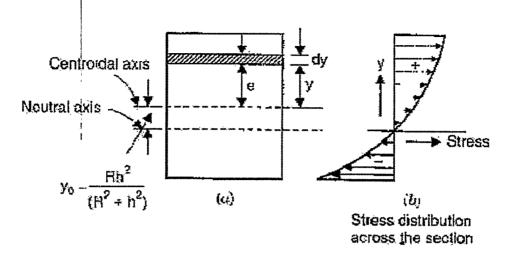
$$=\frac{M}{R\times A}\left[1+\frac{R^2}{h^2}\left(\frac{y}{R+y}\right)\right] \qquad ...(23.1)$$

The equation (23.1), gives the expression for tensile stress. The value of 'y' will be -ve, on the downward side of centroidal axis *LM*. Hence stress will be opposite to the stress given by equation (23.1).

$$\therefore \quad \text{o(compressive)} = \frac{M}{RA} \left[1 + \frac{R^2}{h^2} \left(\frac{-y}{R - y} \right) \right]$$

$$= \frac{M}{RA} \left[1 - \frac{R^2}{h^2} \times \left(\frac{y}{R - y} \right) \right] \qquad \dots(23.2)$$

The expression for the stress given by equation (23.1) or (23.2) is known as *Winkler-Bach Formula*. The distribution of the stress given by equation (23.1) or (23.2) is hyperbolic (and not linear as in the case of straight beams). The stress distribution is shown in following Fig.



Position of Neutral Axis. Let y_0 = distance of neutral axis from centroidal axis. The stress (c) is zero at the neutral axis. Hence position of neutral axis is obtained by substituting $\sigma = 0$ and $y = y_0$ in equation (23.1).

$$0 = \frac{M}{R \times A} \left[1 + \frac{R^2}{h^2} \left(\frac{y_0}{R + y_0} \right) \right] = 0 \quad \text{or} \quad 1 + \frac{R^2}{h^2} \left(\frac{y_0}{R + y_0} \right) = 0$$
or
$$\frac{R^2}{h^2} \left(\frac{y_0}{R + y_0} \right) = -1 \quad \text{or} \quad R^2 y_0 = -h^2 \times R - h^2 \times y_0$$
or
$$y_0(R^2 + h^2) = -h^2 \times R \quad \text{or} \quad y_0 = -\frac{R \times h^2}{R^2 + h^2} \qquad \dots(23.3)$$

But 'y' is the distance from centroidal axis. As y is -ve, this means the neutral axis will be at a distance of 'y' below the centroidal axis. Fig. (a) Shows the position of neutral axis and Fig. (b) Shows the stress distribution.

Sign Convention. The stress across the cross-section will be tensile or compressive according to the following conventions:

- 1. The bending moment M will be positive if it increases the curvature of the beam and negative if it decreases the curvature. (Increase of curvature of beam will decrease the radius of curvature).
- 2. The distance 'y' is positive if it is measured from the centroidal axis towards the convex side of beam, and negative when measured towards the concave side (or towards the centre of curvature).
- 3. The positive stress means tensile stress, whereas negative stress means compressive stress.

Determination of factor 'h2', for various sections

From equation (v), we have

$$\int \frac{y^2 dA}{\left(1 + \frac{y}{R}\right)} = Ah^2$$

or

$$h^2 = \frac{1}{A} \int \frac{y^2}{\left(1 + \frac{y}{R}\right)} dA = \frac{1}{A} \int \frac{Ry^2}{R + y} dA = \frac{R}{A} \int \frac{y^2}{(R + y)} dA$$

There is some difficulty in integrating the term $\frac{y^2}{(R+y)}$, as y is existing in numerator and denominator. Hence y^2 should be illiminated from numerator so that integration can be done easily. By adding and subtracting R^2 to y^2 , integration becomes easy.

$$h^{2} = \frac{R}{A} \int \left(\frac{y^{2} + R^{2} - R^{2}}{R + y} \right) dA = \frac{R}{A} \int \left(\frac{y^{2} - R^{2}}{(R + y)} + \frac{R^{2}}{R + y} \right) dA$$

$$= \frac{R}{A} \int \left[\frac{(y - R)(y + R)}{(R + y)} + \frac{R^{2}}{R + y} \right] dA = \frac{R}{A} \int \left[(y - R) + \frac{R^{2}}{R + y} \right] dA$$

$$= \frac{R}{A} \left[\int y dA - \int R dA + \int \left(\frac{R^{2}}{R + y} \right) dA \right]$$

$$= \frac{R}{A} \left[0 - R \times A + \int \frac{R^{2}}{(R + y)} dA \right]$$

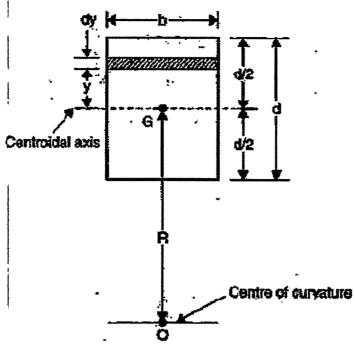
$$= -R^{2} + \frac{R}{A} \int \left(\frac{R^{2}}{R + y} \right) dA$$

$$= \frac{R^{3}}{A} \int \left(\frac{1}{R + y} \right) dA - R^{2} \qquad ...(23.4)$$

By substituting the values of 'y' and dA for various section in equation (23.4), the value of h^2 is obtained.

1) Rectangular Section:

A rectangular section of a curved beam of width b' and depth'd' is shown in Fig. The centre of curvature of the curved beam is at O. The radius of curvature from centroidal axis is R. Consider an elementary strip of width b' and depth dy' at a distance y' from the centroidal axis.



Now area of strip, $dA = b \times dy$ Area of section, $A = b \times d$ From equation (23.4), we have

$$h^{2} = \frac{R^{3}}{A} \int \left(\frac{1}{R+y}\right) dA - R^{2} = \frac{R^{3}}{b \times d} \int_{-d/2}^{*d/2} \frac{1 \times b \times dy}{(R+y)} - R^{2}$$

$$(2) \quad dA = b \times dy)$$

$$= \frac{R^{3} \times b}{b \times d} \int_{-d/2}^{*d/2} \left(\frac{1}{R+y}\right) dy - R^{2} = \frac{R^{3}}{d} \left[\log_{e}(R+y)\right]_{-d/2}^{d/2} - R^{2}$$

$$= \frac{R^{3}}{d} \left[\log_{e}\left(R + \frac{d}{2}\right) - \log_{e}\left(R - \frac{d}{2}\right)\right] - R^{2}$$

$$= \frac{R^{3}}{d} \left[\log_{e}\left(\frac{(R+d/2)}{(R-d/2)}\right) - R^{2} = \frac{R^{3}}{d} \left[\log_{e}\left(\frac{(2R+d)}{(2R-d)}\right) - R^{2} \right] \dots (23.5)$$
Also
$$\log_{e}\left(\frac{2R+d}{2R-d}\right) = \log_{e}\left(\frac{R+d/2}{R-d/2}\right) = \frac{d}{R}\left[1 + \frac{1}{3}\left(\frac{d}{2R}\right)^{2} + \frac{1}{5}\left(\frac{d}{2R}\right)^{4} + \dots\right]$$

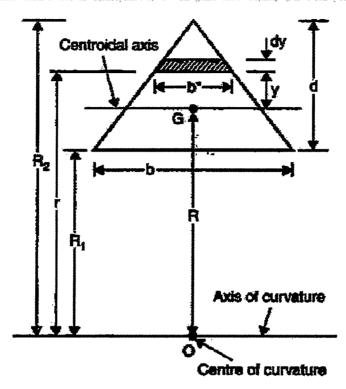
$$= R^{2}\left[1 + \frac{1}{3}\left(\frac{d}{2R}\right)^{2} + \frac{1}{5}\left(\frac{d}{2R}\right)^{4} + \dots\right] - R^{2}$$

$$= R^{2}\left[1 + \frac{1}{3}\left(\frac{d}{2R}\right)^{2} + \frac{1}{5}\left(\frac{d}{2R}\right)^{4} + \dots\right] - R^{2}$$

$$= R^{2}\left[\frac{1}{3}\left(\frac{d}{2R}\right)^{2} + \frac{1}{5}\left(\frac{d}{2R}\right)^{4} + \dots\right] - R^{2}$$

2. Triangular Section

A triangular section of a curved beam of width 'b' and height 'd' is shown in Fig. The centre of curvature of the curved beam is at O. The radius of curvature from the centroidal axis is R. Consider a strip of width b^* and depth 'dy' at a distance 'y' from the centroidal axis and also at a distance 'r' from the axis of curvature.



and

.

$$r = R + y$$
$$dr = 0 + dy = dy$$

Let R_i = distance between base of triangle and axis of curvature. R_2 = distance between apex of triangle and axis of curvature

Area of triangle,

$$A=\frac{b\times a}{2}$$

Area of strip,

$$dA = b^* \times dy$$

$$= \frac{b}{d}(R_2 - r) \times dy = \frac{b}{d}(R_2 - r) \times dr \qquad (v \quad dy = dr)$$

where b^* is obtained from

$$\frac{b^*}{b} = \frac{(R_2 - r)}{d}$$
$$b^* = \frac{b}{d}(R_2 - r)$$

or

Now from equation (23.4), we have

$$h^{2} = \frac{R^{3}}{A} \int \frac{dA}{(R+y)} - R^{2}$$

$$= \frac{R^{3}}{\left(\frac{b \times d}{2}\right)} \int \frac{\frac{b}{d} (R_{2} - r) dr}{r} - R^{2} \qquad (\because R + y = r)$$

(limit of integration will be according to dr in which r is the distance from axis of curvature. Hence limits of integration will be from R_1 to R_2)

$$=\frac{2R^3}{b\times d}\times \frac{b}{d}\int_{R_1}^{R_2}\left(\frac{R_2}{r}-1\right)dr-R^2$$

Problem-2:

Determine: (i) Location of neutral axis, (ii) Maximum and minimum stresses.

When a curved beam of trapezoidal section of bottom width 30 mm, top width 20 mm and height 40 mm is subjected to pure bending moment of + 600 Nm. The bottom width is towards the centre of curvature. The radius of curvature is 50 mm and beam is curved in a plane parallel to depth. Also plot the variation of stresses across the section.

Sol. Given:

 $b_1 = 30 \text{ mm}, b_2 = 20 \text{ mm}, d = 40 \text{ mm}, M = +600 \text{ Nm} = 600 \times 1000 \text{ Nmm}$ R = 50 mm.

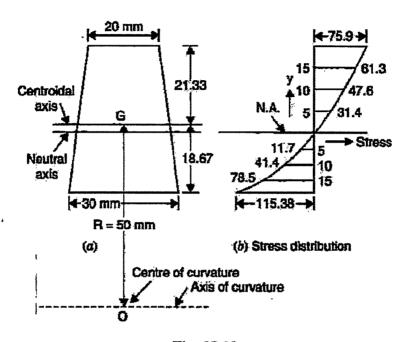


Fig. 23.10

Let us first find the location of centroidal axis. The distance d_1 of centroidal axis from the side b_1 is given by,

 $d_1 = \left(\frac{b_1 + 2b_2}{b_1 + b_2}\right) \times \frac{d}{3} = \left(\frac{30 + 2 \times 20}{30 + 20}\right) \times \frac{40}{3} = \frac{70}{50} \times \frac{40}{3} = 18.67 \text{ mm}$ $d_2 = 40 - 18.67 = 21.33 \text{ mm}$

and

Let us now find R_1 , R_2 and area A.

$$R_1 = R - d_1 = 50 - 18.67 = 31.33 \text{ mm}$$

 $R_2 = R + d_2 = 50 + 21.33 = 71.33 \text{ mm}$

A =Area of trapezoidal section

$$=\left(\frac{b_1+b_2}{2}\right) \times d = \left(\frac{30+20}{2}\right) \times 40 = 1000 \text{ mm}^2$$

Let us now find h^2 for trapezoidal section. Using equation (23.7), we get

$$\begin{split} h^2 &= \frac{R^3}{A} \Bigg[\bigg\{ b_2 + \bigg(\frac{b_1 - b_2}{d} \bigg) \times R_2 \bigg\} \log_e \bigg(\frac{R_2}{R_1} \bigg) - (b_1 - b_2) \Bigg] - R^2 \\ &= \frac{50^3}{1000} \left[\bigg\{ 20 + \frac{30 - 20}{40} \times 71.33 \bigg\} \log_e \bigg(\frac{71.33}{31.33} \bigg) - (30 - 20) \right] - 50^2 \\ &= 125 \left[\{20 + 17.83\} \times 0.8227 - 10 \right] - 2500 = 140.34 \end{split}$$

(i) Location of neutral axis. Let y_0 = distance of neutral axis from centroidal axis. The location of neutral axis is given by equation (23.3) as

$$y_0 = -\frac{R \times h^2}{R^2 + h^2} = -\frac{50 \times 140.34}{50^2 + 140.34} = -2.65 \text{ nm. Ans.}$$

-ve sign means the neutral axis will be below the centroidal axis at a distance of

.mm 39.S

(ii) Maximum and minimum stresses. The stress at any layer at a distance 'y' from

entroidal axis is given by equation (23.1) as

$$\left[\left(\frac{V}{A + A} \right) \frac{2A}{h^2} + 1 \right] \frac{M}{A \times A} = 0$$

The stress will be maximum at the extreme bottom layer where

mm
$$78.81 -= x$$

$$\left[\left(\frac{73.81 - 05}{50.000,000} + 1 \right) \frac{203}{0001 \times 03} = \frac{300.000}{5001}$$

 $\sin A = 10.61 + (-10.61) = 10.88 \text{ N/mm}^2$

= 115.38 M/mm^2 (compressive)

Minimum stress occurs at the extreme top layer where y = 21.33 mm

$$\Omega_{\text{mint}} = \frac{600,000}{50 \times 1000} \left[1 + \frac{50^2}{140.34} \times \left(\frac{21.33}{50 + 21.33} \right) \right]$$

- 12 (1 + 5.326) = 75.9 M/mm³. (tensile)

noitose shi esoron eseestes the section

$$\left[\left(\frac{\chi}{\chi + 0\delta} \right) \frac{s_0 \delta}{4 \kappa_0 A_1} + I \right] \frac{00000}{0001 \times 0\delta} = \left[\left(\frac{\chi}{\chi + \mathcal{H}} \right) \frac{s_{\mathcal{H}}}{s_{\mathcal{H}}} + I \right] \frac{M}{\Lambda \times \mathcal{H}} = 0$$

$$\left[\left(\frac{\chi}{\chi + 0\delta} \right) \times 8.71 + I \right] \Omega I = 0$$

Fig. 23.10 (b) shows the stress distribution across the section.

Problem-3:

Determine: (i) position of neutral axis, and (ii) maximum and minimum stresses when a curved beam of circular section of diameter 100 mm is subjected to pure bending moment of + 11.5 kN m. The radius of curvature is 100 mm.

Sol. Given:

d = 100 mm, B.M., M = +11.5 kNm = 11.5×10^3 Nm = $(11.5 \times 10^3) \times 10^3$ Nmm = 11.5×10^6 Nmm, radius of curvature, R = 100 mm.

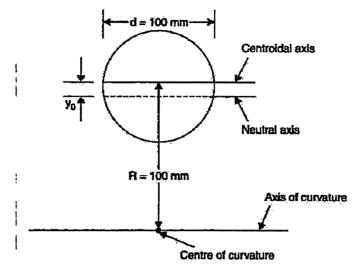


Fig. 23.11

(i) Position of neutral axis. Let y_0 = distance of neutral axis from centroidal axis the position of neutral axis is given by equation (23.3) as

$$y_0 = -\frac{R \times h^2}{R^2 + h^2} \qquad ...(i)$$

The value of h^2 for circular section is given by equation (23.8) as,

$$h^{2} = \frac{d^{2}}{16} + \frac{1}{128} \times \frac{d^{4}}{R^{2}} + \dots = \frac{100^{2}}{16} + \frac{1}{128} \times \frac{100^{4}}{100^{2}}$$
(Here $R = 100$ mm and d also = 100 mm)
$$= \frac{10000}{16} + \frac{10000}{128} = \frac{90000}{128} = 703.125$$

Substituting this value of h^2 in equation (i), we get

$$y_0 = -\frac{100 \times 703.125}{100^2 + 703.125} = -6.57 \text{ mm.}$$
 Ans.

-ve sign means the neutral axis will be below the centroidal axis.

(ii) Maximum and minimum stresses. The stress at any layer at a distance of y from centroidal axis is given by equation (23.1) as

$$\sigma = \frac{M}{R \times A} \left[1 + \frac{R^2}{h^2} \left(\frac{y}{R + y} \right) \right]$$

where $A = \text{Area of cross-section} = \frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times 100^2 = 2500 \,\pi \,\text{mm}^2$

 $M = 11.5 \times 10^6 \, \mathrm{Nmm}$

The stress will be maximum at the extreme bottom layer where y = -50 mm

$$\sigma_{\text{max}} = \frac{11.5 \times 10^6}{100 \times 2500 \,\pi} \left[1 + \frac{100^2}{703.125} \left(\frac{-50}{100 - 50} \right) \right]$$
$$= 14.636 \left[1 - 14.222 \right] = -193.52 \,\text{N/mm}^2$$
$$= 193.52 \,\text{N/mm}^2 \,\text{(compressive)}. \quad \text{Ans.}$$

The stress will be minimum at the extreme top layer where y = +50 mm.

$$\sigma_{\min} = \frac{11.5 \times 10^6}{100 \times 2500 \,\pi} \left[1 + \frac{100^2}{703.125} \left(\frac{50}{100 + 50} \right) \right]$$
$$= 14.636 \left[1 + 4.74 \right] = 84.02 \text{ N/mm}^3. \text{ (tensile)} \quad \text{Ans.}$$

Problem-4:

A hook carries a load of 7.5 kN and the load line is at a distance of 20 mm from the inner edge of the section which is trapezoidal. The load line also passes through the centre of curvature of the hook. The dimensions of the central horizontal trapezoidal section are: inner width =30 mm; outer width = 15 mm; depth =30 mm. Calculate the maximum and minimum stresses. Also plot the variation of stress across the section.

Sol. Given:

Load.

$$P = 7.5 \text{ kN} = 7.5 \times 1000 \text{ N}.$$

Distance of load line from inner edge of the section = 20 mm. Centre of curvature lies on load line. This means the radius of curvature and eccentricity will be equal.

$$R = e$$

Dimensions of trapezoidal section.

Inner width, $b_1 = 30$ mm, outer width, $b_2 = 15$ mm and depth, d = 30 mm.

Refer to Fig. 23.13.

Let us first find the location of centroidal axis.

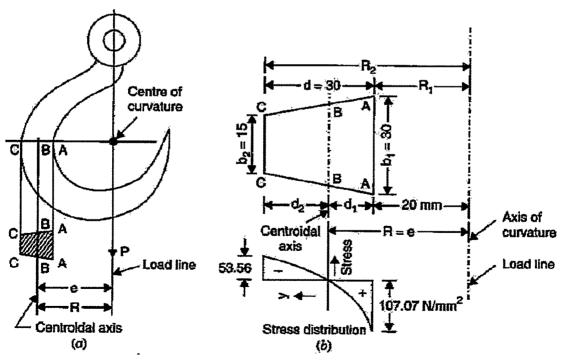
The distance d_1 of the centroidal axis from the side b_1 is given by

$$d_1 = \left(\frac{b_1 + 2b_2}{b_1 + b_2}\right) \times \frac{d}{3} = \left(\frac{30 + 2 \times 15}{30 + 15}\right) \times \frac{30}{3} = \frac{60}{45} \times 10 = 13.33 \text{ mm}$$

$$d_2 = d - d_1 = 30 - 13.33 = 16.67 \text{ mm}$$

From Fig. 23.13, it is clear that

$$R_1 = 20 \text{ mm}, R = 20 + d_1 = 20 + 13.33 = 33.33 \text{ mm},$$



$$R_2 = 20 + d = 20 + 30 = 50 \text{ mm}$$

 $e = R = 33.33 \text{ mm}$

Area of section,

$$A = \left(\frac{b_1 + b_2}{2}\right) \times d = \frac{30 + 15}{2} \times 30 = 675 \text{ mm}^2$$

Bending moment,

$$M = -P \times e$$

(-ve sign is used as the applied load decreases the curvature of the hook)

$$=-(7.5 \times 1000) \times 33.33 = -249975 \text{ Nmm}$$

Let us now find the value of h^2 for trapezoidal section. It is given by equation (S3.7) as

$$h^{2} = \frac{R^{3}}{675} \left[\left\{ b_{2} + \frac{(b_{1} - b_{2}) \times R_{2}}{b} \right\} \log_{e} \left(\frac{R_{2}}{L_{1}} \right) - (31 - b_{2}) - (31 - b_{2}) \right] - 33.33^{2}$$

$$= \frac{33.33^{3}}{675} \left[\left\{ 15 + \frac{30 - 15}{30} \times 50 \right\} \log_{e} \left(\frac{50}{20} \right) - (30 - 15) \right] - 33.33^{2}$$

$$= \frac{54.85}{640} \left[(40 \log_{e} 2.5 - 1510.8 + 10.8) \right] - 33.33^{2}$$

$$= \frac{54.85}{640} \left[(40 \log_{e} 2.5 - 1510.8 + 10.8) \right] - 33.33^{2}$$

$$= \frac{54.85}{640} \left[(40 \log_{e} 2.5 - 1510.8 + 10.8) \right] - 33.33^{2}$$

$$= \frac{54.85}{640} \left[(40 \log_{e} 2.5 - 1510.8 + 10.8) \right] - 33.33^{2}$$

(i) Max. and min. stresses. The bending stress at any point is given by equation (23.1) as

$$\left[\left(\frac{\nabla}{\chi + H} \right) \frac{\text{SE.SE}}{\text{S.SE}} + I \right] \frac{\text{SPSPS}}{\text{SE.SE} \times \text{STO}} = \left[\left(\frac{\nabla}{\chi + H} \right) \frac{S}{S} \frac{A}{A} + I \right] \frac{M}{H \times A} = 0 = dO$$

$$\left[\left(\frac{\nabla}{\chi + H} \right) \times \text{Op. MI} + I \right] II..II - = 0$$

Stress at the section $C-C_1$ where $y=d_2=16.67$ mm,

(evisserqmoo)
5
mm/N 7 3.43 $- = \left[\frac{73.91}{(73.91 + 88.88)} \times 34.41 + 1\right]11.11 -= $_{\infty}$ 0$

Stress at the section A-A, where $y=-d_1=-13.33$,

$$\left[\frac{88.81 \times 34.41}{0.2} - 1\right] 11.11 = \left[\frac{(88.81 -)}{(88.81 - 88.85)} \times 34.41 + 1\right] 11.11 - = {}_{AA}0$$

We know that direct stress (α_0) is given by

(elizate) ²mm/V LL,LI =
$$\frac{000 \times 37}{370} = \frac{q}{\Lambda} = {}_{0}$$
o

The direct stress is uniform at all points in the cross-section

 $\Delta A_{AD} + \Delta C_{D} = AA$ as sents tractional ΔA_{AB}

-(6liznet) *mm/N 70.701 = 36.39 + 11.11 =

=-53.56 N/mm* (compressive). Ans.

(ii) Plotting of stresses. The resultant stress at any distance y from the centroidal axis is

given by,

$$(b =)_{d}b + 0b = t_{0}$$

$$\left[\left(\frac{\chi}{\chi + \tilde{M}}\right) \times \frac{2\pi}{2_{d}} + I\right] \frac{M}{\tilde{M} \times \tilde{M}} + II.II =$$

$$\left[\left(\frac{\chi}{\chi + \tilde{M}}\right) \times \frac{3533^{2}}{8.37} \times \frac{37699 \chi^{2}}{11.11} + II.II =$$

$$\left[\left(\frac{\chi}{\chi + 86.88}\right) \times \frac{3}{8.37} \times \frac{3}{11.11} + II.II = t_{0}$$

$$0 = II.II - II.II = t_{0}$$

$$0 = \chi$$

$$3A$$

$$0 = 11.11 - 11.11 = 50$$
, $0 = 8$ 3A

$$A = \frac{3.888}{4} + 1 = 11.11 - 11.11 = \frac{14.88.5}{3.888 + 5} + 1 = 11.11 = \frac{14.88.5}{3.888 + 5} = \frac{11.11}{3.888 + 5} = \frac{11.11}{3$$

²mm/N 70,78 -=
$$\left[\frac{01 \times 34.41}{01 + 68.88} + i\right]$$
 II..II = 10, mm 0,I = χ 44

$$^{2}\text{mm/M } 38.94 - \left[\frac{31 \times 36.41}{31 + 68.88} + 1\right] 11.11 - 11.11 = 49.86 \text{ M/mm}^{2}$$

At
$$r_{\rm p}=16.67$$
 mm/N $66.63-6$ $r_{\rm p}=-6.63$ M/mm/s (Already calculated)

 $y = -13.33 \text{ mm}, g_{\star} = 107.07 \text{ M/mm}^2$

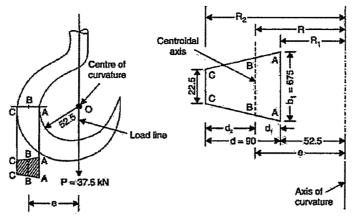
$$^{2}\text{mm/V} 38.85 = \left[\frac{(3-)\times 34.41}{3-88.88} + 1\right] \text{II.II} - \text{II.II} = \frac{1}{3} 3.885 \text{ N/mm}^{2}$$

smm/N
$$88.88 = \left[\frac{(0.1-) \times 38.41}{0.1-68.88} + I\right] LI.II - II.II = 50, mm 0.1 - = 50.44$$

The variation of stress across the cross-section is shown in Fig. 23.13 (b).

Problem-5:

A central horizontal section of a hook is a trapezoidal section shown in Fig. Calculate the maximum compressive and tensile stresses in the section of the hook when it carries a load of 37.5 kN, the load line passes through the centre of curvature.



Sol. Given:

$$P = 37.5 \text{ kN} = 37.5 \times 1000 \text{ N}$$

Load line passes through centre of curvature

 \therefore Eccentricity, e = R (radius of curvature)

where

e = distance between load line and centroidal axis

R = distance between centre of curvature and centroidal axis. and Dimensions of trapezoidal section are :

$$b_1 = 67.5 \text{ mm}, b_2 = 22.5 \text{ mm}, d = 90 \text{ mm}$$

Refer to Fig. 23.14.

Let us first find the location of centroidal axis.

The distance d_1 of the centroidal axis from the side b_1 is given by

$$d_1 = \left(\frac{b_1 + 2b_2}{b_1 + b_2}\right) \times \frac{d}{3} = \left(\frac{67.5 + 2 \times 22.5}{67.5 + 22.5}\right) \times \frac{90}{3} = \frac{112.5}{90} \times \frac{90}{3} = 37.5 \text{ mm}$$

$$d_2 = d - d_1 = 90 - 37.5 = 52.5 \text{ mm}$$

From Fig. 23.14, it is clear that

$$R_1 = 52.5$$
 mm. Then $R = R_1 + d_1 = 52.5 + 37.5 = 90$ mm. $R_2 = 52.5 + d = 52.5 + 90 = 142.5$ mm. $e = R = 90$ mm.

Area of the section,

$$A = \left(\frac{b_1 + b_2}{2}\right) \times d = \frac{67.5 + 22.5}{2} \times 90 = 4050 \text{ mm}^2$$

Bending moment, $M = -P \times e$

(-ve sign is used as the applied load decreases the curvature of the hook)

$$=-(37.5 \times 1000) \times 90 \text{ Nmm} = -3375000 \text{ Nmm}$$

Let us now find the value of h^2 for trapezoidal axis. It is given by equation (23.7) as

$$\begin{split} h^2 &= \frac{R^3}{A} \left[\left\{ b_2 + \frac{b_1 - b_2}{d} \times R_2 \right\} \log_e \left(\frac{R_2}{R_1} \right) - \left(b_1 - b_2 \right) \right] - R^2 \\ &= \frac{90^3}{4050} \left[\left\{ 225 + \frac{67.5 - 22.5}{90} \times 142.5 \right\} \log_e \left(\frac{142.5}{52.5} \right) - \left(67.5 - 22.5 \right) \right] - 90^2 \\ &= 180 \left[(93.75 \times \log_e 2.714^3) - 45 \right] - 8100 \\ &= 8750.26 - 8100 = 650.26 \text{ mm}^2 \end{split}$$

Direct stress, $\sigma_0 = \frac{P}{A} = \frac{37.5 \times 1000}{4050} = 9.26 \text{ N/mm}^2$ Bending stress at any layer at a distance y from centroidal axis is given by,

$$\sigma = \frac{M}{A \times R} \left[1 + \frac{R^2}{h^2} \left(\frac{y}{R + y} \right) \right]$$

$$= -\frac{3375000}{4050 \times 90} \left[1 + \frac{90^2}{650.26} \left(\frac{y}{90 + y} \right) \right] \qquad [x \quad M = -3375000 \text{ N mm}]$$

$$= -9.26 \left[1 + \frac{12.456y}{90 + y} \right] \qquad ...(i)$$

Bending stress at section C–C where $y = d_2 = 52.5$ mm becomes as

$$\sigma_{cc} = -9.26 \left[1 + \frac{12.456 \times 52.5}{90 + 52.5} \right] = -51.75 \text{ N/mm}^2$$

Resultant stress at section C-C becomes

$$(\sigma)_{cc} = \sigma_0 + \sigma_{cc}$$

= 9.26 - 51.75 = -42.49 N/mm³ (Compressive). Ans.

Bending stress at section A - A where $y = -d_1 = -37.5$ mm becomes as

$$\sigma_{AA} = -9.26 \left[1 + \frac{12.456 \times (-37.5)}{90 - 37.5} \right]$$
 [Substitute $y = -37.5$ in equation (i)]
= 73.12 N/mm²

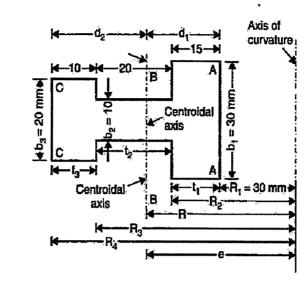
 \therefore Resultant stress at section A-A becomes as

$$(\sigma_r)_{AA} = \sigma_0 + \sigma_{AA} = 9.26 + 73.12 = 82.38 \text{ N/mm}^2$$
 (Tensile). Ans.

Problem-6:

A hook carries a load P and the load line is at a distance of 30 mm from the inner edge of the central horizontal section which is of I-section with the dimensions shown in Fig. 23.15. The centre of curvature of the hook is in the load line. Determine the magnitude of the load P if the maximum stress in the hook is not to exceed the permissible stress of 120 Nlmm2. What will be the maximum compressive stress in the hook for that value of the load?

Sol. Given:



Max. permissible stress, $\sigma_r = 120 \text{ N/mm}^2$

As the load line passes through centre of curvature, the distance between load line and centroidal axis is equal to the distance between axis of curvature and centroidal axis.

or

$$e = R$$

Dimensions of I-section are

$$b_1 = 30 \text{ mm}, b_2 = 10 \text{ mm}, b_3 = 20 \text{ mm}$$

 $t_1 = 15 \text{ mm}, t_2 = 20 \text{ mm}, t_3 = 10 \text{ mm}$

Also from Fig. 23.15

$$R_1 = 30 \text{ mm}$$

 $R_2 = 30 + 15 = 45 \text{ mm}, \quad R_3 = R_2 + t_2 = 45 + 20 = 65 \text{ mm}$
 $R_4 = R_3 + t_3 = 65 + 10 = 75 \text{ mm}$

and

Area of cross-section,
$$A = b_1 t_1 + b_2 t_2 + b_3 t_3$$

= $30 \times 15 + 10 \times 20 + 20 \times 10$
= $450 + 200 + 200 = 850 \text{ mm}^2$

Let us first find the location of centroidal axis. This can be done by taking moments of each areas about AA. Then

$$A \times d_1 = (30 \times 15) \times \frac{15}{2} + (10 \times 20) \times (15 + 10) + (20 \times 10) \times (15 + 20 + 5)$$
 or
$$850 \times d_1 = 3375 \times 13000 + 8000$$

$$d_1 = \frac{3375 \times 13000}{850} = 19.26 \text{ mm}$$

$$d = 15 + 20 + 10 = 45$$
 and
$$d_2 = d - d_1 = 45 - 19.26 = 25.74 \text{ mm}$$
 Now
$$R = R_1 + d_1 = 30 + 19.26 = 49.26 \text{ mm}$$
 Also
$$e = R = 49.26 \text{ mm}.$$

Let us now find the value of h^2 for I-section. It is given by equation (23.10) as the value

$$\begin{split} h^2 &= \frac{R^3}{A} \left[b_1 \log_e \left(\frac{R_2}{R_1} \right) + b_2 \log_e \left(\frac{R_3}{R_2} \right) + b_3 \log_e \left(\frac{R_4}{R_3} \right) \right] - R^2 \\ &= \frac{49.26^3}{850} \left[30 \log_e \left(\frac{45}{30} \right) + 10 \log_e \frac{65}{45} + 20 \log_e \left(\frac{75}{65} \right) \right] - 49.26^2 \\ &= 140.625 \left[30 \times 0.4054 + 10 \times 0.3677 + 20 \times 0.1431 \right] - 2426.5 \\ &= 140.625 \left[12.16 + 3.677 + 2.862 \right] - 2426.5 = 203.04 \end{split}$$

The bending stress at a distance y from centroidal axis is given by

$$\sigma_b = \frac{M}{A \times R} \left[1 + \frac{R^2}{h^2} \left(\frac{y}{R + y} \right) \right] \qquad ...(i)$$

Here M = Bending moment due to load P

 $=-P \times e$ (-ve sign is used as applied load decreases the curvature of the hook).

 $=-P \times 49.26 \text{ Nmm} \text{ (where } e = R = 49.26 \text{ mm)}$

Substituting the value of M and other known values in equation (i), we get

$$\begin{split} \sigma_b &= \frac{-P \times 49.26}{850 \times 49.26} \left[1 + \frac{49.26^2}{203.04} \left(\frac{y}{49.26 + y} \right) \right] \\ &= -1.176 \times 10^{-3} \, P \left[1 + 11.95 \left(\frac{y}{49.26 + y} \right) \right] \qquad ...(ii) \end{split}$$

The bending stress will be maximum at section AA where y = -19.26 mm. This stress will be tensile. Substituting y = -19.26 in equation (ii), we get

$$(\sigma_b)_{\text{max}} = -1.176 \times 10^{-3} P \left[1 + 11.95 \times \frac{(-19.26)}{(49.26 - 19.26)} \right]$$
$$= +7.846 \times 10^{-3} P \text{ N/mm}^2 \text{ (tensile)}$$

The direct stress, $\sigma_0 = \frac{P}{A} = \frac{P}{850} \text{ N/mm}^2 \text{ (Tensile)}$

.. Maximum resultant stress becomes as

OF

$$(\sigma_r)_{\text{max}} = \sigma_0 + (\sigma_b)_{\text{max}} = \frac{P}{A} + 7.846 \times 10^{-3} P \text{ N/mm}^2$$

But this stress should not exceed the maximum permissible stress of 120 N/mm²

$$\therefore \frac{P}{850} + 7.846 \times 10^{-3} P = 120 \text{ or } P \left[\frac{1}{850} + 7.846 \times 10^{-3} \right] = 120$$

$$P \left[\frac{1 + 850 \times 7.846 \times 10^{-3}}{850} \right] = 120 \text{ or } P \left[\frac{1 + 6.6691}{850} \right] = 120$$

$$\therefore P = \frac{120 \times 850}{7.6691} = 13,300 \text{ N} = 13.3 \text{ kN. Ans.}$$

Maximum Compressive stress due to load, P = 13.3 kN

The bending stress (compressive) will be maximum at the section C-C where $y = d_2 = 25.74$ mm. Substituting the value of y = 25.74 in equation (ii), we get maximum compressive bending stress.

:. Compressive
$$(\sigma_b)_{\text{max}} = -1.176 \times 10^{-3} \times 13300 \left[1 + 11.95 \times \left(\frac{25.74}{49.26 + 25.74} \right) \right]$$
(Here $P = 13,300 \text{ N}$)

$$= -79.78 \text{ N/mm}^2$$

Direct stress,
$$\sigma_0 = \frac{P}{A} = \frac{13300}{850} = 15.64 \text{ N/mm}^2 \text{ (tensile)}$$

... Maximum resultant compressive stress at CC

=
$$\sigma_0 + (\sigma_b)_{\text{max}}$$
 (compressive)
= 15.64 - 79.78 = -64.13 N/mm² (compressive). Ans.

Problem-7:

Problem 23.7. The curved member shown in Fig. 23.16 has a solid circular cross-section 0.10 m in diameter. If the maximum tensile and compressive stresses in the member are not to exceed 150 MPa and 200 MPa respectively, determine the value of load P that can safely be carried by the member.

(AMIE, Summer 1996)

Sol. Given:

Dia of section, d = 0.10 m

Max. tensile stress,

$$(\sigma_r)_t = 150 \text{ MPa} = 150 \times 10^6 \text{ N/m}^2$$

Max. compressive stress,

$$(\sigma_c)_c = 200 \text{ MPa} = 200 \times 10^6 \text{ N/m}^2$$

Refer to Fig. 23.16 (a)

Area of cross-section.

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} (0.1^2) = 7.854 \times 10^{-3} \text{ m}^2$$



«—0.15 m—**»** ^Р

 $0.05 \, m$

e = Eccentricity = distance between load line and centroidal axis

= 0.15 + 0.05 + 0.05 = 0.25 m

R =Radius of curvature

= distance between centre of curvature and centroidal axis

= 0.05 + 0.05 = 0.1 m

Let us first find the value of h^2 for circular section. It is given by equation (23.8) as

$$h^{2} = \frac{d^{2}}{16} + \frac{1}{128} \times \frac{d^{4}}{R^{2}} + \dots$$

$$= \frac{0.1^{2}}{16} + \frac{1}{128} \times \frac{0.1^{4}}{0.1^{2}} \qquad (x R = 0.1 \text{ m also } d = 0.1 \text{ m})$$

$$= 7.031 \times 10^{-4} \text{ m}^{2}$$

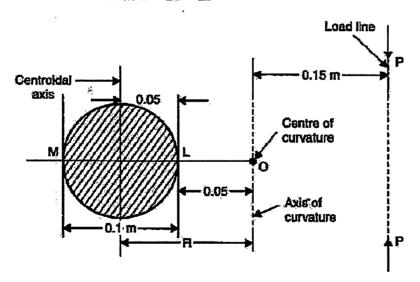


Fig. 23.16 (a)

Bending moment, $M = P \times e$ (+ve sign is used as load P increases the curvature) = $P \times 0.25 = 0.25 P$ Nm

The bending stress at any section at a distance y from centroidal axis is given by equation (23.1) as

$$\sigma_{b} = \frac{M}{A \times R} \left[1 + \frac{R^{2}}{h^{2}} \times \left(\frac{y}{R + y} \right) \right]$$
 [Here $\sigma_{b} = \sigma$]
$$= \frac{0.25 P}{(7.854 \times 10^{-3}) \times 0.1} \left[1 + \frac{0.1^{2}}{7.031 \times 10^{-4}} \times \left(\frac{y}{0.1 + y} \right) \right]$$
 ...(i)

As the load P is compressive, hence direct stress (σ_0) will be compressive at all points of the section.

Direct stress,
$$\sigma_0 = \frac{P}{A} = \frac{P}{7.854 \times 10^{-3}}$$
 N/m² (compressive)
$$= -\frac{P}{7.854 \times 10^{-3}}$$
 (-ve sign due to compressive stress)
$$= -127.32 P$$

As the load is compressive, the bending stress will be compressive at the inner point L of the cross-section where y=-0.05 m and it will be tensile at the outer point M of the cross-section where y=0.05 m.

1st Case. Max. compressive bending stress will be obtained by substituting y = -0.05 m in equation (i).

$$\therefore \text{ Compressive } (\sigma_b)_{\text{max}} = \frac{0.25 \, P}{(7.854 \times 10^{-3} \times 0.1)} \left[1 + \frac{0.1^2}{7.031 \times 10^{-4}} \times \left(\frac{-0.05}{0.1 - 0.05} \right) \right] = 318.3 \, P \left[1 - 14.22 \right] = -4208.8 \, P$$

.. Max. compressive stress, $(\sigma_r)_c = \sigma_0 + (\sigma_b)_{max}$ (compressive) = -127.32 P - 4208.8 P = -4336.12 P.

But maximum compressive stress is given as $200 \times 10^6 \text{ N/m}^2$

... Maximum compressive stress, $(\sigma_r)_c = -200 \times 10^6 \text{ N/m}^2$

(-ve sign is due to compressive stress)

$$\therefore -200 \times 10^6 = -4336.12 P$$

$$P = \frac{200 \times 10^6}{4336.12} = 46124 \text{ N} = 46.124 \text{ kN} \qquad ...(ii)$$

2nd Case. Max. tensile bending stress will be obtained by substituting y = +0.05 m in equation (i).

$$\therefore \quad (\sigma_b)_{\text{max}} \text{ (tensile)} = 318.3 \ P \left[1 + \frac{0.1^2}{7.031 \times 10^{-4}} \times \left(\frac{0.05}{0.1 + .05} \right) \right] = 1827.32 \ P$$

 $\therefore \text{ Max. tensile stress, } (\sigma_p)_i = \sigma_0 + (\sigma_b)_{\text{max}} \text{ (tensile)}$ = -127.32 P + 1827.32 P = 1700 P

But max tensile stress, $(\sigma_r)_r = 150 \times 10^6 \text{ N/m}^2$ (given)

$$1700 P = 150 \times 10^6$$

$$P = \frac{150 \times 10^6}{1700} = 88235.3 \text{ N} = 88.235 \text{ kN} \qquad ...(iii)$$

Comparing equations (ii) and (iii), the safe load will be lesser of the two.

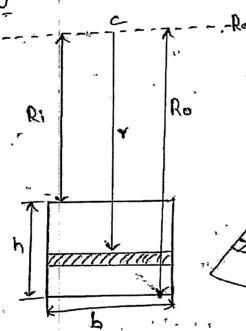
$$P = 46.124 \text{ kN}$$
. Ans.

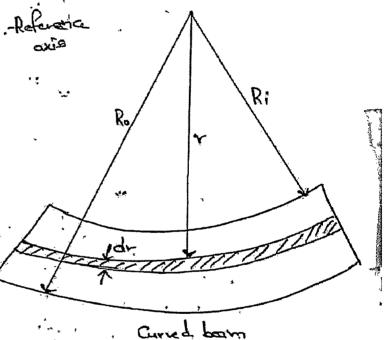
Expression for radius of neutral axis for different sections: -

Neutral Asis: -

Neutral aria is formed by the intersection of the neutral layer with the neutral section of the beam. The layers above the neutral axis will be shortened, due to compression and the layers below the neutral axis will be changeted due to tension. The neutral axis will remain neutral i.e., it will neither be elongated nor shortened. The neutral axis of a section is so located that, the moment of the entire area about the axis is zero.

Rectangular Cross Section: -





Let b- width of rectangular cross section.

h- Depth of rectangular cross section.

A - Area of cross section = "bxh

R: - Radius of curvature of inner libre

Ro-Radius of curvature of outer libre.

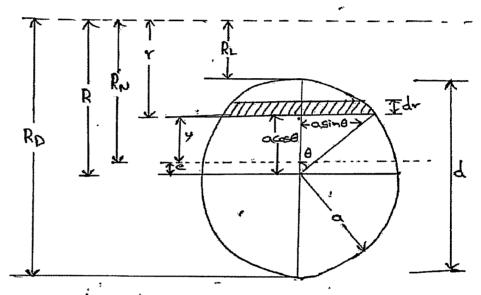
The neutral axis of this Good Section is determined by the relation.

$$R_{N} = \frac{A}{\sqrt{\frac{R_{0} dA}{r}}} = \frac{bh}{\sqrt{\frac{R_{0} da}{r}}} = \frac{h}{\sqrt{\frac{R_{0} da}{r}}} = \frac{h}{\sqrt{\frac{R_{0} da}{r}}} = \frac{h}{\sqrt{\frac{R_{0} da}{r}}}$$

$$R_{N} = \frac{h}{[\ln r]_{R_{i}}^{R_{o}}}$$

$$R_{N} = \frac{h}{\log(\frac{R_{o}}{R_{i}})}$$

Grader coss sections -



Let d= Diameter of circular section

R:= Radius of currature of inner libre

R = Radius of currature of controidal axis.

Ro = Radius of curatine of neutral axis.

Ro = Radius of curvature of outer libre.

The neutral oxis of the cross-section is calculated from the relation

$$R_{N} = \frac{A}{\int \frac{dA}{r}} \rightarrow \Omega$$

From figure $Y = RN-Y = (R_0 - e) - (a \cdot cos \theta - e) = R - a cos \theta \rightarrow 2$ Integrating the equation (3),

i, dr = asinodo

Aca of strp, dA = (2a.sino).dr = (2a.sino) (asino.do)

Thus
$$\int_{A}^{A} \frac{dA}{r} = \int_{0}^{\pi} \frac{2a^{3} \cdot 2in^{3}\theta}{(R - a\cos\theta)} \cdot d\theta$$

$$= \int_{a}^{\pi} \frac{2a^{3} \cdot 2in^{3}\theta}{a(R - \cos\theta)} \cdot d\theta$$

Let $\frac{R}{a} = b$ and $\frac{2in^{3}\theta}{(b - \cos\theta)} \cdot d\theta$

Adding and subtanting $(b\cos\theta + b^{3}) = (b\cos\theta + b^{3}) = (a\cos\theta + b^{3}) = (a$

Are of circular cross section
$$A = \frac{R}{4}$$
?

Alos from eq (1)

 $R_N = \frac{1}{2} \frac{1}{R} - \sqrt{R^2 - a^2} \frac{1}{2}$

.: Neutral axis of the circular section,

 $R_N = \frac{1}{2} \frac{1}{R} - \sqrt{R^2 - a^2} \frac{1}{2}$

.: Neutral axis of the circular section,

 $R_N = \frac{1}{R} - \sqrt{R^2 - a^2} \frac{1}{2}$

.: Neutral axis of the circular section of $R_N = \frac{1}{R} - \frac{1}{R} \frac{1}{2}$

.: Neutral axis a curred bears of $R_N = \frac{1}{R} \frac{1$

The neutral axis of the cross-section is colorlated from the relation $\int \frac{dA}{r} = \int \frac{b_1 dr}{r} + \int \frac{b_2 dr}{r} = b_1 \int \frac{dr}{r} + b_2 \int \frac{dr}{r}$ = b. loger | R, + be loger | R, JdA = b loge Ri + be loge Ro Area of Transs section A = b,h,+be Neutral outs of T- Section RN = b, log R1 + be log R0

I- Chose section: -Consider a cured beam of I-Section as shown in the figure, bo - width of I-section at outer fibre. bi - width of I-section at inner Pibre., E; to-Thickness of inner and outer flanges.

h-Depth of the I-section:

t-Thickness of web Ri-Radius of curvature of inner fibre Ro-Radius of currature of outer libre. RN-Radius of curvature of neutral axis. R-Radius of curvature of controidal axis. The neutral axis of the cross section is calculated from the relation. Ana of strip dA = bxdy = bxdr

The neutral axis of the cross- Section is Calculated from the relation

Ara & stip, dA = bxdy

Total ones of I-section,
$$A = ti(b; -b) + to(bo-b) + tih$$

$$A = \begin{cases} \frac{dA}{r} = \int_{0}^{\infty} \frac{b_{i}dr}{r} + \int_{0}^{\infty} \frac{b_{i}d$$

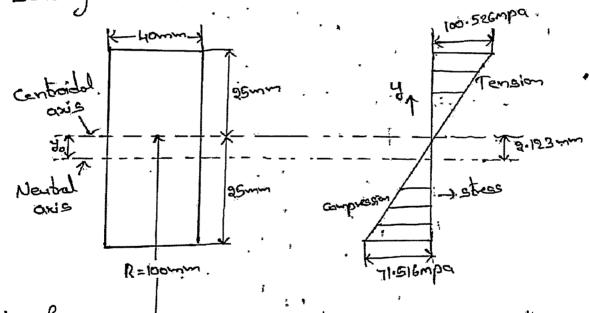
A curved borr of rectangular section, initially unstressed, is subjected to bending moment of 1400N-m which tends to straighten the borr. The section is 4cm wide by 5cm deep in the plane of bending, and the mean radius of curvature is 10cm. Plane of bending, and the mean radius of unrature is 10cm. Find the position of the recital axis and magnitudes of the greatest find the position of the recital axis and magnitudes of the greatest bending alues and draw a diagram to show approximately how the stees varies across the section.

Width of the beam, b=4cm= 40mm.

Depth of the beam, d=5cm=50mm.

Mean radius of the curvature, R= 10cm=100mm.

Bending moment, M=1400N-m= 1400000 N-mm



Axis of _____ & ___ Rectangular Beam

i, stesses at the inner and outer surface: -

The stress at any point in rectangular crossisection at a distance y.

1 20 = - 2.183mm 1 9 = - Rxh2 = - 9.123mm. labiotine of the neutral axis how he central de central de la labiotista - is to sition of the Newbal Axis: -(2018224ma) pgM 212.15 = 50 K 2mm/4 312.17 = 0 $\left[\frac{(96+09)}{56}\right] \frac{L16}{3001} + 1$ $\left[\frac{(94\times95)\times99}{9000091}\right] = 0$ $\frac{1}{\sqrt{2}} = \frac{1}{\sqrt{2}} = \frac{1$ The value of y' is positive at the outer surface. Thus, stess of Ontside Face:-= 100-526 A/mm2 . > [OF = 100 526 Mpa (Tensile) $\left[\left(\frac{56-001}{56-} \right) \frac{L16}{500} + 1 \right] \frac{(04 \times 05) \times 001}{0000041 -} = \frac{1}{50}$ $\left[\frac{P-1}{(P-1)+9}\right]^{\frac{1}{2}} + 1 = \frac{1}{10}$ (Emm bxd=A DxA). the bending moment will be negative as it. tends to straighten thebase The value of "y" is negotive at the inner surface and also. Strees at Inner Surlaces -[Emmlig = 34] K. Emm 116 = 912.316 = 91 2001 - (05 - 001 xg) wy | 05 = $V_{S} = \frac{q}{b_{2}} \left| \nabla v \left[\frac{3b - q}{3b + q} \right] \right| - b_{S}$ The value of 12 for rectiongular cross section is given ag

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sketch a diagram to show the variation of stress games the section. Also

straighten it, defermine the estering of the inner and outer surfaces and

A curred beam of rectangular section. It some width and homen depth

bending movent of 40000 m is applied to the beam which tends to

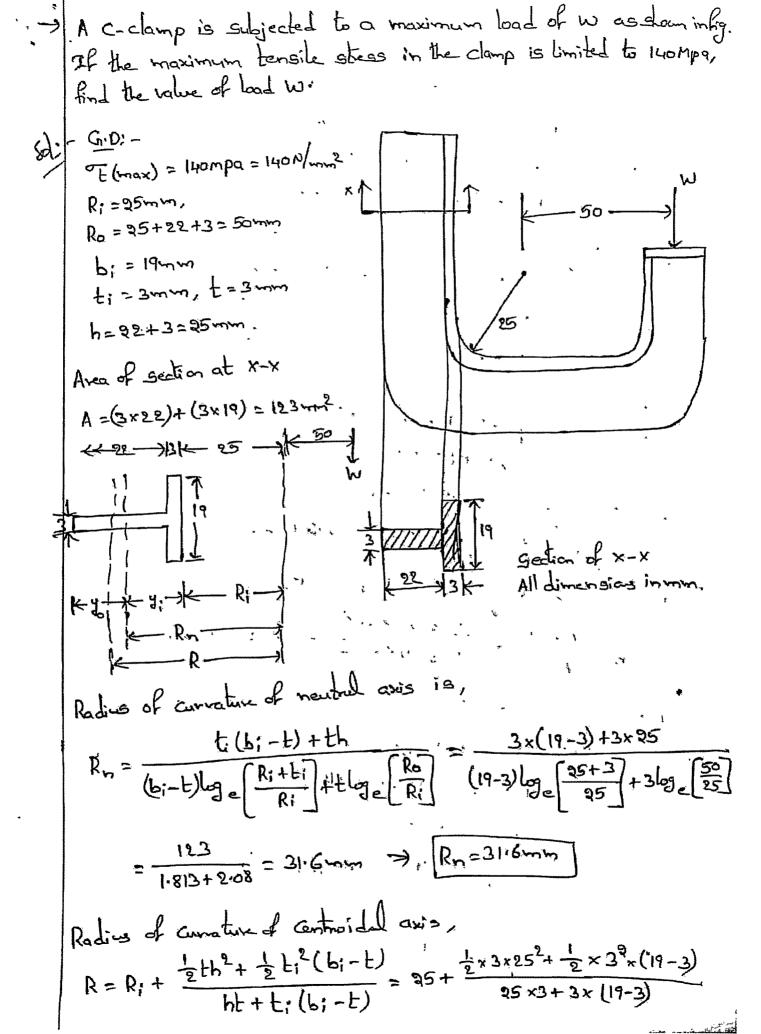
and mean radius of construe of borns is initially insteased. If a

= 603 [2x60 +40] - 60? = 5400(0-693)-3600 > L2=142.995=143mm2 143mms stees at Imer surface -The value of "y" is negative at the inner surface and also the bending moment will be negative as it tends to straighten the bar. (Area A=bxdmnz) $rac{1}{R} = \frac{-M}{RA} \left[\frac{R^2}{1 + \frac{R^2}{R^2}} \left(\frac{-y}{R^2 + \frac{(-y)}{R^2}} \right) \right]$ $\frac{-400000}{60\times(40\times30)} \left[1 + \frac{60^2}{143} \left[\frac{-20}{60-20} \right] \right]$ -5.556 (1-12.587) 07 = 64.377 N/mm2 : 07 = 64.377 Mpa (Tensile) stess at Outside Face: The value of "y" is positive at the outer surface $\sigma_0 = \frac{-1}{RA} \left[1 + \frac{R^2}{h^2} \left[\frac{y}{R+y} \right] \right]$ = -400000 [+ 602 20 60+20] 5.5556 (1+6,338) 00 =-40.770 N/mm3 1:00 = 40.770mpa (Compressive) ii) Position of the Neutral Axis: -Let "y" be the distance of neutral axis from centroidal axis = 7 = -8×1/2 > = -60×143 = -3.277 = -3.277 =

The inside diameter of a circular ring is somm. The ring has a circular ring is somm. The ring has a circular cross-section of 60mm diameter. A compressive load of 15kN is applied on the ring is shown in figure. Find the stresses at Pand Q. Also find the position of the neutral axis. 80/2 G.D: -Inside diameter of (Section at Pay) aralar ring dis 80mm conf Ring diameter d= 60mm Compressive load W= 15kN. Mean radius of curvature of circular ring, Gomm $R = \frac{di}{9} + \frac{d}{9} = \frac{80}{9} + \frac{60}{9}$ R = 40+30 =70 mm. > (1. R=70mm) Area of coss-section, A= Tx d2= Tx 602 = 2827.433mm3. Regultant stres = Direct stres + Bending stres. The direct stress is given as; d= -W (" Negative sign) The bending stress is given as $\sigma_{P}^{2} = \frac{W}{A \cdot B} \left[\frac{1}{1 + \frac{1}{12}} \left[\frac{A + A}{12} \right] \right] = \frac{M \cdot B}{A \cdot B} \left[\frac{A}{1 + \frac{1}{12}} \left[\frac{A}{12} \right] \right]$ Stress at Outside Face (ie at P): -Thus, the resultant stress for outside surface is given as, $\sigma_0 = -\frac{W}{A} + \frac{WR}{AR} \left[1 + \frac{R^2}{R^2} \left[\frac{y}{R+y} \right] \right],$ $\sigma_0 = \frac{W}{A} + \frac{R^2}{L^2} \left[\frac{y}{R+Y} \right].$ For outside surface, the value of y' = & > y = 30mm. The value of "h" for circular section is given by,

mon828.88 = 84E.E-OT. To evilua of to el eise los Lostinan en (E99: GHEXOL- - of Eben Labiodors mont sino lecturar & sombald of "L" Jal Pastion of Neutral Axis: (20 = -79:363. Whom? > 30 = 79:363 Mpa (Compressiv) (E99.548) (E71.7.686) = F-M-15. 17 $\sqrt{\frac{E-1}{MA}} + \frac{MR}{MA} \left[1 + \frac{R^2}{R^2} \left[\frac{-y}{A} \right] \right]$, eo my 21 surve bien in sente frustresi en -: (de foce (i.e. ot de): -(diens) 09MY15.15 = 50 [0E+0] (E99.548) x 554.1288 = 00051 Opo mi souler avado Primitizadus Form Edd. 24 & Edd. 02+200 - 120 51 + 31 - 31 - 34 --- + 30 + 11 + 31 = 34

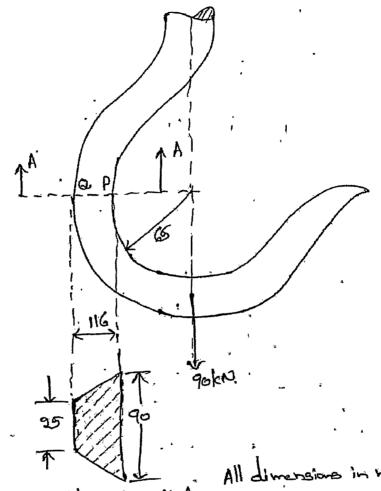
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R = 23.20mm Distance between neutral axis and antidal axis, e= R-Rh = 33.20-31.60 = 1.6mm. Distance between load wand Centroidel axis, x=50+R=50+33.20=83.20mm. Bending moment about centroidel axis, M= W. x = "W x 83. 20 = 83.20 W N- mm. The section at x-x is subjected to direct tensile load w and a bending monet of 83.20W, the maximum tensile thess will oxcurat Distance of point P from neutral axis, 4:= Rn- R:= 31.6-25=6.6mm. Direct tensile stres pt x-X, . . . E = W = 123 = 0.00813 W/ N/m2. Maximum bending about P, 0 bi = M.yi = 83.20W ×6.6 = 0.112W N/mm2 Maximum tensile stress, (tomax) = of tobi 140 =0.00813 W+ O.42 W 140 = 0.12013W : W= 140 = 1165.4N. W= 1165.4N

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A crone hook has a traperoidal section at A-A as shown in fig. find the maximum stress at points pand a.



(%)

At Section A'-A

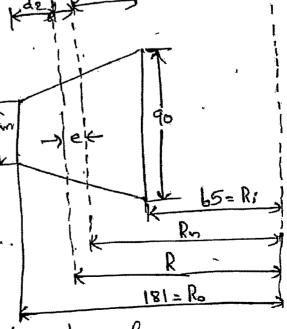
C.D: -Inner radius, Ri= 65mm Outer radius, Ro=65+116=181mm. Width of inner end, bi = 90mm. Width of outer end, bo = 95 mm:

Depth h=116mm Direct tensile load, W=90KN.

Distance from the rental axis to the inner surface,

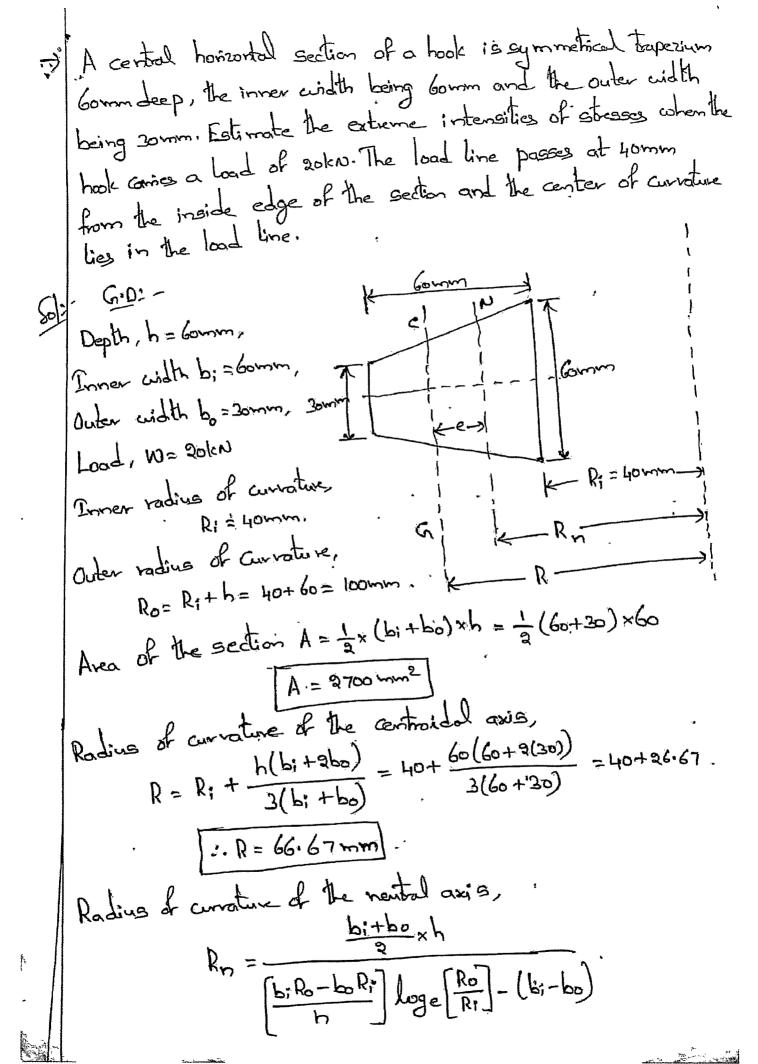
: di= Rn-65.

the neutral oxis to the outer surface, . d2= 181- Rn.



An open ring has 7-section. It is subjected to a tensile load of 80 kN as shown in figure. Find the stresses at point p and a. * Harm (section at Pa) 80/c2 (n.D: - | W= 80kn = 80 × 103 m , Height b = 80mm. Length be = 120mm, Thickness, tiste = 20mm. Area of T-Cross Section, A = bit, + bete = 80x 20+120x20 Coordinate of Central, d= A,x,+Aoxe - (80 x30) x (0 + (120 x 20) x (20+ 120) = 52 mm. Thus R= 150+52=202mm, R=150mm, R=170mm and R3 = 150+20+ 120 = 290mm. The value of he for T-Section is given by $h^2 = \frac{R^3}{A} \left| b_1 l \ln \frac{R_2}{R} + t l \ln \frac{R_3}{R_0} \right| - R^2$ $= \frac{(202)^3}{1000} \left| 80 \text{km} \left(\frac{170}{150} \right) + 20 \text{km} \left(\frac{290}{170} \right) - (202)^2 \right|$ ("t=t,=te) = 206.62×20.695-(202)2 = 42644.158 - 40804 = 1840.158 mm2

P= 1840 . 128 mm 5 The bending moment is such that it tends to decrease the curature of the ring. Hence, it is taken as negative. stees at point "P". $\frac{1}{1+\frac{R^2}{A}}\left[\frac{y}{R+y}\right] = \frac{-W}{A} \cdot \frac{R^2}{h^2} \left[\frac{y}{R+y}\right]$ Where = 2 = R3 - R = 290 - 202 = 88mm. -80×103 × (302)2 × (302+88) op = - 20 x22.174 x 0.303 p = 134.573 Mpa (Compressive) Stress at point a" :- " $\sigma_{\Delta} = -\frac{W}{A} - \frac{WR}{AR} \left[1 - \frac{R^2}{h^2} \left[\frac{Y}{R-Y} \right] \right]$ $R^{2} = \frac{W}{\Lambda} \cdot \frac{R^{2}}{\Gamma^{2}} \left[\frac{Y}{R-Y} \right]$ $=\frac{80\times10^{3}}{1000}\cdot\frac{(202)^{2}}{1840\cdot158}\left[\frac{52}{202-52}\right]$ 00 = 20 x 22 . 17 4 x 0 . 347 : 0 = 153-888 Mpa (Tensile)



POWER TRANSMISSION SYSTEMS AND PULLEYS

When the power is to be transmitted between two co-axial shafts, connecting elements like couplings or clutches can be employed on the other hand, if the power is to be transmitted between two non co-axial shafts but may be kept parallel or non parallel and at some distances, we need some intermediate driving elements like belts, chains, or gears. When the co-axial shafts are connected by couplings, their speeds will not differ whereas we can reduce or increase the speed of driven shaft through these intermediate driving links. Shortly saying that the drives are the intermediate mechanism between the driving and driven shafts in order to transmit the power or energy produced in one machine to another or between two members of a machine along with the variation of shaft speeds.

The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. The amount of power transmitted depends upon the following factors:

- 1. The velocity of the belt.
- 2. The tension under which the belt is placed on the pulleys.
- 3. The arc of contact between the belt and the smaller pulley.
- 4. The conditions under which the belt is used.

It may be noted that

- a. The shafts should be properly in line to insure uniform tension across the belt section.
- b. The pulleys should not be too close together, in order that the arc of contact on the smaller pulley may be as large as possible.
- c. The pulleys should not be so far apart as to cause the belt to weigh heavily on the shafts, thus in-creasing the friction load on the bearings.
- d. A long belt tends to swing from side to side, causing the belt to run out of the pulleys, which in turn develops crooked spots in the belt.
- e. The tight side of the belt should be at the bottom, so that whatever sag is present on the loose side will increase the arc of contact at the pulleys.
- f. In order to obtain good results with flat belts, the maximum distance between the shafts should not exceed 10 metres and the minimum should not be less than 3.5 times the diameter of the larger pulley

CLASSIFICATION OF POWER TRANSMITTING DRIVES

Modern machines utilize mechanical, hydraulic, pneumatic and electrical drives. The design principles of some commonly adopted mechanical drives are discussed, i.e., the power transmitting elements may be mechanical items.

Mechanical drives may be classified based on the following conditions.

- a) According to the physical conditions of transmission they may be classified into.
 - i) Friction drives such as belt and rope drives, and
 - ii) Toothed drives such as gears and chain drives
- b) According to the method of linking the driving and driven members, they may be grouped into,
 - i) Drives with direct contact between the driving and driven members such as gears,
 - Drives with an intermediate link between the driving and driven members auch as belts, ropes and chain drives.
- c) According to positions of shaft axes as ,
 - flexible drives: Here the slight variation of shaft axes from parallelism may be permitted because this variation will not affect much the proper function of drive and also the slight variation of centre distance may not be minded much.
 - Ex: Belt drives, rope drives, chain Drives.
 - ii) Rigid drives: Here the variations of shaft axes from parallelism and centre distance will not be permitted because of the rigid construction and direct contact of the driving and driven members.
 - iii) Ex: Gear drives.

ELEMENTS OF A POWER DRIVE

Each transmission mechanism comprises two essential shafts namely the driving(input) shaft and the driven (output) shaft.

The members of power suppliers, lke shafts and pulleys of a motor are called as driving members and the members of power receivers like shafts and pulleys of a machine(say lathe or Rice mill) may be called as driven members.

Each drive, whether it may be a belt, chain, or gear drive, has its specific features and fields of application. The choice of drive depends on the amount of power to be transmitted, peripheral distance between the axes of the mating members.

BELT DRIVE:

It is a mechanical drive in which the driving shaft and driven shaft are connected by a flexible link(i.e belt) through pulleys mounted on the shafts.

Generally, the belts and chin drives are called as flexible drives because they allow the designer considerable flexibility in location of driving and driven machineries and tolerances are not critical as in the case of gear drives. Another advantage of flexible drives, especially of belt drives, is that they reduce vibration and shock transmission.

SELECTION OF A BELT DRIVE

Following are the various important factors upon which the selection of a belt drive depends:

- 1. Speed of the driving and driven shafts.
- 3. Power to be transmitted,
- 5. Positive drive requirements,
- 7. Space available, and

- 2. Speed reduction ratio,
- **4.** Centre distance between the shafts.
- 6. Shafts layout,
- 8. Service conditions.

TYPES OF BELTDRIVES

The belt drives are usually classified into the following three groups:

- **1. LIGHT DRIVES.** These are used to transmit small powers at belt speeds upto about 10 m/s, as in agricultural machines and small machine tools.
- **2.MEDIUM DRIVES.** These are used to transmit medium power at belt speeds over 10 m/s but up to 22 m/s, as in machine tools.
- 3.HEAVY DRIVES. These are used to transmit large powers at belt speeds above 22 m/s, as in compressors and generators.

TYPES OF BELTS

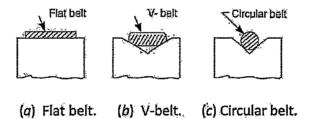


Fig. 3.1. Types of belts.

Though there are many types of belts used these days, yet the following are important from the subject point of view:

- 1. FLAT BELT. The flat belt, as shown in Fig. 3.1 (a), is mostly used in the factories and workshops, where a moderate amount of power is to be transmitted, from one pulley to another when the two pulleys are not more than 8 metres apart.
- 2. V-BELT. The V-belt, as shown in Fig. 3.1 (b), is mostly used in the factories and workshops, where a moderate amount of power is to be transmitted, from one pulley to another, when the two pulleys are very near to each other.
- 3. **CIRCULAR BELT OR ROPE.** The circular belt or rope, as shown in Fig. 3.1 (c), is mostly used in the factories and workshops, where a great amount of power is to be transmitted, from one pulley to another, when the two pulleys are more than 8 meters apart.

If a huge amount of power is to be transmitted, then a single belt may not be sufficient. In such a case, wide pulleys (for V-belts or circular belts) with a number of grooves are used. Then a belt in each groove is provided to transmit the required amount of power from one pulley to another.

MATERIAL USED FOR BELTS

The material used for belts and ropes must be strong, flexible, and durable. It must have a high coefficient of friction. The belts, according to the material used, are classified as follows:

LEATHER BELTS. The most important material for the belt is leather. The best leather belts are made from 1.2 metres to 1.5 metres long strips cut from either side of the back bone of the top grade steer hides. The hair side of the leather is smoother and harder than the flesh side, but the flesh side is stronger. The fibres on the hair side are perpendicular to the surface, while those on the flesh side are interwoven and parallel to the surface. Therefore for these reasons, the hair side of a belt should be in contact with the pulley surface, as shown in Fig. 3.2. This gives a more intimate contact between the belt and the pulley and places the greatest tensile strength of the belt section on the outside, where the tension is maximum as the belt passes over the pulley.

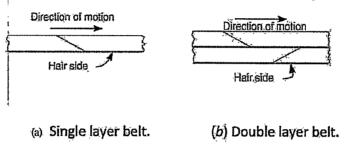


Fig. 3.2. Leather belts.

The leather may be either oak-tanned or mineral salt tanned e.g. chrome tanned. In order to increase the thickness of belt, the strips are cemented together. The belts are specified according to the number of layers e.g. single, double or triple ply and according to the thickness of hides used e.g. light, medium or heavy.

The leather belts must be periodically cleaned and dressed or treated with a compound or dressing containing neats foot or other suitable oils so that the belt will remain soft and flexible.

COTTON OR FABRIC BELTS. Most of the fabric belts are made by folding canvass or cotton duck to three or more layers (depending upon the thickness desired) and stitching together. These belts are woven also into a strip of the desired width and thickness. They are impregnated with some filler like linseed oil in order to make the belts water proof and to prevent injury to the fibres. The cotton belts are cheaper and suitable in warm climates, in damp atmospheres and in exposed positions. Since the cotton belts require little attention, therefore these belts are mostly used in farm machinery, belt conveyor etc.

RUBBER BELT. The rubber belts are made of layers of fabric impregnated with rubber composition and have a thin layer of rubber on the faces. These belts are very flexible but are quickly destroyed if allowed to come into contact with heat, oil or grease. One of the principal advantage of these belts is that they may be easily made endless. These belts are found suitable for saw mills, paper mills where they are exposed to moisture.

BALATA BELTS. These belts are similar to rubber belts except that balata gum is used in place of rubber. These belts are acid proof and water proof and it is not effected by

animal oils or alkalies. The balata belts should not be at temperatures above 40° C because at this temperature the balata begins to soften and becomes sticky. The strength of balata belts is 25 per cent higher than rubber belts.

TYPES OF FLAT BELT DRIVES

The power from one pulley to another may be transmitted by any of the following types of belt drives:

OPEN BELT DRIVE. The open belt drive, as shown in Fig. 3.3, is used with shafts arranged parallel and rotating in the same direction. In this case, the driver A pulls the belt from one side (i.e. lower side RQ) and delivers it to the other side (i.e. upper side LM). Thus the tension in the lower side belt will be more than that in the upper side belt. The lower side belt (because of more tension) is known as **tight side** whereas the upper side belt (because of less tension) is known as **slack side**, as shown in Fig. 3:3.

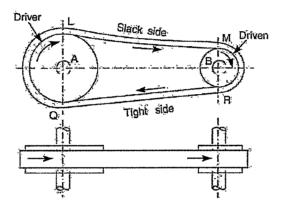


Fig. 3.3. Open belt drive.

CROSSED OR TWIST BELT DRIVE. The crossed or twist belt drive, as shown in Fig. 3.4, is used with shafts arranged parallel and rotating in the opposite directions.

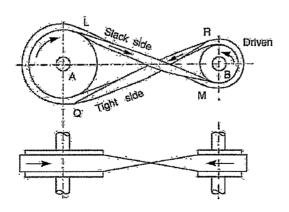


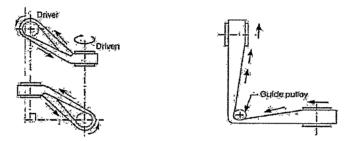
Fig. 3.4. Crossed or twist belt drive.

In this case, the driver pulls the belt from one side (i.e. RQ) and delivers it to the other side (i.e. LM). Thus the tension in the belt RQ will be more than that in the belt LM. The belt RQ (because of more tension) is known as tight side, whereas the belt LM (because of less tension) is known as slack side, as shown in Fig. 3.4.

A little consideration will show that at a point where the belt crosses, it rubs against each other and there will be excessive wear and tear. In order to avoid this, the shafts should be placed at a maximum distance of 20 b, where b is the width of belt and the speed of the belt should be less than 15 m/s.

QUARTER TURN BELT DRIVE. The quarter turn belt drive also known as right angle belt drive, as shown in Fig. 3.5 (a), is used with shafts arranged at right angles and rotating in one definite direction. In order to prevent the belt from leaving the pulley, the width of the face of the pulley should be greater or equal to 1.4 b, where b is the width of belt.

In case the pulleys cannot be arranged, as shown in Fig. 3.5 (a), or when the reversible motion is desired, then a quarter turn belt drive with guide pulley, as shown in Fig. 3.5 (b), may be used.

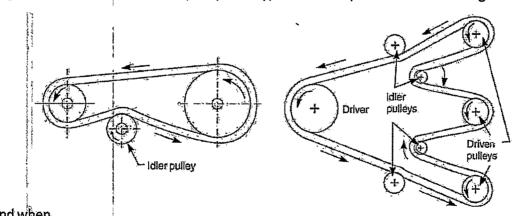


a) Quarter turn belt drive.

(b) Quarter turn belt drive with guide pulley

Fig. 3.5

BELT DRIVE WITH IDLER PULLEYS. A belt drive with an idler pulley, as shown in Fig. 3.6 (a). is used with shafts arranged parallel and when an open belt drive cannot be used due to small angle of contact on the smaller pulley. This type of drive is provided to obtain high velocity ratio



and when

1. Belt drive with single idler pulley. (b) Belt drive with many idler pulleys.

Fig. 3.6

the required belt tension cannot be obtained by other means.

When it is desired to transmit motion from one shaft to several shafts, all arranged in parallel, a belt drive with many idler pulleys, as shown in Fig. 3.6 (b), may be employed.

COMPOUND BELT DRIVE. A compound belt drive, as shown in Fig. 3.7, is used when power is transmitted from one shaft to another through a number of pulleys.

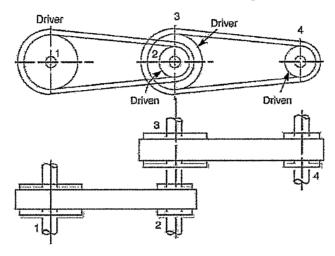


Fig. 3.7. Compound belt brive.

STEPPED OR CONE PULLEY DRIVE. A stepped or cone pulley drive, as shown in Fig. 3.8, is used for changing the speed of the driven shaft while the main or driving shaft runs at constant speed. This is accomplished by shifting the belt from one part of the steps to the other.

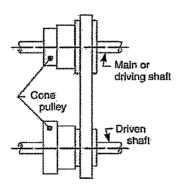


Fig. 3.8. Stepped or cone pulley Drive

FAST AND LOOSE PULLEY DRIVE. A fast and loose pulley drive, as shown in Fig. 3.9, is used when the driven or machine shaft is to be started or stopped when ever desired without interfering with the driving shaft. A pulley which is keyed to the machine shaft is called fast pulley and runs at the same speed as that of machine shaft. A loose pulley runs freely over the machine shaft and is incapable of transmitting any power. When the driven shaft is

required to be stopped, the belt is pushed on to the loose pulley by means of sliding bar having belt forks.

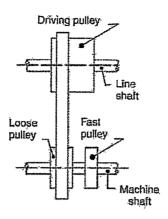


Fig. 3.9. Fast and loose pulley drive.

WORKING CHARACTERISTICS OF FLAT BELTS

Flat belts are made of leather, cotton fabrics, rubber, synthetic fibres etc. Their maximum tensile strength(i.e.ultimate strength) varies as follows:

- 1. For leather belting it varies from 20 N/mm² to 35 N/mm²
- 2. For cotton or fabric belting it varies from 35 N/mm² to 40 N/mm²

By assuming that the factor of safety as 8 to 10,the working stress may be considered from 1.75 N/mm² to 2.8 N/mm², which may take the belt to have the life of about 15 years.

It has been found that for efficient transmission of power the belt speed of 17.5 m/s to 22.5 m/s may be adopted if the belt speed is increased, the centrifugal force is also increased which tries to pull the belt away from the pulley which results the decreasing of power transmitting capacity of the belt.

When discussing about the working characteristics of flat belts, some technical parameters must be familiarized. The following some of such parameters

VELOCITY RATIO OF BELT DRIVE

The velocity ratio(speed ratio) is the ratio of the speed of the driven pulley to the speed of driver pulley.

Let d, n = Diameter and speed in r.p.m of driving pulley (usually smaller size)

D, N = Diameter and speed in r.p.m of driven pulley (usually bigger size)

The distance travelled per unit time(i.e. linear velocity) is same for both pulleys. (when the belt is no slip).

$$V = \frac{\pi dn}{60} = \frac{\pi DN}{60}$$

dn=DN

The speed ratio(i.e velocity ratio) = V.R = $\frac{N}{n} = \frac{d}{D}$

Suppose the thickness of belt is to be considered, then the effective diameters are

For driving pulley $d^{l} = d + (2 \times \frac{t}{2}) = d + t$ and driven pulley $D^{l} = D + (2 \times \frac{t}{2}) = D + t$

Then $i = \frac{N}{n} = \frac{d^{I}}{D^{I}} = \frac{d+t}{D+t}$

SLIP OF THE BELT

since the belt is to transmit power through friction, if must be given initial tension in order to hold the pulley with sufficient grip. Due to age or centrifugal tension caused on the belt, the grip may be slightly lost, Hence, because of less grip, during starting, the driving pulley alone rotates slightly without pulling the belt or the belt alone moves without rotating driven pulley. This type of difference of motions between the pulley and belt is termed as belt slip. The main draw back of the slip is that it will reduce the velocity ratio.

Let S1 = % of slip between driving pulley and belt = $(\frac{v_1 - v_S}{v_L})$

S2 = % of slip between driven pulley and belt = $(\frac{v_{b-v_2}}{v_b})$

Then total slip S = S1 + S2

The velocity ratio $i = \frac{N}{n} = \frac{d}{D} \left(1 - \frac{s}{100} \right)$ when t is neglected

Or $i = \frac{N}{n} = \frac{d+t}{D+t} (1 - \frac{S}{100})$when t is considered, where t = belt thickness.

CREEP OF BELT

When the belt passes from slack side to the tight side, a certain of the belt extends and it contracts again when the belt passes from the tight side to the slack side. Due to these changes of length, there is a relative motion between the belt and the pulley surfaces. This relative motion is termed as creep. The total effect of creep is reduce slightly the speed of the driven pulley or follower. Considering creep, velocity ratio is given by

$$\hat{\mathbf{I}} = \frac{N}{n} = \frac{d}{D} \left(\frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}} \right)$$

where $\sigma_1 \& \sigma_2$ =stress in the belt on the tight and slack side respectively

E = young's modulus for the material of the belt

Note: since the effect of creep is very small, therefore it is generally neglected.

CENTRIFUGAL TENSION

When the belt runs at lower speed, the initial tension given to the belt will be sufficient to keep the belt on the pulley with required grip, on the other hand, if the belt speed increases, due to centrifugal action, the belt will try to fly off from the pulley. At the same time, the tensions at the tight side and slack side will increase. The force applied on the shaft due to centrifugal action is called as centrifugal tension.

Let T_1^i = Tension in the tight side

 T_2 = Tension in the slack side

 $T_c = centrifugal tension = mv^2$

It is known that, the total tensions at tight side and slack side are given by

$$T_{t1} = T_1 + T_c$$
 and $T_{t2} = T_2 + T_c$

Since the centrifugal tension depends on the belt velocity, at low speeds the centrifugal action and its tension may be neglected. But for the higher speeds, the centrifugal tension will be taken into account.

Le $T_{t1} = T_{1}$ and $T_{t2} = T_{2}$ at low speeds, and $T_{t1} = T_{1} + T_{c}$ and $T_{t2} = T_{2} + T_{c}$ high speeds.

Also since the centrifugal force tries to pull the belt away from the pulley resulting the decrease of power transmitting capacity, the linear velocity of the belt is limited to 17.5 to 22.5 m/s, in order to control the centrifugal tension.

If μ is the coefficient of friction between the belt and pulley and θ is the angle of contact for driving pulley in radians, then it is found that the ratio of driving tensions is

$$\frac{T_1}{T_2} = e^{\mu\theta}$$
when the centrifugal tension (T_c) is neglected.

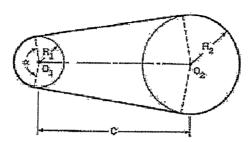
$$(T_1-T_c)/T_2-T_c)=e^{\mu\theta}$$
when the centrifugal tension (T_c) is considered.

LAW OF BELTING

Law of belting states that the centre line of the belt, as it approaches the pulley must lie in a plane perpendicular to the axis of that pulley, otherwise the belt will run off the pulley.

DESIGN OF FLAT BELTS

Flat Belt Drives



The essential parameters of flat belt like width, thickness, length and the type of belt are determined based on two methods.

- 1. Using fundamental formulas
- 2. Using manufacturer's catalogues

USING FUNDAMENTAL FORMULAS

When the driving pulley rotates the driven pulley by belt, the belt pulling side is known as tight side and the belt releasing side is known as slack side. If the centre distance and the selected materials of belt and pulleys are kept proper, the belts can have sufficient grip over the pulley without any slip and the power transmission is properly maintained.

For designing the belt based on fundamental formulae, we should know the tensions (i.e loads) on tight side and slack side of the pulleys, power-torque relationship, coefficient of friction between the contact surfaces of pulleys and belt, diameters of pulleys and so on. Some of such useful formulas are

1. Power transmitted by belt

$$P = (T_1 - T_2) v....N-m/s$$

Where T_1 = Tensions in the tight side in Newton's T_2 = Tensions in the slack side in Newton's \dot{V} = velocity of belt in m/s

2. Velocity of belt (v) = $\frac{\pi dn}{60000}$ m/s.

Where d = Diameter of smaller pulley in mm n= speed of smaller pulley in r.p.m

3. Ratio between the tensions of tight and slack side

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

Where,

 μ =coefficient of friction between the contact surfaces of pulley and belt θ = Arc of contact in radians

4. The relationships used to find out the length and arc of contact are common with manufacturer's catalogue.

USING MANUFACTURER'S CATALOGUES

In this method, the manufacturer's are producing certain types of belts whose widths and thickness have already been standardized by them. Their loading capacity (i.e belt rating) are also experimentally determined by the concerned manufacturers. They may also adopt certain safety factors like service factors, angle of contact factors for obtaining better design. To meet our requirement, we may select a particular belt based on working conditions and it is compared with their available data in practice.

Design of belts by this method is based mainly on two concepts:

1. How much power(i.e maximum power (or) Design power) to be transmitted.

2. What may be the power transmitting capacity(i.e.belt rating) of the selected belt. For determining the design power and belt rating, we must consider certain correction factors like service factors, arc of contact factors and so on. Some of the factors and data adopted by manufacturers are given in the jalal data book page 23.7.

Arc of contact

Consider the driving pulley and the driven pulley are connected by a flat belt as shown above. The angle subtended by the overlaying belt on the pulley is known as angle of contact or arc of contact (θ) .

Let d = Diameter of smaller pulley(driver)

D = Diameter of bigger pulley(driven)

C = centre distance between pulleys

Then Arc of contact factor(θ) for smaller pulley

For

i) Open belt drive
$$\theta = 180^{\circ} - (\frac{D-d}{c}) 60^{\circ}$$

ii) Cross belt drive
$$\theta = 180^{\circ} + (\frac{p+d}{c}) 60^{\circ}$$

iii) Quarter turn belt drive
$$\theta = 180^{\circ} - (\frac{d}{c}) 60^{\circ}$$

II) Load rating (or) Belt capacity

The load rating have been developed for 1800 of arc of contact, 10 m/s belt speed per mm width, and per ply and they are to be corrected for actual arc of contact, actual speed, actual plies and through these data the required width is determined.

Load rating per mm width per ply at 1800 arc of contact at 10 m/s belt speed:

'HI SPEED' 878 g duck belting: 0.023kw (or) 0.0314 h.p.

'FORT' 949 g duck belting : 0.0289 kw (or) 0.0392 h.p.

III) Length of belt(L)

Open belt drive L = 2C +
$$\frac{\pi}{2}$$
 (D + d) + $(\frac{D-\hat{d}}{4E})^2$

Cross belt drive
$$L = 2C + \frac{\pi}{2} (D + d) + (\frac{D + d}{4C})^2$$

Quarter turn belt drive
$$L = 2C + \frac{\pi}{2} (D + d) + (\frac{D^2 + d^2}{2C})$$

iv) Belt tensions: If the length of belt, determined by using the above relations is used as it is, then the belt will not transmit the power properly due to slip of belts(or) insufficient grip and hence certain amount of length should be reduced from the above calculated length to overcome the above deficiency. The reduction of length is based on the type of plies.

For,

- 1. Belt of 3 plies 1.5 % of L
- 2. Belt of 4,5 &6 plies1% of L
- 3. Belt of 8 plies 0.5 % of L

Should be reduced in order to get belt tensions.

v) Pulley widths:

Generally the pulleys should be slightly wider than belt widths pulleys width table is given in jalal data book 23.9.

DESIGN PROCEDURE FOR FLAT BELT BASED ON MANUFACTURER'S TABLES

 From the given conditions like power, type of working conditions, diameter of pulleys, speed ratio etc, determine maximum power(i.e design power) as

Design power = Rated power(i.e given power) x service factor(i.e load correction factor) x Arc of contact factor.

Select service factor based on nature of load and applications (jalal data book page 23.7 and table 23.6) and choose arc of contact factor based on contact of the belt on the smaller pulley jalal data book 23.7.

- 2. Decide the type of belt (i.e number of plies of the belt) depending upon the belt speed and diameter of smaller pulley, JBD -23.8
- 3. Then calculate the belt rating(i.e power transmitting capacity per mm width) for the above said belt using corresponding ralations.
- 4. Find the required width by dividing the design power by belt capacity, and adopt the next standard available width.
- 5. Determine the length of belt based on type of drive and then reduce certain amount of length so as to get initial tension (i.e. to make the belt to hold the pulley firmly for getting proper grip).
- 6. Find out pulley dimensions and draw the arrangement of belt drive.

Note: if calculated width is not available in the manufacturer's table, hen higher capacity belt(i.e belt of higher ply) may be selected.

FLAT BELT PULLEYS:

The flat belt pulley is a cylindrical member, similar to fly wheel in which thickness is small and width is large as compared to flywheel. The flat belt is laid on the pulley and the power is transmitted from the driving shaft to driven shaft by the belt drive set up similar to flywheel, pulleys are also made intoi) solid or web type and ii) Rim and hub type models. If the size of thepulley is to be very large, it is made into spilitted rim model. The rim type pulley is shown below.

MATERIALS FOR PULLEY

The material for making pulley depends on the size and velocity of pulley, working environments etc. The commonly used pulley materials are i) cast-iron ii) steel iii) wood and iv) compressed papers etc.

DESIGN OF CAST-IRON PULLEYS FOR FLAT BELT DRIVE

In a belt drive, since the speed or velocity ratio is the inverse ratio of the diameters of driving and driven pulleys, selection of pulley diameters must be proper to have the required velocity ratio. Also since the pulley resembles with flywheels the design of pulley is very similar to flywheel design. The parameters of the main parts of pulley such as pulley rim, hub and arm are determined as follows.

DIMENSIONS OF PULLEY RIM

- i) Pulley diameter:
- ii) The rim diameter (i.e pulley outer diameter) can be determined from the velocity ratio and centrifugal force. Due to centrifugal force acting on the rim, the rim has the tendency to expand which is restrained by the arms. The tensile stress induced in the arm due to centrifugal force is given by

$$\sigma_t = S_t = \rho v^2$$

where ρ =Density of rim material in kg/m3 =7200 kg/m3 for cast iron

v=Linear velocity of rim in m/s

$$= \frac{\pi DN}{60}$$
 where D = Diameter of pulley in m

N =speed of pulley in r.p.m

The diameter obtained from the above relation can be standardized from the dimensions given in jalal data book 23.13 table in 23.9 page.

iii) Rim width(B)

The width of rim is decided by the width of belt. If the belt is known, the rim width(i.e pulley width) is given some extra dimension as represented in table 23.11 jalal data book 23.9 page for safe operation. Some standard dimensions for pulley width have been presented in table 23.12 in JBD 23.9

iv) Rim thickness

The thickness of pulley rim may approximately be taken as

$$t = \frac{D}{200} + 3 \text{ mm(single belt)}$$

$$t = \frac{D}{200} + 6 \text{ mm(double belt)}$$

where D = Diameter of pulley in mm.

v) Crown thickness(h)

Pulley is provided a slight conical shape or convex shape in its rim's outer surface in order to prevent the belt from running off the pulley due to centrifugal force. This is known as crowning of pulley. Usually the crowning height(i.e crown thickness) may be of 1/96 of pulley face width. The standard values of crown thickness are given in table 23.13 & 23.15 in jalal data book page 23.9 & 23.10.

DIMENSIONS OF ARM

1) Number of arms

The number of arms may be taken as N =4 for pulley diameters upto 450 mm N=6 for pulley diameters over 450 mm.

When the pulley diameter is less than 200 mm, the pulley is usually made with solid disc(i.e web type) instead of providing arms.

3. Cross-section of arms

Cross section of arm is usually elliptical with major axis(x) equal to twice the major axis(y). The cross-section dimensions of the arm are determined by treating the arm as a cantilever which is assumed to be fixed at the hub end and carrying a concentrated load at the tim end. The length of cantilever is taken equal to the radius of pulley. Also it is assumed that at any given time, the power is transmitted from the hub to rim to hub through only half the total number of arms.

> Let T-Torque transmitted R = Radius of pulley N= Number of arms

The tangential force acting on one am at the rim end due to tirque is given by

Wor
$$F = \frac{2T}{nR}$$

 $W \text{ or } F = \frac{2T}{nR}$ Bending moment applied at the hub end due to the tangential force,

$$M = F \times R = \frac{2T}{nR} \times R = \frac{2T}{n}$$

Section modulus for the arm is $Z = \frac{\pi}{32} x^2 y$

The induced bending stress Sb = $\frac{M}{Z} = \frac{2T}{nZ}$

By equating this induced stress with the allowable bending stress, the dimensions of arm x and y can be determined. The allowable bending stress for cast-iron material is taken as 16 to 20 N/mm².

The arms are tapered from hub to rim. The tapers usually 1/48 to 1/32.

Dimensions of hub:

Let ds = Diameter of shaft

Di = Inner diameter of hub

D0 = outer diameter of hub

L= length of hub

Then.

Di = ds and do=1.5 ds + 25 mm but not more than 2ds

L=(2/3 to 1)B, where B is the width of pulley.

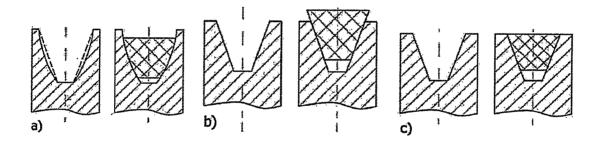
Note: The maximum speed of cast iron pulley is taken as follows

- a) For solid pulley with flat or crown face:1500 m/min
- b) For split pulley with flat or crown face: 1000 m/min

V-BELT DRIVES

v-belts are another type of flexible connectors for transmitting power from one pulley to another whose centre distances are approximately upto 3 meters. Their cross-section is trapezoidal or similar to a wedge. Generally v-belts are made endless(i...e each belt is made in a circular form) with various cross-section which may be differentiated by various grades. The belts are adopted on grooved pulley, grooves being v-shaped or having two inclined sides with flat bottom. The contact between the belts & pulleys is obtained in the inclined surfaces of pulleys in contrast with flat belt where contact is obtained at the top surface of the pulley.

A Properly installed should v-belt should fit tightly against the sides of the pulley grooves without projecting beyond the rim or touching the bottom of the groove. The correct method of mounting a wedge -shaped belt in a grooved pulley is shown in fig below. In any grade belt, the included angle between the inclined surfaces is about 40°.



Comparing to flat belt; v-belt is adopted for positive drive (i.e drive without slip).

MATERIALS USED

Usually V-belts are made of cord and fabric, impregnated with rubber, the cord material being cotton or rayon.

Types of V-belts

Generally v-belts are classified into various grades based on their power transmitting capacity as A,B,C,D and E. The cross-sectional areas are in inceasing order from A to E.

Depending upon the places of applications, v-belts are manufactured into single v-belts, multiple v-belts and ribbed belts.

The multiple v-belt and ribbed belt are manufactured in such away that they may have a number of separate single v-belts joined together to act as a single unit, for transmitting very high power, a calculated number of single v-belts or a suitable multiple v-belt drive, all the belts should be stretched at the same rate. (i.e drive with many number of single v-belts). Then only we can say that the load or (power) is distributed evenly between the belts. If any one belt is worn-out, all the belts should be replaced by new belts instead of changing the broken belt alone so as to have the even distribution of load and constant velocity of all belts . such a deficiency may be avoided in the case of multiple v-belt. Drive for getting clear positive drive, toothed or timing belt is generally preferred.

Advantages and Disadvantages of V-belt Drive Over Flat Belt Drive

Following are the advantages and disadvantages of the V-belt drive over flat belt drive.

ADVANTAGES

- 1. The V-belt drive gives compactness due to the small distance between the centres of pulleys.
- 2. The drive is positive, because the slip between the belt and the pulley groove is negligible.
- 3. Since the V-belts are made endless and there is no joint trouble, therefore the drive is smooth.
- 4. It provides longer life, 3 to 5 years.
- 5. It can be easily installed and removed.
- 6. The operation of the belt and pulley is quiet.
- 7. The belts have the ability to cushion the shock when machines are started.
- 8. The high velocity ratio (maximum 10) may be obtained.
- 9. The wedging action of the belt in the groove gives high value of limiting ratio of tensions. Therefore the power transmitted by V-belts is more than flat belts for the same coefficient of friction, arc of contact and allowable tension in the belts.
- 10. The V-belt may be operated in either direction with tight side of the belt at the top or bottom. The centre line may be horizontal, vertical or inclined.

DISADVANTAGES

- 1. The V-belt drive cannot be used with large centre distances.
- 2. The V-belts are not so durable as flat belts.
- 3. The construction of pulleys for V-belts is more complicated than pulleys for flat belts.
- 4. Since the V-belts are subjected to certain amount of creep, therefore these are not suitable for constant speed application such as synchronous machines, and timing devices.
- 5. The belt life is greatly influenced with temperature changes, improper belt tension and mismatching of belt lengths.
- 6. The centrifugal tension prevents the use of V-belts at speeds below 5 m/s and above 50m/s.

DESIGN OF V-BELTS USING BASIC FORMULAS

Similar to flat belts, v-belts are also designed based on i) fundamental formulas and ii) manufacturers catalogues, since v-belts vary considerably in cross-section and amount of reinforcement, the design of v-belts is usually based on the tables given by the manufacturers than the first method (i.e using fundamental formulas) any how, before learning in detail about the design using manufacturers tables, it is better to have some basic ideas about the design using fundamental formulas.

1. Ratio of driving tensions

$$\frac{T_1}{T_2} = e^{\mu\theta/\sin(\alpha)/2}$$

Where T1 and T2 are tensions at tight and slack side respectively

 θ = Angle of contact in radians

 α = Angle subtended by sides of v-belts Power transmitted by a belt n.S.I or M.K.S unit as

$$P = (T_1 - T_2) v....N-m/s$$

Where T_1 = Tensions in the tight side in Newton's

T₂ = Tensions in the slack side in Newton's

V = velocity of belt in m/s

DESIGN OF V-BELTS USING MANUFACTURER'S TABLE

The manufacturers produce v-belts in different grades as A, B, C, D, e which can be used to transmit different ranges of power, the range being in increasing order from A to E because of increasing order of area of cross-section in the same series.

One important point is that when selecting flat belt, we should give preference to thin wider belt for optimum power transmission because the area of contact between belt and pulley is more in thin wider belt than thick narrow belt. But in the case of v-belt, thick belt is more preferred to thin belt which may reduce the total number of belts to be operated for transmitting a particular quantity of power.

Here also, the design of v-belt depends on two concepts

- 1. Design power (i.e Total power after considering safety factors or correction factors)
- 2. Belt rating (i.e power transmitting capacity of one belt)

Consider a v-belt drive as shown in fig below

Required to transmit the power of P from one pulley of diameter d to another pulley of diameter D the pulleys being situated at a distance of C. The work may be intermittent or continuous.

For designing such a belt we may follow certain steps.

- 1. At first based on amount of power to be transmitted select the type of belt from A to E grades JBD page 24.3, table 24.1
- 2. Calculate design power using the relation as

For obtaining the above correction factors, find out the service conditions, arc of contact and pitch length and then choose suitable factors from page 24.3 tables JBD 24.2,24.3,24.4 respectively. Since the v-belt can be operated as open belt type we can make use of the following formula for finding pitch length and arc of contact. i.e The pitch length

$$L = 2C + \frac{\pi}{2} (D + d) + (\frac{(D - d)^2}{4C})$$

Arc of contact = drive =
$$\theta = 180^{\circ} - (\frac{D-d}{c}) 60^{\circ}$$

- 3. Note the inside length corresponding to pitch length from table 24.4 for belt specifications.
- 4. Determine the belt rating (i.e power transmitting capacity of one belt) using suitable formula adopted by the manufacturers given in table 24.5 or 24.6 to 24.10.

- 5. Obtain number of belts required to transmit the entire design power as number of belts = Design power Beltrating
- 6. Correct the centre distance according to the selected pitch length using formulas (A) which have been given below of table 24.11 and give initial tension to belt.(Refer condition B, below of table 24.11)
- 7. Also determine parameters of v-groove pulleys using table 24.12.

DESIGN OF V-GROOVED PULLEY

The pulley required for v-belt drive must have sufficient number of v-grooves in its outer surface in order to fit the v-belts in the correct position. Mostly cast-iron or steel are adopted as the materials for v-belt pulley.

The dimensions of the v-groove of the pulley is decided by the dimensions of the v-belt. The width of pulley depends on the number of belts laid on the pulley. The diameter of pulley is found out from the linear velocity of the belt nd the velocity ratio. The number of arms, their cross-sections and hub dimensions are determined similar to flat belt pulley. Some standard values for various parameters of v-groove are given in table 24.12.

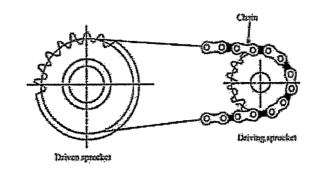
Arc of contact angle =
$$2 \cos^{-1} \frac{D-d}{2C} = 180^{\circ} - 60^{\circ} \frac{D-d}{2C}$$

D = pitch dia of larger pulley, d = pitch dia of smaller pulley, C = centre distance

CHAIN DRIVES

A chain drive is a mechanical device which belongs to the category of drives with intermediate link is obtained by chains. It may also be considered to be intermediate between belt and gear drives, in that it has features in common with both.

Chains are suitable for long as well as short centre distance drives and give a more compact drive than is possible with belt.



APPLICATION

Modern chain drives are employed in these places where we require velocity ratios upto 10, chain velocities upto 20 m/s and power ratios upto 150 kN.

Chain drives are quite extensively used in transportation machineries like motor cycles, bicycles, automobiles and conveyors. And technological machines like agricultural machinery, oil well drilling rigs, machine tools.

ADVANTAGES

- 1. The chain drives are having more power transmitting capacities compared with belt drives when the centre distance between the shafts is large(i.e.5 to 8 meters).
- 2. Their efficiency if higher (85%)
- 3. Their small size compared with V-belt for particular power transmission provides a compact set up.
- 4. The chain drives exert less load on the shaft since no initial tension is required as compared with belt drives.
- They can be operated for wide range of centre distances of power transmitting sprockets.
- 6. One chain can be operated to transmit power to several numbers of sprockets.
- 7. Maintenance is very easy because the chains can easily be repaired or replaced.

DISADVANTAGES

- 1. Design of chain drive is more complicated.
- 2. The operation of chain drive is noisy.
- 3. Their production cost is high.
- 4. They need carful maintenance by providing housing to the chains to save them from dirst.
- 5. They require more accurate assembly of shafts than belt drives.

CLASSIFICATION

1. POWER TRANSMITTING CHAINS

Treated for transmitting mechanical energy from one shaft to another.

2. HAULING CHAINS

Used for carrying loads in conveying machinery.

3. LOADING OF LIFTING CHAINS

Served for suspending, hoisting and lowering loads, mainly in material handling machineries.

CLASSIFICATION OF POWER TRANSMITTING CHAINS

- a. BASED ON THE TYPES OF CHAINS EMPLOYED
 - 1. Roller chain
 - 2. Bush chain
 - 3. Silent chain
- b. BASED ON THE NO OF CHAINS REQUIRED
 - 1. Single row chain
 - 2. Multi row chain
- **°c. BASED ON THE NO OF DRIVEN SPROCKETS**
 - 1. Normal type drive (i.e. one driven sprocket)
 - 2. Special type drive(i.e. several driven sprocket)

COMPONENTS OF CHAIN DRIVE

The essential components of a chain drive area) chains b) sprockets c) chain housing and d) slack adjusters.

a) CHAINS:

The chain is main element of drive which determine its reliability and service life, A

chain consists of rigid links which are hinged together to provide flexibility for wrapping action around the driving and driven wheels. The wheels having surfaces which are shaped to confirm to the type of chain used are called sprockets. The design dimensions and mechanical; characteristics of chains are standardized. The strength of chain is determined by its ultimate strength established by the manufacturer's. The schematic diagram for various types of chains are show.

Regarding the design parameters, the pitch (p) which is linear distance between the centers of consecutive rollers and the width (b) which is the space between the inner link plates, are considered as the main geometrical characteristics of chains.

usually the driving chains such as roller, bush and silent chains are made in various standards. Among them, the simplex, duplex and triplex chains are commonly used. Some times chains of four, five and six rows are made.

A roller chain consists of alternate inner and outer links connected by hinged joints. Each joint comprises a pin of diameter(dp) pressed into the outer plates and bush secured in the holes of the inner plates. Roller safeguards the sprocket's teeth against wear. The diametral area or projection area of the connect surface for such a joint is given by

$A = D_p.b$

Where b is the width of inner link plates.

BUSH CHAINS

Bush chains differ from roller ones in that they have no rollers. Such a chain, is lighter in weight, but the absence of rollers intensifies the wear of teeth on the sprocket's due to sliding friction between the teeth and bushes.

When heavy loads are to be transmitted, chains with a larger pitch may be employed. In this case the sprockets have larger diameters. To diminish their diameters and reducing chain pitch values, use of multi-row chains.

A silent chain consists of a series of toothed plates pinned to gether in rows across the width of the chain. The structure of silent chains are more intricate, more expensive and require a good maintenance.

The materials used for making different elements of chain are as follows, plates are made of cold-rolled band of medium carbon or alloy steels. Bent plates are made of alloy steels. The parts of chain-pins, bushings and inserts are made of carburizing steels and are hardened to 50-65 RC. The life of chains can be prolonged by chroming the chain joints.

b) SPROCKETS:

1

The operating capacity of a chain drive largely depends on the quality of the sprockets. The sprockets are made of cast-iron or hardened steel. The teeth of a sprocket are shaped depending on the type of chain. The small size sprocket is known as pinion-sprocket and the big size sprocket is known as wheel sprocket.

$$\alpha = \frac{360}{Z}$$
d= pitch circle dia of sprocket = $\frac{P}{\sin(\frac{\alpha}{z})}$

c) CHAIN HOUSING

The housing is a cover made of thin plates and it protects the drive from dust and dirt, preserves grease and damps the noise of the drive. The housing should be as small as possible but should not interfere with adjustment of the distance between the shafts when the chain becomes stretched.

d) SLACK ADJUSTERS:

Chain sag is regulated and the required tensions are ensured by means of movable bearings known as slack adjusters. The movable bearings are in the form of slides in which sprocket shaft is installed.

FAILURES OF CHAIN DRIVES

The chain drive may fail due to the following causes.

- 1. Wear in the joints leading to elongation of the chain and its faulty engagement with the sprockets. The allowable elongation is 1.5 to 2.5%.
- 2. Wear of the sprocket teeth.
- 3. Turning of pins and bushing in the plates where they are press fitted is a frequently encountered reason for the failure of roller chains.
- 4. Fatigue failure of the plates at the eyes.
- 5. Chipping and breaking of the rollers.
- 6. Poor lubrication and improper maintenance.

DESIGN OF DRIVING CHAINS

Similar to belts the driving chains or power transmitting chains are designated in different ways by different concerns.

As per Indian standard (IS 2403 – 1964), roller chains are designated by the roller diameter and number of stands. For example 10.05 IS:2403 -1964 stands for single stand (simplex) chain of 10.16 mm roller diameter. similarly 12.0 D IS:2403 -1964 stands for double stand (duplex) chain of 11.90 mm roller diameter.

According to International standard organization (ISO) chains are designated by their roller dia followed by the type of chain and the number of stands. For example 10A -1 represents the chain of 10.16 mm, roller diameter and A type single stand (simplex) chain. Similarly 10A -2 is the double stand (duplex) of above category and so on.

According to rolon chain manufacturers, chains are designated by the number of stands followed by the chain numbers. For example, R50, DR50 and TR50 are simplex, duplex

and Triplex types of roller chains whose chain number is 50. Similarly B35, DB35 and TB35 are simplex, duplex and triplex types of bush chains whose chain number is 35.

DISGN CRITERIA OF CHAIN DRVES

The chain drives can be employed to transmit from a fraction of kilowatt to 3500 kw, but in most cases upto 100 kw. When selecting a suitable chain to transmit the design power, when should consider the characteristics of certain parameters which are essential in the design point of view. The following are some of such parameters:

1. Chin velocity

The velocity of chain and the speed of rotation of sprockets are limited by chain wear which may increase with velocity. Hence the optimum chain velocity may be taken about 15 m/s. For high speed drives with high quality chains and proper lubrication, the velocity may be adopted up to 30 m/s.

The average velocity of chain (v) is given by

The speed of rotation of the sprockets is limited by the impact stresses of the chain on the sprockets.

2. Speed ratio:

The speed ratio is decided by the allowable overallsize of the drive, angle of contact (arc of meshing the chain on the smaller sprocket) and the number of teeth. Susually the speed ratio may taken upto 10, it is found out from the condition of quality of average velocities of the chain on the sprockets as

$$Z_1n_1p = Z_2n_2p$$
 (since chain pitch is constant)

Speed ratio
$$i = \frac{n1}{n2} = \frac{Z2}{Z1}$$

Where n1, z1 = speed & number of teeth of small sprocket(i.e pinion sprocket)

N2, z2 = speed & number of teeth of big sprocket(i.e wheel sprocket)

3. Number of teeth of sprockets:

The number of teeth of sprockets is limited by the wear of chain joints, dynamic(impact)

loads and also the noise made by the drive. The less the number of teeth, the greater the wear of the chain because the angle of chain engagement with sprocket is $(\frac{360}{Z})$ and hence more impact load on less numbered teeth sprocket. The minimum number of teeth on sprockets in power drives with roller chains is Zmin =19 to 23 for speeds, 17 to 19 medium speeds and 13 to 15 for low speeds.

Due to chain wear, the chain may elongate and hence it may shift outward upon the sprocket teeth profiles. The smaller the angular pitch of the sprocket (angle between adjacent teeth) the greater the outward shift. This outward shift limits the maximum number of teeth on sprocket, which for roller chain is taken from 100 to 120.

4. Distance between the sprocket axes:

The minimum centre distance (i.e distance between the axes of sprockets) is determined from the condition that the angle of contact of the chain with the small sprocket should be at least 1200 usually the centre distance may be specified in terms of pitches.

The optimum centre distance is given by

A = (30 to 50)p where p = chain pitch and the maximum centre distance is $a_{\text{max}} \leq 80p$.

5. Chain pitch:

This is the principal parameter of chain drive. Chains with larger pitch have a higher "load carrying capacity" but allow considerably low speed of rotation and operate with higher dynamic loads and noise. Usually a chain with the minimum allowable pitch may be selected for the given load, the pitch value may be selected as $\frac{\alpha}{80} \le p \le \frac{\alpha}{25}$ where " a " is the centre distance.

6. Arrangement of chain drives:

Chain drives are arranged so that the chain travels in a vertical plane, the relative height position of the driving and driven sprockets being arbitrary. The optimum position of the line of centers is horizontal or inclined at an angle up to 450 to the horizontal. Vertical arrangement of the chain drive requires more careful adjustment of the chain tension because the sagging of the chain in this case does not provide for self tensioning.

SELECTION OF CHAINS- INFLUENCING FACTORS

In accordance with the principal criterion of chain drive performance-wear resistance of chain joints, the load carrying capacity of a chain can be determined from the condition that the pressure developed in the joints must not exceed the allowable value.

Usually chains are manufactured in different sizes with different structureres such as simplex, duplex, and triplex chains and so on. Each chain is having a criterion amount of load carrying capacity, among these various chains, for transmitting the given power, a suitable

chain is selected based on its minimum breaking strength required and the induced stress on the bearing area of the selected chain. To overcome the chain failure, the selected chain should have more strength than the breaking load and the induced stress should be less than the allowable value.

Similar to belt drives, manufacturers formulated a certain method to select a suitable chain base on experimental data which are given in the following tables.

Let P = Power to be transmitted in watts

Q = Breaking load required for the chain in Newton's

V = chain velocity in m/s

Kn = Factor of safety

Ks = service factor

o= Allowable bearing stress in N/mm²

A = projected bearing area in N/mm²

The power, that can transmitted (i.e safe power) on the basis of breaking load is given by

$$P = \frac{Qv}{k_n k_s}$$
 watts

On the basis of allowable bearing stress, the power to be transmitted is given by

$$P = \frac{\sigma A v}{k_s}$$
 watts

The required service factor (ks) may be determined based on various working conditions, such that

$$Ks = K1.K2.K3.K4.K5.K6$$

Where K1 = load factor

K2= Factor for distance regulation

K3 = Factor for centre distance of sprockets

K4 = Factor for the position of the sprockets

K5 = Lubrication factor

K6 = Rating factor

The values of factors in various conditions are as follows:

Load factor(K1)

Constant load

1.0

Variable load or load with mild shocks

1.25

Variable load or load with heavy shocks

1.5

Factor fro distance regulation(K2)

Adjustable supports 1.0

Drive using idler sprocket 1.1

Fixed centre distance 1.25

Factor for centre distance of sprockets (K3)

$$\frac{Lp}{Z_{14}Z_{2}} > 1 \text{ or } a_{p} < 25P$$
 1.25

$$\frac{Lp}{Z1+Z2}$$
 = 1.5 or a_p = 30 to 50P 1.0

$$\frac{Lp}{Z1+Z2} \ge 2.0 \text{ or } a_p < 60 \text{ to } 80P$$
 0.8

Where Lp = Length of chain in multiplies with pitch

Factor for the position of sprockets (K4)

Lubrication factor (K5)

Rating factor(K6)

Single shift of 8 hours a day K6=1.0

Double shift of 16 hours a day K6=1.25

Continuous running K6=1.5

SELECTION PROCEDURE FOR CHAIN DRIVE

According to manufacturer's method, the following procedure can be adopted to select an appropriate chain for transmitting given power.

 Depending upon the amount of power to be transmitted and other working conditions such as available space, chain speed, position of chain drive etc, select the type of chain like bush chain or roller chain.

- Assuming the centre distance between the chain sprockets in terms of pitches (usually from 30 to 50 P) determine the pitch of chain and adopt its standard value.
- 3. Determine the design power to be transmitted by considering safety factors as Pd =Px Ks
- 4. Calculate the developed load fro breaking the chain due to design power using the expression as

$$P = \frac{Qv}{k_n k_s}$$
 or $P = \frac{P.Ks.Kn}{v}$ watts

Where Kn = factor of safety which depends on speed of sprocket pinion and pitch. From table

Ks = service factor which depends on various operating conditions

Ks = K1.K2.K3.K4.K5.K6

V = chain velocity in m/s

$$=\frac{Z1n1p}{60000}$$
 m/s or $v=\frac{Z2n2p}{60000}$ m/s

Here Z1,Z2 are number of teeth of sprocket pinion and sprocket wheel, n1,n2 are the speeds in rpm of sprocket pinion and sprocket wheel, and p is the pitch of chain.

Q = Breaking load in Newton's (i.e the developed tension or force on the chain due to power transmission)

P = Rated power in watts.

5. For the determined pitch, choose a suitable chain from the table which should have more breaking (i.e resisting) strength than the above calculated breaking load.

Le (Q) > Q where (Q) is the strength of selected chain and Q is the load developed (also called as breaking load)

Due to design power.

6. Find out the actual factor of safety this selected chain using the relation as

$$\operatorname{Kn} = \frac{(Q)}{\sum F}$$
 where $\sum F = \operatorname{Ft} + \operatorname{Fc} + \operatorname{Fs}$ from table

Here (kn) = Actual factor of safety which should be greater than allowable value i.e (Kn) > Kn

 F_t = tangential force due to power transmission in Newton's = $(\frac{P}{r})$ where P is power in watts, v is the velocity of chain in m/s

F_c = centrifugal tension in Newton's

= $(\frac{\dot{W}u^2}{g})$ where W is the weight of chain per meter length in Newton's

Fs = tension due to sagging of chain in Newton's,

Fs =k.w.a where k is sagging coefficient from table and **a** is the centre distance in meters.

This table factor of safety is checked with the adopted value.

7. Determine the induced stress over the projected area of the chain using the relation as

$$\sigma = \frac{P_{dv}}{Av}$$
 or $\sigma = \frac{P_{cK_{Sv}}}{Av}$ N/mm²

- 8. Find the length of chain using the relation given in the table and provide allowance for initial sagging.
- 9. Evaluate the pitch diameter of pinion-sprocket (d1) and wheel-sprocket (d2) using

$$d_1 = \frac{p}{\sin(\frac{180}{21})}$$
 and $d_{21} = \frac{p}{\sin(\frac{180}{22})}$

10. Draw a neat sketch of chain drive with calculated specifications.

Note:

The empirical formula to determine the pitch of the chain in mm is given by

$$P \le 10 \left[\frac{60.67}{n^L} \right]^{2/3}$$

Where n^{i} = speed of small sprocket in rps(i.e. n^{i} = $n_{1}/60$)

CHORDAL (POLYGONAL) ACTION OF ROLLER CHAIN

In a chain drive, the smooth running of the chain, can be effected by the less number of sprocket-teeth. This type of unsmooth running is termed as chordal action of the chain. It can be explained as follows.

When the roller of a chain approaches the sprocket and has just, it has the radius r_c , known as chordal radius, as shown fig(a). When the roller travels through an angle of $\alpha/2$ (half of the pitch angle), the roller has the radius r, which is the pitch radius and it is cleared that $r > r_c$. For N rom, the chain velocity for radius r is $2\pi r_c N$.

Since the velocity due to r is greater than the velocity due to r_{cr} it is obvious that, the chain is undergoing from minimum velocity to maximum velocity for every rotation which may affect the smooth running. The chordal rise Δr sis given by,

$$\Delta r = r - r_c = r - r \cos\left(\frac{\alpha}{2}\right) = r(1 - \cos\left(\frac{\alpha}{2}\right))$$

This chordal rise $\Delta\eta$ can be reduced by increasing the number of teeth Z, because $\alpha=\frac{360}{Z}$. The effect of chordal action can be reduced to negligible if there are 25 or more teeth on the sprocket. In order to distribute wear evenly on all the chain links, it is the usul practice to have number of teeth on the smaller sprocket and an even number of pitches in the chain.

LINK CHAINS -AN OUTLINE STUDY

Link chains, also called as welded chains, are formed bending the required number of steel rods of specified length into oval shaped links and their ends are welded together after connecting each link with the other as shown in fig below. The dimensions of a link chain are its pitch(p), outside width(B) and the diameter of chain rod(d).

Depending upon the value of pitch, the link chains are classified in to

- a) Short link chains (when p≤ 3d)
- b) Long link chains (when p>3d)

Welded chains should be tested under load equal to one half the breaking load and no permanent set is permitted after the test.

Link chains are mostly used in some of the hand operated hoist mechanisms in which drum or pulley diameter(D) should not be less than 20d. In case of power mechanism D = 30d.

The general formula for selecting the welded chain in tension is Ps = $\frac{Pb}{fs}$

Where Ps = safe load to be carried by chain (or design load)

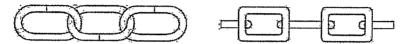
Pb =Breaking load

Fs = factor of safety (4 to 8)

The main advantages of link chains are i) good flexibility in all directions ii) simple design and the notable disadvantages are iii) Heavy susceptibility to jerk and overloads iv) Intensive wear of the links in the joints and sometimes sudden filura.

SILENT CHAIN AND SIGNIFICANCE

The silent chain resembles with toothed or timing belt. The silent chain, also known as inverted toothed chain is constructed from a series of flat plates. Each plate has two projections or teeth. Each link, as it entres the sprocket, pivotes about the pin of the adjacent link which is in contact with the sprocket. Thus the contact is gradual equal to almost three times the speed of simple roller chains. Even through the silent chain and timing belt, the power transmission capacity of silent chain is many times that of timings belt. The schematic diagram of a silent chain is shown in fig. The silent chain can be classified based on the type joint between the links such as i) Reynold chain and ii) Morse chain



(a) Chain with oval links.

(b) Chain with square links.

Fig. Hoisting and hauling chains.

The silent chains are mostly employed for high speed & high power applications. Because of the complicated structure of silent chain, its manufacturing cost is more and at the same time its run off from the sprocket(i.e shift) will be almost prevented. The silent chains are more heavier, more expensive and require more careful maintenance. Due to the above reasons, they have limited applications.

Ex1.

A belt is required to transmit 18.5 kW from a pulley of 1.2 m diameter running at 250rpm to another pulley which runs at 500 rpm. The distance between the centers of pulleys is 2.7 m. The following data refer to an open belt drive, = 0.25. Safe working stress for leather is 1.75 N/mm2. Thickness of belt = 10mm. Determine the width and length of belt taking centrifugal tension into account. Also find the initial tension in the belt and absolute power that can be transmitted by this belt and the speed at which this can be transmitted.

Data:

Open belt drive; N=18.5 kW; n_1 =500 rpm=Speed of smaller pulley; d_2 =1.2 m=1200 mm=D=Diameter of larger pulley; n_2 =250 rpm=Speed of larger pulley; C=2.7 m=2700 mm; μ =0.25; σ_1 =1.75 N/mm²; t=10 mm

(i) Diameter of smaller pulley

$$n_1 d_1 = n_2 d_2$$

 $500 \times d_1 = 250 \times 1200$

:.Diameter of smaller pulley d, = 600 mm = d

(ii) Velocity

$$v = \frac{\pi(D+t)n_2}{60.000} = \frac{\pi(1200+10)250}{60.000} = 15.839 \text{ m/sec.}$$

(iii) Centrifugal stress

$$\sigma_{\rm c} = \frac{\rm wv^2}{\rm g} \times 10^6$$

Assume specific weight of leather as 10×10⁶ N/mm³

$$\pi \sigma_{c} = \frac{10 \times 10^{-6}}{9810} \times 15.839^{2} \times 10^{6} = 0.25573 \text{ N/mm}^{2}$$

(iv) Capacity

Since coefficient of friction is same for both smaller and larger pulleys, capacity = $e^{i\theta}$,

$$\theta_{s} = \pi - \left\{ 2\sin^{-1}\left(\frac{D-d}{2C}\right) \right\} \frac{\pi}{180}$$

$$= \pi - \left\{ 2\sin^{-1}\left(\frac{1200 - 600}{2 \times 2700}\right) \right\} \frac{\pi}{180} = 2.92 \text{ radians}$$

$$\therefore e^{u0} = e^{0.25 \times 2.92} = 2.075$$

(y) Constant

$$k = \frac{e^{\mu\theta} - 1}{e^{\mu\theta}} = \frac{2.075 - 1}{2.075} = 0.52$$

(vi) Width of belt

Power transmitted per mm² area =
$$\frac{(\sigma_1 - \sigma_c)kv}{1000}$$

= $\frac{(1.75 - 0.25573)0.52 \times 15.839}{1000} = 0.01231 \text{ kW}$

(vii) Length of belt

Length of open belt
$$\hat{L} = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}(D\theta_L + \theta_s^2 d)$$

$$\theta_L = \pi + \left\{ 2\sin^{-1}\left(\frac{D-d}{2C}\right) \right\} \frac{\pi}{180}$$

$$= \pi + \left\{ 2\sin^{-1}\left(\frac{1200 - 600}{2 \times 2700}\right) \right\} \frac{\pi}{180} = 3.364 \text{ radians}$$

$$L = \sqrt{4 \times 2700^2 - (1200 - 600)^2} + \frac{1}{2} (3.364 \times 1200 + 2.92 \times 600)$$

$$\therefore L = 8260.96 \,\text{mm}$$

(viii) Initial tension

$$2\sqrt{T_o} = \sqrt{T_1} + \sqrt{T_2}$$

$$T_1 = \sigma_1 A = 1.75 \times 1503.18 = 2630.566 N$$

$$\frac{\sigma_1 - \sigma_C}{\sigma_2 - \sigma_C} = e^{\mu 0}; \frac{1.75 - 0.25573}{\sigma_2 - 0.25573} = 2.075 \text{ ; } \therefore \sigma_2 = 0.97586 \text{ N/mm}^2$$

$$T_2 = \sigma_2 A = 0.97586 \times 1503.18 = 1466.894 N$$

$$2\sqrt{T_o} = \sqrt{2630.566} + \sqrt{1466.894}$$

$$\therefore T_o = 2006.552 \text{ N}$$

(ix) Absolute power

For maximum power transmission

$$\sigma_{c} = \frac{\sigma_{1}}{3} = \frac{1.75}{3} = 0.5833 \text{ N/mm}^{2}$$
Also $\sigma_{c} = \frac{w}{g} v^{2} \times 10^{6}$

$$\therefore 0.5833 = \frac{10 \times 10^{-6}}{9810} \times v^{2} \times 10^{6}$$

$$\therefore v = 23.92 \text{ m/sec}$$

$$\therefore \text{Power transmitted } \text{mm}^{2} = \frac{(\sigma_{1} - \sigma_{c}) \text{kv}}{1000}$$

$$= \frac{(1.75 - 0.5833)0.52 \times 23.92}{1000}$$

$$= 0.0145 \text{ kW}$$

$$\therefore \text{Total absolute power} = \text{Area of } c/s \text{ of belt } \times \text{ power per mm}^{2}$$

$$= 1503.18 \times 0.0145 = 21.7961 \text{ kW}$$

$$\therefore \text{ Absolute power} = 21.8 \text{ kW}.$$

EX : 2

Select a V-belt drive to transmit 10 kW of power from pulley of 200 mm diameter mounted on an electric motor running at 720 rpm to another pulley mounted on compressor running at 200 rpm. The service is heavy duty varying from 10 hours to 14 hours per day and centre distance between centre of pulleys is 600 mm.

Data:

 $N=10\,\mathrm{kW};\ d_1=200\ \mathrm{mm}=d;\ n_1=720\,\mathrm{rpm};\ n_2=200\,\mathrm{rpm};\ C=600\,\mathrm{mm}$ Heavy duty 10 hours to 14 hours per day.

Solution :

i. Diameter of larger pulley

$$n_1d_1 = n_2d_2$$

 $720 \times 200 = 200 \times d_2$
 $\therefore d_1 = 720 \text{ mm} = D = \text{diameter of larger pulley}$

ii. Select the cross-section of belt

Equivalent Pitch diameter of smaller pulley $d_e = d_p F_b$ where $d_p = d_i = 200 \text{ mm}$

$$\frac{n_1}{n_2} = \frac{720}{200} = 3.6$$

From Table

when
$$\frac{n_1}{n_2} = 3.6$$

Smaller diameter factor $F_b = 1.14$

$$d = 200 \times 1.14 = 228 \text{ mm}$$

iii. Velocity

$$v = \frac{\pi d_1 n_1}{60000} = \frac{\pi \times 200 \times 720}{60000} = 7.54 \text{ m/sec}$$

iv. Power capacity

For 'C' cross-section belt

$$N^* = v \left[\frac{147}{V^{0.09}} - \frac{143.27}{d_e} - \frac{2.34v^2}{10^4} \right]$$
$$= 7.54 \left[\frac{1.47}{7.54^{0.09}} - \frac{143.27}{228} - \frac{2.34 \times 7.54^2}{10^4} \right]$$
$$N^* = 4.4 \text{ kW}$$

Number of bolts:

$$i = \frac{NF_a}{N*F_c.F_d}$$

for heavy duty 10-14 hours/day correction factor for service $F_a=1.3$

$$L = 2C + \frac{\pi}{2} (D+d) + \frac{(D-d)^2}{4C}$$
$$= 2 \times 600 + \frac{\pi}{2} (720 + 200) + \frac{(720 - 200)^2}{4 \times 600} = 2757.8 \text{ mm}$$

The nearest standard value of nominal pitch length for the selected C-cross section belt L = 2723 mm, Nominal inside length = 2667 mm,

For nominal inside length = 2667 mm, and C-cross section belt, correction factor for length Fe = 0.94

Angle of contact
$$\theta = 2\cos^{-1}\left(\frac{D-d}{2C}\right)$$

= $2\cos^{-1}\left(\frac{720-200}{2\times600}\right) = 128.64^{\circ}$

From Table

when $\theta = 128.64^{\circ}$

Correction factor for angle of contact $F_d = 0.86$ (Assume V-V belt)

$$\therefore i = \frac{10 \times 13}{4.4 \times 0.94 \times 0.86} = 3.655$$

EX.3

Select a roller chain drive to transmit power of 10 kw from a shaft rotating at 750 rpm to another shaft to run at 450 rpm. The distance between the shaft centers could be taken as 35pitches.

Data:N= 10 kw; n1 = 750 rpm; n2 = 450 rpm; C = 35 pitches

1. Pitch of chain

$$p \le 25 \left(\frac{900}{n_1}\right)^{\frac{2}{3}}$$
$$\le 25 \left(\frac{900}{750}\right)^{\frac{2}{3}}$$

< 28.23 mm

From table 21.64, the nearest standard value of pitch p = 25.4 mm

Select chain number 208 B

Breaking load $F_u = 17.9 \text{ kN} = 17900 \text{ N}$

Measuring load w = 127.5 N

2. Number of teeth on the sprockets

$$\frac{n_1}{n_2} = \frac{750}{450} = 1.667$$

From Table 21.60 for $\frac{n_1}{n_2} = 1.667$, select number of teeth on the sumaller sprocket $z_1 = 27$

Now
$$\frac{n_1}{\hat{n}_2} = \frac{z_2}{z_1}$$

$$\frac{750}{450} = \frac{z_2}{27}$$

Number of teeth on larger sprocket z,=45

3. Pitch diameter

$$d = \frac{p}{\sin\left(\frac{180}{z}\right)}$$

Pitch diameter of smaller sprocket $d_p = \frac{p}{\sin\left(\frac{180}{z_1}\right)} = \frac{25.4}{\sin\left(\frac{180}{27}\right)} = 218.79 \text{ mm}$

Pitch diameter of larger sprocket $d_2 = \frac{p}{\sin\left(\frac{180}{z_2}\right)} = \frac{25.4}{\sin\left(\frac{180}{45}\right)} = 364.124 \text{ mm}$

4. Velocity

$$v = \frac{pz_1n_1}{60000} = \frac{25.4 \times 27 \times 750}{60000} = 8.57 \text{ m/sec}$$

Power N =
$$\frac{F_0.v}{1000 k_1 k_s}$$
 --- 21.115a (DDHB)
 k_t = Load factor = 1.1 - 1.5
 k_t = Service factor
-- 1.2 for 24 hours operation (Assume 24 hours operation)

= 1.2 for 24 hours operation (Assume 24 hours operation)

Take
$$k_i = 1.3$$

 $\approx 10 = \frac{F_0 \times 8.57}{1000 \times 1.3 \times 1.2}$

 $F_0 = 1820.3 \text{ N}$

6. Allowable pull

$$F_a = \frac{F_u}{n_a}$$
 where n_a = Working factor of safety

From Table 21.75 for $n_1 = 750$ rpm and p=25.4 mm

Select the working factor of safety $n_o = 11.7$ [n_o is not equal to 10.7, printing error in DDHB]

$$= F_2 = \frac{17900}{117} = 1529.914$$

7. Number of strands

$$\bar{t} = \frac{F_0}{F_0} = \frac{1820.3}{1529.914} = 1.189$$

 λ . Number of strands i = 2

8. Check for actual factor of safety

Actual factor of safety
$$\pi_n = \left(\frac{F_u}{F_0 + F_{cs} + F_s}\right)^{i}$$

$$F_0 = \frac{1000 \, \text{N}}{v} = \frac{1000 \times 10}{8.57} = 1166.86 \, \text{N}$$

$$F_c = \frac{wv^2}{g} = \frac{127.5 \times 8.57^2}{9.81} = 954.56 \, \text{N}$$

$$F_s = k_{sg} \, \text{w C}$$
 From Table 21.58 for horizontal drive, $k_{sg} = 6$

$$F_{s} = 6 \times 127.5 \times \frac{35 \times 25.4}{1000} = 680.085 \text{ N}$$

$$\therefore n_{a} = \left(\frac{17900}{1166.86 + 954.56 + 680.085}\right) \times 2 = 12.778$$

Since $n_a > n_o$, the selection of the chain is safe.

9. Length of chain in pitches

$$\begin{split} \mathbf{L}_{\rm p} &= 2\,\mathbf{C}_{\rm p}\cos\alpha + \frac{z_1+z_2}{2} + \alpha \left(\frac{z_2-z_1}{180}\right) & --21.122\,\text{(DDHB)} \\ \alpha &= \sin^{-1}\!\left(\frac{\mathbf{d}_2-\mathbf{d}_1}{2C}\right) & --21.122\,\text{(DDHB)} \\ &= \sin^{-1}\!\left(\frac{364.124-218.79}{2\times35\times25.4}\right) = 4.6886^{\circ} \\ \therefore \ \mathbf{L}_{\rm p} &= 2\times35\cos4.6886 + \left(\frac{27+45}{2}\right) + 4.6886\left(\frac{45-27}{180}\right) \\ &= 106.2346\,\text{pitches} \end{split}$$

The nearest even number of pitches is 106

$$L_p = 106$$
 pitches

11. Correct centre distance

$$L_{p} = 2\frac{C}{p}\cos\alpha + \frac{(z_{2} + z_{1})}{2} + \alpha\frac{(z_{2} - z_{1})}{180}$$

$$106 = 2 \times \frac{C}{25.4}\cos 4.6886 + \left(\frac{27 + 45}{2}\right) + 4.6886\left(\frac{45 - 27}{180}\right)$$

$$\therefore C = 886 \text{ mm}$$

80: - 6:0: -Diameter of the diver pulley

: Radius, + = loomm.

Diameter of the follower pulley ! : Radius, re= 200 mm.

Centre distance of the driver,

speed of driver pulley, N, = 400 pm

Tension in the belt, T.= 1.21cm

T1=1-2×103N:

Coefficient of friction, 11 = 0.25

Angle of contact: -

The angle of contact 0=180 + 20

Sind = x,+ 42 (Crossed balt)

Sind = 100 +200 > Sind = 0.15

, x = sin (0-15) = 8-627°.

: 0 = 180°+2d = 180°+2 (8.627)

0 = 197.254 = A7.254 × 1100

1: 0 = 3.442 rad

.; Length of the Belt: The length of the crossed Belt, L=TT (+,++2) +12x + (+,++2) = 11 (100+200) +2×2000 + (100+200)2

T= 4487. 1418 -: length of the belt , L= 4.987 m.

Power Transmitted: -

Transion on tight side of bilt

To = Tensia on Hack side of belt

wich 2.3/09(12) = 000 1111

2.2 log (T) = 0.25×3.443= 0.861

log (Te) = 0.374, 7, To = 2.366.

 $T_2 = \frac{T_1}{2.366} = \frac{1.2 \times 10^3}{2.366} = 507.785 \text{N}$

Velocity of belt,

V= TId, N1 = TX0.8×400 = 4.189~/8.

Power transmitted by is given by

P= (T1-T2) V = (1.2 ×103 - 507.185) × 4.189

= 2902.202W : P = 2.902 kW

> Design a flat belt dive for the following : data; Dive: Ac motor, operating speed is 1440 rpm and operates for over 10 hours. The equipment driven is a compressor, which runs at 900 rpm and the required power transmission is solow. T= 1121- 148 mm is taken as 4110mm

Pouer P= 20 km. Operating speed, N. 5, 146107pm Compressor Speed, No = 900 pm and the material of the belt is leather Velocity valio = 10 = 1440 = 16 Velocity = TTLN1 = 20 = TTx dx 1440

:. d=865.258 mm From standards, d= 280mm. But, D=1.6d x 280 = 448mm. Drameter of bigger pulley Daysom Actual speed ratio,

D = 450 = 1.607 Conter distance x=2(D+d) x=2(450+280)=1460mm.

From standards "x" is assumed to be 1500 mm

Consider an open belt dive, the bugh of the belt is given by 1= 1 (D+d) + 2x + (D-d)2

= T (450 +280) +(2 x1500) .+

From standards the required belt length : F = Allowin Assume the belt speed, V= som/s Angle of contact is determined as Sind= [0-1] = [450-286] = 0.057

1. d=sin-1 (0.057) = 3.268°. Angle of contact on smaller pulley 0, = 180-2x = 180-2(3-268) = 173-464° Angle of contact on larger pulley Diameter of smaller pulley, d=38mp 02=180+2×= 180+2(3.268)=186.536. The osefficient of fiction for bealth belt

is assured as 0.4 >) 11=0.4. Designed pover P'= Serice Factor (Cr) x

Power rated (P)

For velocity vatio V, R = 161, the Service Factor (cr) is 1.3.

1 pl = 1.3 x 20 = 26km.

The designed power is also given as P1= (bt)(0-PV)(1- 1- 12) V

For leather belt, or = ampa = en/mme and; P = 1000 kg/m3.

: p1=26x10,3=(bt)(2x106-103x22) 1-30.4x3.10

.. pf =1114 mmg. from standards t = 7mm and b = 159.14 = 160mm .. A bother helf of thickons t= man, width b=160 mm & belt-length L= 4110 mm satisfies thedaigh

Maximum tension in the belt, . Trax = Td+Tc = obt 2201-439+0-527b=2.5×106xbx16-3×9.5×10-3. P = d A . Jammi Width of the belt ib = 94.79 mm.

-) It is required to select a flat open belt drive for a compressor running at 720,pm, which is diven by a 25kW, 1440 rpm motor space is available for a center distance of 3m.

G.D: - Pouce P=25kin · Operating special (No. = 1460rpm. Compressor speed No = 7201pm. Centre distance X=3m=3000mm. Assume, the belt speed iv= 20m/s and the material of the bell- is kathan

Velocity ratio = No = 1440 = Velocity , V= TLANI GO × 1000

From standards, d = 280mm. Diameter of smaller pulley desson

But D= 1.62 100 = 448 mm

Diameter of bigger pulley, D=450m Consider an open belt dire, then

the length of belt is given by. r= 1 (D+q) + 2x+ (D-q)

= 1 (450 +280) + (2 × 3000)+

L= 7149.08mm]

From standards, the required belt length is taken as 7150mm. Angle of contact is determined as, Sind = (-2x) = (450-280) = 0.028 a d = sin-1 (0,028) = 1.60.

Angle of contact on smaller pulley, 0 = 180° - 2d= 180° - 2(1.6°) = 176.8°. Angle of contact on larger pulley,

· A2 = 180+2 × = 1800+2(1.60) = 183-20.

The coefficient of friction for leather belt is assumed as only.

Designed, power, P'= service factor (G) * Power rated (P)

: P'= 1,3 x 25'= 32.5kW.

[Assiming, the service factor (Cr) 15 1.3]

The designed power is also given as

P'= (bt) (5-PU?) [1-en] V.

For leather belt,

0= 3 mpa = 3 mm2 and p=1000 kg/m2

A flat belt is required to transmit 30 kW from a pulley of 1.5m effective diameter running at 300 pm. The angle of contact is spread over 11/24 of the circumference. The coefficient of fiction between the belt and pulley surface is 0.3. Determine taking centrologal tension belt and pulley surface is 0.3. Determine taking centrologal tension into account, width of the belt required. It is given that the belt into account, width of the belt required. It is given that the belt thickness is 9.5mm, density of belt material is 100kg/m3 and the permissible working stess is 9.5mpa.

فاحك

Power, P=30kw Speed; N=300 rpm Effective diameter of pulley d=1.5m Angle of contact, . 0= 11 x Grambere = 111 × 360° = 165° = 165 × 150 = 3.88 rad. Coefficient of friction between belt and pully surface, 4=0.3 Thickness of the belt, t=9.5mm Density of the belt, P= 100 kg/m3 Permissible stras, - = 2.5 mpa Velocity of belt, V= TdN = 11 × 1.5 × 300 = 23.562 m/s Area of cross section of the belt A=b*t = b × 9.5 =9.5 x bym2=9.56 ×10-6m2 Mass of the belt per instellength m= Area x length x Density =9.56 × 10-6 × 1×100

m=9.5bx 10-4/cg/m-Centifyed tension, : W.Lc=mV2 =9.56 × 10-4 × (93.562)2 Tc=0.5276N. Power, P= (T,-T2)V Ti-Tension in tight side in N. Te-Tension in slack side in N. 30×103= (T,-T2) ×23.56. $T_1 - T_2 = \frac{30 \times 10^3}{92.56} = 1273.345N - 1000$ But, T1 = e40 = e0.3 x 2.88. Ti = 2.372 >) Ti = 2.372T2 -> @ By substituting of 10 in of 10 2.372T2-T2=1273.345 1.372T2 = 1273.34 > T2=928.094N From eg (2) T, = 2.372 x28.094 = 2201.439N 1. T, = 2201.439N &Te = 928.094N.

.: P1=325 x103. :. 32.5 × 103 = (bt) (2 × 106 - 1000 × 202) [1- (0.4 × 1.6) × 20. 32.5 × 103 = (bt) (1.6 × 106) (1-653) ×20. PF = 130 5 inng.

From the standards, t=8mm and b=174mm. Therefore, na letter belt of thickness, t=8mm, width, b=174mm and belt length. L=7,150mm satisfies the design anditions.

A belt 100mm x 10mm is transmitting power at 15m/s. The angle of contact on the driver (smaller) pulley is 1960. If the permissible stress for the belt material is an /mm2: determine the power that an be transmitted at this speed. Take the density of leather as 1000 lg/m3, and coefficient of fretion as 0.25. Calculate the imaximum power that can be transmitted by this belt, and the speed at which this can be transmitted.

81 - GD: -Width of the belt, b= 100mm = 0.1m Thickness of the belt, t=10mm=0.01m Contribugal tension is given by, Velocity of the bell. V= 15m/g.

Angle of contact, 0 = 156°. Permissible stess for the belt material 0-=2N/mm2.

Density of the belt material P=1000lg/m3 Coefficient of friction 1 = 0.25

Total Ension on the tight side of thebell

T=Tti= oxbxt

TE = 2 × 100 × 10 = 2000 10.

: T= TE = 2000 N.

Mass of the belt par meter length is given by

I'm = Area of cross-section x length is Doneity m=0.1x0.01x1x1000 = 11g/m. Tc=mv2= 1x152=225N. Considering the certifical tension, tension on the tight side of the belt :. T = T-Tc =2000-225=1775N , Let, To be tension on the slack side Tr = e MB Ti = e 0.25 × 156 × TT 180 = 1.975.

 $T_{Q} = \frac{\gamma_{1}}{1.975} = \frac{1775}{1.975}$:. T2 = 898.734N

Power transmitted by the belt, P= (T,-T2) V P=(1775-898-734)×15= 13143-99 wate= 13.144kw. 1. P = 13. 144kw Maximum pover transmission occurs when the speed of the belt is, Nmax = \ TEI = \ \frac{3x1}{2000} = 25.85m/s. Maximum antingel tension Temax = m.V max = TEI Tem = 2000 = 666.6670. Maximum power transmitted by the belt is given by, Prox = (It = MN/max) [1- end Vmax. Pmax = (2000 - 666.669) [1 - 1.975] x25-82 = 16995, 439W. @r) 16,995kW

In an open flat belt drive, the pulley diameters are sommand 450 mm and the corresponding and of lop are 160° and 200°. The smaller pulley runs at 240 rpm. The coefficient of friction between the pulley and belt is 10.3. It is found that the belt is on the point of slipping when 5kw. power is tansmitted. To inchase the power transmitted, two atternatives are suggested he, i, Increasing the initial tension by 15% and il, Increasing the coefficient of friction by 15%. which of the above two methods would be more effectives. SA: G.D= Dia of pully i d, = 300 mm = 0.3m Dia of pulley 2 de = 450mm = 0.45m Angle of lap 0,=160°.
Angle of lap 02=200°. Speed of pulley-1, N1=240 mpin Coefficient of friction, 420.3 Power P=5kW=5000 watter For smaller pulley, ___ | " / 1 ... Mg =0./3×160× 180 /=0-838; For larger Pulley, "T.". MB2=0-3 × 200 × 170 = 1:047 since, 40, of smaller pully is less than 40 2 of bigger pully, He smaller pulley is the weaken pulley. "" Therefore the design is based on smaller pulley, " 1 . 1) Peripheral relocity of the belt !! $V = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.3 \times 240}{60} = 3.77 \text{m/s}.$ To = 1925.986N. IFT, and Te are the tensions on

tight side and slack side,

respectively, bun

Power transmitted by the belt, P=(T,-Te)v 5000 = (TI-T2) ×3.77 . >> T1-T2= 1326.26~ → 0 Te = E HO 3/ TE = 20-838 9-311 . T1 = 2.3 11T2 → 3 sub eg@ in eq@ 12:347/2-12 = 1326.26 · 72'= 1326.26 = 1011.6400. Ti =2.311Te= 2.311x 1011.64 X 7, = 2337.9N 1) Pocer Transmitted when Initial Tension is Ironased by 1511:1 Initial tension in Rebell, To = Title To = 2337-9,4,1011.64 = 1674,770. : Increased tension, To = 1674.77 ×1.15. Let Ti and Tet be the new tension on the tight side and sladeside of the belt respectively. Scanned by CamScanner

1. To = Ti+TE T, +. Te = 2701 = 2 x 1925-986 TI+T21 = 3851.972N -33 But ratio of tempions Ti is content 1-e Ti = 2.211 T' = 2.311 TE' - 1 Sub eq (1) in eq (1) 2.311 T2 + T2 = 385.1.972. T21 = 116'3.386N Ti=8.31178 = 2.311×1163.386 T. = 2688.585N. : Power transmitted. $P_{i} = \left(T_{i}^{i} + T_{2}^{i}\right) V_{i}^{i+1}$ P1 = (8688.585 -1163.386) x3.77 P1 = 5750W (cr) 5175kw) ii) Pover bunamitted when the coefficient of Priction is Increased by 15% -Increased coefficient of faction M = Mx1:15 = 0-3×45 = 0-345. Let, I' and Te" be the new tensions on the tight si de and dack side of the belt respectively Then 1 Till = = "Lia, Till = 0.345 × 160 × 180 = 2.621 T"= 2.621 T2" -> 5 where initial tension is anstant.

 $T_{0} = \frac{T_{1}^{11} + T_{2}^{11}}{2}$ $T_{1}^{11} + T_{2}^{11} = 2T_{0} = 2 \times 1674.77 = 3349.54$ $T_{1}^{11} + T_{2}^{11} = 3349.54$ 50 lying eq 6 and 6. $2.621 T_{2}^{11} + T_{2}^{11} = 3349.54.$ $T_{2}^{11} = 925.032N$ $T_{1}^{11} = 2.621 T_{2}^{11} = 2.621 \times 925.032$ $T_{1}^{11} = 2424.508N$

Power transmitted. P2=(T1"-T2")V

P2=(2424.508-925.032) x3.77

:P2=5653.025w 6v). 5.653kw.

Bing the power transmitted by.

in area sing initial tension is more.

transfer for increasing power initial

tension should be increased.

A pump is divien by an electric motor through an open type flat belt dise. Determine the belt specifications for the following data. Motor pulley diameter (dg) = 300mm, Pump pulley diameter (dg) = 600mm, Coefficient of friction (Mg) for indu. Pulley = 0.25, Coefficient of friction (Mg) for indu. Pulley = 0.25.

Center distance between the pulleys = 1000mm;

Rotational speed of the instar = 1440pm.

Pover transmission = 20 km;

Density of belt material (P) = 1000 lg/m³.

Motor Pulley diameter, de = 300mm Pump pulley dramater, de = 600 mm Gefficient of friction for motor pulley,

coefficient of friction for pump pulley, ML=0.2.

Centre distance between the pulleys,

speed of motor, N, = 1440 ypm

Power, P=20KN=20x103W.

Density of belt material, D=1000/g/m3.

Permissible stres, o = 2 mpa = 2 n/m [: TI = 1.99172] -> 2

Thickness of belt, t=5mm = 0.005m

Velocity of belt,

V = Tids. N1 = Tx300 x 1440

1. V=22.619m/s.

Area of cross-section of belt. A=bxt =b =5mm2.

Mass of bell permetre length, ma Acaxlength x Density.

= (bx E) x l x)

=6×10-3×0.005) ×1×1000

m = 0.005 b kg/m.

centrifigal tension Tc=mv2 =0.05bx (22.619)2

Te =2.5586 N.

Pover P= (T,-T2) V

20×103 = (T,-T2) ×22.619

T1-T2= 884.212 -> 0

But Ti = EMB.

sina largerpully governs the design,

d=sin-1 \ d1-d3 = sin-1 \ \ \frac{600-300}{2x1000}

.. d = Sim (0.15) = 8.627.

OL = 180+2d= 180+2(8.627)

= 197,254° = 3.443 mad.

05 = 180 - 22 = 180 - 2 (8.627)

=162.746 = 2. Burad.

Tr = (ML. DL) = e (0.2 x3.443) = 1.99)

sub eq 1) in eq 0

(1-991T2-T2) = 884.212.

= T2 = 892,24NN

TI = 1.991 Te = 1.991 x 892.242.

TI = 1776.454N

Maximum tension in the belt.

Tmax = Ti+Tc=obt.

2×10 x 6x0.005 = 1776.454+2.5586

9997-442b= 1776.454

1 b=0.178m

: Width b= 178 mm

G.D: - Power P = 20/cm = 20×103 W.

Diameter of pully, de = 900 mm = 0, 900m Angle of, contact, 0 = 180-22

Centre distance, &= 1000mm =

Coefficient of Riction, 4=0.2"

Allowable tension, T, = 1000 N.

Density of belt, 1 = 1000/g/m3.

Top width of belt, a = 40mm = 0.040m

Bottom width of belt, b=20mm=0.020m

Depth of belt h= 25 mm = 0.025m

And
$$sind = \frac{d_2 - d_1}{3x}$$

Diameter of shears, d, = 350mm=0.250 , d=Sin-1 (0.325) = 18.966.

0 = 180 - 2 × 18,966. = 142.068.

10 = 142.0×180 = 2.48 radians

Chrone angle, ap = 40° (given)

y β = 30°

Using the relation,

2.3 log (Ti) = LO Cosec B.

2.3 log [72] = 0.2 x 2.48 x Casec(20°)

But for maximum power,

· Velocity, Vmax = V. Tmax

and Trax = TitTe

Tmax= 1000 + Tmax

[: for maximum power, To = max

Tmax = 1000

material and their the same cross-sectional and. In each case, The v-belt pulley groove angle is 600. The belts are of the same fiction for the flot belt and v-belt is same and is equal to 0.35. robites at the same speed as the growed pulley. The coefficient of pulley is some as the pitch diameter of the groved pulley. The pulley changing over from a flet belt to a vebelt dine. The diameter of Alat - Determine the percentage increase in power capacity made possible in is when it belts required are ?. . 2 200. 2 = 683.1PPP = N Member of belles required no Row turismiled per tell bellinging botol Power Emished per bell, (314; 429-0001) = V× (6.T-17) = 9 Jan 20. El = xom/ K. elm 120. El = DEE. OFT V = 10021 / E xom/ MB148P-8= M. K 2001× SIPEX EDIX 021-0= M "I.Q x J x A = M K Jand x Abrad x Dand = in Jod & sport .. Langh of the belt,

Langh of the belt,

Light of the belt,

Light of the belt,

Light of the belt,

Light of the belt,

Light of xLx A.

Light of the belt,

Light of xLx A. langh of he belt, 12 = JUNOX = 1500 = 5001 × MIZ = 5001 . Confitted tension in belt for movement Laguadiano . NOOSI = xom [K 0001 = [= 1 - 1 | xom]

De angle & lap 15, 160.

Angle of V-belt grove, 2B = 60° Angle of lap, 0 = 160 = 160 × TT = 9.793 rad. Coefficient of friction for flat and V-belt M=0.35. Flat Belt Diver - be the tensions for flat beltidive. The relation between tight and slack side is given as To = e 40 = e 0.35 x 2.793 = e 0.978 = 2.659 > - T2 = 2.659 Power transmitted P= (TI-TE) Vi P= (T1 - T1) V ... >) . P = 0.624 V1T1 Let Prand Tenbe the tensions for the V-belt dive. Power bandwitted P = (Ti-Te') V2 = P= [Ti-Ti-Ve. Since the material and cross sectional area of both the bells are same. Also, the pulley rotates at same speed as grooved pully. Hence ! Perantage increase I'm power capacity, = P-P: x100 = 0.375 × 100

-) Design a v-belt dive to connect a soku, 1440pm instor to a compressor running at 480 mp.m for 15 hours per day. The approximate center distunce is 1.2m. Sol: G.D: - Power, P= 20/20 speed of pulley -1, N1 = 1440rpm Speed of pulley - 2, No = 480 pm. centre distance, &= 1/2m= 12001 Service = 15 hours perday. Correction Factors-In this application, a motor of acked is diving a compressor for 15 hours per day. The correction factor according to service (15hous/day) is 14 Design powers ... The design power is given as, P'= Fax P= 1.1x(20) = 22/2W. Type of Cross-Section: -According to designed power and speed of motor, the cross-sectional . | V-belt is B. Pitch Diameter of Pullays: from the standards of coss-section the minimum pitch diameter to B-section belt is soomin. :. Diameter of smaller pulley d= 200mm, Velocity vatio, V.R= (440 = 3. Diameter of large pulley, D=3xd=3x200=600mm.

Hength of Bell-diver- $\Gamma = 3x + \frac{3}{4(D+q)} + \frac{4x}{(D-q)_5}$ = (2x1200) + 17 (200+600) + (600-200)2 L ~ 3690 mm Desirable pitch length: -The desirable pitch length for B-section belt is 4000mm or 3600mm. Thus, the pitch leight of belt is assured as Actual center distance, Γ= 3x + 3 + (D-q) ; $3600 = 2x + \frac{\pi(600 + 200)}{2} + \frac{(600 - 200)^2}{4x^2}$: x=1154-256mm. The actual center distance, x=1154.356m Number of Bellisz - " The number of belts is igner by Fc = Conection factor for pitch light. From standards, For Breeding and 3600mm pitchleyth Fc = 1.10 Fd = Correction factor for arc of antact,

A V-belt drive consisting of two belts operates on two pulleys of orm and 0.45m pitch diameters: the smaller pulley being the driver pulley. The antie distance of the driver is 0.75m and he groove angle of the pully is 20. The cross sectional area of each belt is storm? and coefficient of friction is 0.25. The density of the belt material is 1100 leg/s.

The permissible stress for the belt material is 50 /m. Determine the power that can be transmitted when the driver pulley is notating at 1400 mm.

Number of belts h=2

Dianeter of diver pulsy d, =0.3m

Dianeter of follower pulsy de=0.45m

Centre diaterra of the diver, x=0.75m

Angle of grove, 2B=30.

Cross-sectional area of belt A=510mm

Coefficient of friction, 1=0.25

Density of moterial, P=1100kg/m³.

Permissible stress, o=50/mm².

Speed of diver pulley N, =1400 rpm.

The mass of belt per metre length,

m= Area x length x Density

V= Tx 0.3x 1400 = 21.991m/s.

Centrifugal tension,

Tc = mv = 0.561 x (21.991)2

Tc = 271.3020.

Maximum Tension Track = 5 x A

Track = 5 x 510 = 25500.

The tersion in tight side of the bet,

T = Track - Tc

= 2550 - 271.302 = 2278.6980

Velocity of belt, V= TId, NI

Angle of Contact on the smaller pulley, 0=180=20. But, sind = de-di = 0:45-0.3 =0:11 > | 2=5.7290 : :. A = 180° - 2 (5.729°) = 168.522° 3 (0'=168:522 × 11 = 2.94 | rad 2.3 log [7] = MB Cosec B: = 0.25 x 2:941 x cosec (15) $\frac{T_{1}}{T_{2}} = 3.438 \Rightarrow T_{2} = \frac{T_{1}^{(1)}}{3.438} = \frac{2278.698}{3.438} = \frac{(662.798N)}{3.438}$ Te = 662.798N. Power transmitted, P=(T, T2) V×n = (2278.698-662.798) ×21.991 x2 = 107050 1. 12. P=11.071 Km -) It is required to select a V belt dive to compect a 15kW, 2880pm normal tarque Ac motor to a centrifiged pump, running at. approximately 2400 pm, for a service of 18 hours perday. The center distance should be approximately yourm. Hesing that the pitch diameter of the dining pulley is 125 mm. , Poum p.= 15kw 1) Connection Factor Fa @ Design power P'= 18kW Speed of pulley-1 N/ = 2800 ym. @ Type of Goss section Speed of pulley - 2, 102 = 2400 ypm self is Btype @ Pitch Diameter of pulley Centre distance x=400 mm. service = 18 hours perday. 3 length of Beltiding Same model problem is there is L= 1232.36. @ Desirable pitch length material Collow the procedure 1 1 1 1 = 388:85 mm to get the solutions,

A rope drive is to transmit, 250kw how a pulley of 1.2m diameter, wiring at a speed of 200 pm. The angle of lap may be taken as Tradians. The groope half angle is 23.50. The ropes to be used are somm in diameter. The mass of the rope is 1-3 kg per meter length and each tope has a maximum pull of 9.2 km. the coefficient of friction between rope and pulley is 0.3. Determine the number of ropes required If the overhang of the pulley is 0.5 misuggest suitable size! for the pulley short if it is i made of steel with a shear stress of 40 mpor. Shorty it it is in it is in it is a special power towarited per rope of Diametre of pulley, d=1.2 = 1200mm = (T.-T2) VIII Speed N=300 pm,
[1738.1-147.3] × 18.85

[1738.1-147.3] × 18.85

[1738.1-147.3] × 18.85 · Dumber of ropes required Rope diameter, de = 50mm 1 Maga of tope; m= 1.3 kg/m; Maximum tension, T= 2.2 kN-Coefficient of faction 4=0:3.

Shear stees To Horya or N/mm2. Velocity of the belt.

V= TAN: TX1.2'x200 =18.85~0/31. 1738. (+147:3+2×461:8) x0-5×9 Te=1002=1.7(18.85)=1491;910. a Tension in the tright sidery 17e= 12+M2 Tree Macrec Bie 6 13 xIII x posec 22.50 (1957.75)2+ (12608)2 $\frac{T_1}{2} = \frac{11-8}{11/8} \Rightarrow \frac{T_1}{11/8} \stackrel{!}{=} T_2$ Te = 14.938 x1040 + mm) T2 = 1,7381 > T2 = 147.3N) Bull Tell Tox TXD30 and 141938×106= T. × 40×03 Number of ropes required : D= 125mm

3) Suggest the suitable size of 6x 19 hoiding steel wire rope for an inclined mine shaft of 1000m length and inclination of the rails 60 with the how zarital. The weight of the boaded skip is 100 km. The maximum acceleration is limited to 1.5 m/s2. The diameter of the dummon, which the rope is being award may be taken as sotimes the diameter of the 1 rope The Carthation is 2000/cm of everyth mormal to the incline and friction of the tope on the guide roller, is 50 N/EN of vieight mountain to the inding. Assume a factor of safety of 5. The following properties of The resignit of the rope per meter = 0.334 de 20. Breaking load = 500den, Wire diameter = 0.063dmm. Area of wies in tope = 0.28 d2 mm2, Equivalent destic modulis = 82 los/13 Working load for the wine rope Solit GiDine .. " " " " length of hoising steel wire tope, he township it is and Breaking load Angle of indication of rails with the His Wa = Good 1 1= lood 2N. honzontal, 0 = 60° 11115 Weight of the tope The cought of the loaded slop, W= Weight of the rope per meter x F=100 ku = 100 x103 N.

Maximum acceleration, a=1.5 m let. length of the tope. W=0.03342×1000=33.4220. Drain diameter, D=80 x Dia of rope=80d Weight of the tope along the indination Cor faction = 200./100 of wight. Mij= M'00330%= 33.495x c00300 Rope faction = 500/100 of weight. 1-1, WE = 28-9 25420; Tension in the rope due to straight Factor of salety Fos=5. Weight, of the rope per meter = 0.0234d T= Face 300 = 100 × 103 × Cas 300 Broking hoad = 500d20 T=186602,5410 Fretantoid in the rope due to a wight, Wire diameter dw = 0.062 d mm.

Area of civis in rope, A = 0.38dimin

Scanned by CamScanner

Fiction boad in the tope duto, up fiction,

III ME = 30 x 100 cos 60°.

WR = 50 x 33. 4 de x 6066.

WA = 10000!

Equivalent Bending load We = 100xA.

where ob = Bending stress in rope = D. Equivalent Rending load, We = Edw x A . fraid light is There had in the tope, WI = [-3-] 11/(. = 1) [100 × 102 + 32-4d2] × 1.5. 4) por Jam WI = (15290, 586 +5, 107 d2) N. Total load on the rope is the submitted the above loads, ire Wit = Wx + T + Wp + Wp + Wc + WI. = 28.925 22 + 86602.54+(1000+0.835 24 24.539 2+ 15290.520+ Equation World Www. 3 Www. Wr. Word 25 3 dr. 50.346 mm Man 51mm (skindard value) 17 - M The dignoter not 6x 19 horstony steel chope is 51 mm. A CI flat pulley turnits solow at a speed of storpm. The pully overhangs the nearest bearing by acoum. Hasining the ratio of the belt tempors as a. Determine i, Shaft diameter ii) Aylly diameter and Cross-section of eight arms Assuming number of arms, n=8. Consider contribujel stess or hoop stess of = DV2.

where of = Permissible troopstess (Assure it as 25mpa).

P = Density of C.I. = Tacolig /m3 1 ... : 25×16 = 7200× V2 "> V= 58.92 m |sec i) Shaft Diameter; A. ... P=(T-T2) VIII) (30x1000) = (T,-T2) x 58.93. TI-TO' = BB9.40 - 1 Ass 11/1-2= 2 The state ring of the stand there Sub eq @ m q 0 1 1 mil 2T2 15T2 = 339 4 - > T2 = 339.40 Also Ti = 181 x 329- 4= ,678,810. · (W. T. T. 239. 4 x 2 = 259140-m. Also it Te = THE XT washer them to what his right of 395.8 × 1000 = 16 13 × 45 (Assure T/= 45mps) 2= 25.5 mm 20.36 mm 11 (1 .1 .1 . it 1). 12. ii) Pulley Didnister and Cross section of light. Arms: ... 1 Pulley dianeter is dready calculated in short dianeter in a D=2m. 1 311 . 11.01 11.1 1. . 1.11

Maximum bending monet to be resisted by each arm,

Maximum bending monet to be resisted by each arms to be effective at Mmax = 1239.4 = 184.85 N-m = 84.85 × 103 (: A8kmed of = 25) 1 3 1 84.85 x103 > 1 a = 41mm An overhung cost iron pulley transmits 75kw at/Boorpyn. The dianter of the pulley is 500mm and angle of Haptist 180. Find, in Diameter of the pulley The density of Contis , 7200 to line.

in Widh of the belt, if the coefficient of faction between the belt and

the pulley is possed. Ill, Djameter of the shaft, if the distance of the pulley center line from the nearest bearing of Booking Dimension of the boy for seawing the pulley on to the short. taken as elliptical, the major ours twice the minor gods length. The following stress may be talan for design purposes! Shoft fant ley: Jensian - 80 mpa, Shear - 50 mpa Pulley rains & Tension - 15 mpa, tadio and the first of the service of the servic B) - G.D. - Power 1P=17.5kw= 7500 W. Speed N=300rpm. Lap angle 10 = 180 = Trad. Density of CII /2 = 7200 kg/m3, coefficient of faction His 0.25 Number of and son = 6, Distance from hearst bearing, 1=300 mm.

Asses of the best of Density Density of the State of the sent of t Sina V is grade then low less the field of t Edix 2.7 × 00 1 1 1 1 200 = 1 1 1 200 0 1 20 0 0 1 200 The office = 101/100 and 11 is a death of the one = 3 pm.

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Let L= length of the key. Let k= length or the reg.

Considering shearing of the key, value should let bi = Minor axis, ai = Major axis=26/20 27 1/8" mediand Let b, = Minor axis, assuring that help didle number of ours are

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Section modules, 2 = 33873 = 79.57 N=m. 126 N=mm

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Section modules, 2 = Tel 3 | Tel x 1/22 | prix (26) 2 minuted Bending stess (b) 110 b = 12 b = 1 (1) 11 x b; = 79.57 x 1p3 (1.1) [27] [27] [57] [57] [57] [57] [57] b,3=13508.22 (m) (e) (1) (m) (m) -1 1 - (1.1.) pii = 3/1320.8°5°5 = 33.81mm, > 1/101 = 31mmily : Mojor axis, a, = 24x2 = 48mm. > [a,= 48mm]

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Title line rebuilty of the line of the problem is x is is) Design a chain drive to actuate as sompressor from a 15kwelectic motor running at 1000 pm. The compressor speed being 350pm. The minimum anter distance is 500 mm. He compiesor opentes 16 houspardy. The chain tension may be adjusted by shifting themator ansliders. CAH 881 = CP 1:5 x1x1.25 Sol G.D: - Rated power = 15km / Speed of clectric water, 10, = 1000 mm " Design power = 15 x 1875. JE - Speed of Comprison , No = 350 ppl . I in the distance tak a pooring. From table, For chain NO.12, From table of relicity ratio, UP=3 of help pt 1910 shows land on Number of teeth on small sproduct. Max wild diameter of = 12.07 mm. [T:=25] (: For roller chain) Number of teeth on targe spirckets Biolog Local, WB = Sale N = 59 × 103 N. T2 = T1x N1 = 25 x 1000 Pitch circle diameter of shall sprocket, 12 = 72 9 . JIT-11. Design power = Roted power x + 91 = 1800 = 18.02 cosec (180) semis factor (kg) Bitch einde draneter of large spocket, Seria Paction) ks = le/x kpx k2 de = 9 casec (180) = 19.05 Greec (180) k, = 1.5, for variable load with heavy shade 1= H36 1435 mil la = 1 dop "horication" = 4.1 de = 0.436m ks = 1.95, 16 hous beholding -Pitch line relacity of small sprocket, Scanned by CamScanner

d, = Posse (180) = 19.05 cased (180) = 0.152 m Pitch circle diameter of large spinclet de = proced (180) do o 1960 50 50 (180) 1 = p. 636 m > 1 di=0.436 m Pitch line relocity of small sprackety of 17.1 = 7.958m/s,

V1 = \frac{\pid_1 N_1}{60} = \frac{\pi \times 0.152 \times 1060}{-60} = 7.958m/s, ، دا مام طفعد له مولسوا مع والمعالم المعالم ا Manual on Chair Man Rated Bover 15 184kN: ford on chair bilitabline reports for some winds all' (2.1): - Kated power & 1884m | MH881 = M (2) | 100 x 1x 1.36 From table, Factor of solety for holler chain the with which is less For table, Factor of solety for tollet channels workprosent in the solety of the state of the solety of the state of the solety of the state of the solety o 1001 | 1001 | 1001 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 1005 | 10 سر 6 و من المناس المنا



DEPARTMENT OF MECHANICAL ENGINEERING

MID & ASSIGNMENT EXAMINATION QUESTION PAPERS WITH SCHEME AND SOLUTIONS

NARASARAOPET ENGINEERING COLLEGE (AUTONOMOUS): NARASARAOPET

DEPARTMENT OF MECHANICAL ENGINEERING III B.TECH II - SEMESTER ASSIGNMENT TEST – I, JANUARY - 2023

SUBJECT: DESIGN OF MACHINE ELEMENTS-II	DATE: 03-01-2023
DURATION: 30 MIN	MAX MARKS: 10

Q	£	Course	Knowledge	Marks
N		Outcome		Wiaiks
		(CO)	Bloom's	
1	Solve the following problem.		Taxonomy	
	Design a journal bearing for a centrifugal pump running at 1440 r.p.m. The diameter of the journal is 100 mm and load on each bearing is 20 kN. The factor ZN/p may be taken as 28 for centrifugal pump bearings. The bearing is running at 75°C temperature and the atmosphere temperature is 30°C. The energy dissipation coefficient is 875 W/m 2 /°C. Take diametral clearance as 0.1 mm.	1	Apply (K3)	10
2	Solve the following problem. A journal bearing of 50mm diameter and 80mm long, has a bearing pressure of 6 N/mm 2, the speed of the journal is 1000 rpm. The ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil, whose absolute viscosity of 75 0 C may be taken as 0.015 kg/m-s. The room temperature is 25 0 C, Determine (i) the amount of artificial cooling required and (ii) the mass of the coolant oil required, if the difference between the outlet and inlet temperature of the oil is 10 0 C. the specific heat of the oil is 1900 J/kg/ 0 C and heat dissipation coefficient is 500 W/m 2 / 0 C	1	Apply (K3)	10
3	Distinguish between full and partial journal bearings.	1	Analyze (K4)	5
4	Differentiate between hydrostatic and hydrodynamic bearing providing their neat sketch.	1	Analyze (K4)	5
5	Explain the design procedure for journal bearing.	1	Evaluate (K5)	5

NARASARAOPET ENGINEERING COLLEGE (AUTONOMOUS): NARASARAOPET

DEPARTMENT OF MECHANICAL ENGINEERING III B.TECH II - SEMESTER ASSIGNMENT TEST – II, FEBRUARY - 2023

SUBJECT: DESIGN OF MACHINE ELEMENTS-II	DATE: 27-02-2023
DURATION: 30 MIN	MAX MARKS: 10

Q. No	Questions	Course Outcome (CO)	Knowledge Level as Per Bloom's Taxonomy	Marks
	Solve the following problem. Following data is given for the piston of a four-stroke diesel engine: Cylinder bore != 250 mm Material of piston rings = Grey cast iron Allowable tensile stress = 100 N/mm2 Allowable radial pressure on cylinder wall = 0.03 MPa Thickness of piston head = 42 mm Number of piston rings = 4 Calculate: (i) radial width of piston rings; (ii) axial thickness of piston rings;(iii) gap between the free ends of piston ring before assembly; (iv) gap between the free ends of piston ring after assembly;(v) width of top land;(vi) width of ring grooves;(vii) thickness of piston barrel; and (viii) thickness of barrel at open end.	3	Apply (K3)	10
2	Solve the following problem. Design a cast iron piston for a four stroke IC engine for the following specifications, Cylinder bore=120 mm,Stroke length=150 mm,Maximum gas pressure=5 MPa, Brake mean effective pressure=0.7MPa,Fuel consumption=0.25 Kg/KW/hr,Speed=2400 rpm Assume any other data necessary for the design	3	Apply (K3)	10
3	Solve the following problem Following data is given for a four-stroke diesel engine: Cylinder bore = 250 mm Length of stroke = 300 mm Speed = 600 rpm, Higher calorific value of fuel 44 000 kJ/kg,Indicated mean effective pressure = 0.6 MPa Mechanical efficiency = 80% Maximum gas pressure = 4 MPa Fuel consumption = 0.25 kg per BP per hr (i) Calculate the thickness of piston head by	3	Apply (K3)	10

	F			
	strength consideration. (ii) Calculate the			
	thickness of piston head by thermal			
	consideration. (iii) Which criterion decides the			
	thickness of piston head?			
1	(iv) State whether the ribs are required?			
	(v) If so, calculate the number and thickness of			
	piston ribs. (vi) State whether a cup is required			
	in the top of piston head?			
	(vii) If so, calculate the radius of the cup			
4	Derive an expression for radius of neutral axis	4	1 (TT 4)	10
	for circular and rectangular section	4	Analyze (K4)	10
5	a)List the assumptions made in the derivation	4	A 1 (TZ 4)	
	of stresses in a curved bar	4	Analyze (K4)	5
	b)Derive an expression for radius of neutral	neutral		_
	axis for T section	4	Evaluating(K5)	5

NARASARAOPET ENGINEERING COLLEGE: NARASARAOPET DEPARTMENT OF MECHANICAL ENGINEERING III B. TECH II-SEMESTER MID-II EXAMINATION

SUBJECT: DESIGN OF MACHINE ELEMENTS-II

(DESIGN DATA BOOK IS ALLOWED)

DURATION: 90 min

MAX MARKS: 25

			Answer all question	ns .	, , , , , , , , , , , , , , , , , , ,	*3
Q.no		Questions	7 10 1	CO	Knowledge Level	Marks
1.	Solve the following Following data is gi	ven for a four-s	troke diesel engine:	, , ,		***
	Cylinder bore = 250 Speed = 600 rpm, H kJ/kg,Indicated mea	igher calorific v	alue of fuel 44 000	-		H 5 Tr
	Mechanical efficience MPa Fuel consumption	y = 80%.Maximition = 0.25 kg	um gas pressure = 4 per BP per hr (i)	3	Applying (K3)	5
	Calculate the thick consideration. (ii) Caby thermal consideration	lculate the thick	head by strength; ness of piston head) * `	÷ 4.	14, " "
	Solve the following p			14		2,
2.	A crane hook has a tin Fig.2. Find the max	rapezoidal sectio	on at A-A as shown points, P and Q.	ţ*	* * *	1
	^1 <u> 0 </u>		· ;	4,	Applying (K3)	10
	116' 125 1	T sold 20 L	\$, a	27 7	i i
		* 12	* * * * * * * * * * * * * * * * * * *	, graq ".	\$ # \$ \$	
3.	Solve the following pro-	the diameter of	of the driver and	ž ř	3 . t	
	follower pulleys are The center distance of rotates at 400rpm. Find and both the pulleys a .What is he power permissible tension in friction between belt at	the driver is 2md the angle of count the length of capacity of the the belt is 1.2kN	The driver pulley ntact between belt f the belt required are driver, if the and coefficient of	5	Applying (K3)	10

Question-1

- 1) Given data -> 1 malk
- Thickness of piton head by Strength consideration -) a marks Thickness of piston head by Thomas consideration a marks
- a) Given data -> 1 mack.

Area of section - 4 marks
Bending moment - 3 marks
Bending stress - 2 marks

3) Orven data) 1 mark

Length of the Belt - 3 marks

Angle of contact - 3 marks

power capacity - 3 marks

NARASARAOPET ENGINEERING COLLEGE: NARASARAOPET DEPARTMENT OF MECHANICAL ENGINEERING III B. TECH II-SEMESTER MID-I EXAMINATION

SUBJECT: DESIGN OF MACHINE ELEMENTS-II (DESIGN DATA BOOK IS ALLOWED)	 -	•	DATE:31/01/2023
DURATION: 90 min			MAX MARKS: 25

Answer all questions

	Answer an questions						
Q.no	Questions	CO	Knowledge Level	Marks			
1.	Solve the following problem Design a journal bearing for a centrifugal pump from the following data: Load on the journal = 20 000 N; Speed of the journal = 900 r.p.m.; Type of oil is SAE 10, for which the absolute viscosity at 55°C = 0.017 kg/m-s; Ambient temperature of oil = 15.5°C; Maximum bearing pressure for the pump = 1.5 N/mm2.						
			Applying (K3)	10			
	Calculate also mass of the lubricating oil required for artificial cooling, if rise of temperature of oil be limited to 10°C. Heat dissipation coefficient = 1232 W/m2/°C	İ	· · · · · · · · · · · · · · · · · · ·				
2.	Solve the following problem. Design I-Section of an connecting rod for a car with the following data: Piston diameter = 68mm. Stroke length = 80 mm. Length of connecting rod = 160mm. Maximum explosion pressure = 3.5 Mpa.	2	Applying (K3)	10			
	Weight of the reciprocating parts = 2.5 kg Speed =4000 rpm. Compression ratio 8:1		2				
3.	Solve the following problem The cylinder of a four-stroke diesel engine has the following specifications: Brake power = 3 kW, Speed = 800 rpm, Indicated mean effective pressure = 0.3 MPa, Mechanical efficiency = 80%. Determine the bore and length of the cylinder liner.	3	Applying (K3)	5			

Wander Bore = 3 moutes Length of cylinder = 9 moutes

E-coiterno

Given date - 1 mall.

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Scheme of Evolution

NARASARAOPET ENGINEERING COLLEGE (AUTONOMOUS): NARASARAOPET DEPARTMENT OF MECHANICAL ENGINEERING

III B. TECH II - SEMESTER ASSIGNMENT TEST – I, MARCH - 2022

SUBJECT: DESIGN OF MACHINE ELEMENTS II	DATE: 15-03-2022
DURATION: 30 MIN	MAX MARKS: 10

Q. No	Questions	Course Outcome (CO)	Knowledge Level as Per Bloom's Taxonomy	Marks
1	Solve the following problem. A journal bearing of 50mm diameter and 80mm long, has a bearing pressure of 6 N/mm ² , the speed of the journal is 1000 rpm. The ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil, whose absolute viscosity at the operating temperature of 75°C may be taken as 0.015 kg/m-s. The room temperature is 25°C, determine (i) the amount of artificial cooling required and(ii) the mass of the coolant oil required, if the difference between the outlet and inlet temperature of the oil is 10°C. the specific heat of the oil is 1900 J/kg/°C and heat dissipation coefficient is 500 W/m ² /°C.	. 1	Applying (K3)	10
2	Solve the following problem. It is required to design a journal bearing for a centrifugal pump running at 1200 rpm. The journal is 8cm and load on each journal is 1800kg. The factor ZN/P for bearing is 2600. The length and diameter ratio are 1.2. Take correction factor as 0.002. The bearing and atmospheric temperatures are 80°C and 30°C respectively.	1	Applying (K3)	10
3	a) Solve the following problem. A journal bearing 80mm in diameter and 120mm long runs at 600rpm. It uses oil having viscosity of 60 centipoise. Radial clearance provided is 0.15mm. Determine the safe load, bearing can take. Assume Sommerfeld number = 1.43 x 10° b) Solve the following problem. A 50mm diameter 75mm long journal bearing is loaded with bearing pressure of 2Mpa. The shaft in bearing rotates at 500rpm. Heat is lost from surface at the rate of 11.6 Joules/m²/sec/°C. The housing area is 8 times the projected area. If the room temperature is 280°C. Determine the surface temperature of bearing. Coefficient of friction is 0.015.	1	Applying (K3)	10
4	Solve the following problem. A journal bearing is to be designed for a centrifugal pump for the following data: Load on the journal = 12 kN; Diameter of the journal = 75 mm; Speed = 1440 r.p.m; Atmospheric temperature of the oil = 16°C; Operating temperature of the oil = 60°C; Absolute viscosity of oil at 60°C = 23 centipoise. Give a systematic design of the bearing.	1	Applying (K3)	10
5	Solve the following problem. Design a journal bearing for a centrifugal pump running at 1440 r.p.m. The diameter of the journal is 100 mm and load on each bearing is 20 kN. The factor ZN/p may be taken as 28 for centrifugal pump bearings. The bearing is running at 75°C temperature and the atmosphere temperature is 30°C. The energy dissipation coefficient is 875 W/m²/°C. Take diametral clearance as 0.1 mm.	1	Applying (K3)	10

1.a) Advantages and disadvantages of the V-belt drive over flat belt drive:

Advantages:-

- 1. The V-belt drive gives compactness due to the small distance between centers of pulleys.
- 2. The drive is positive, because the slip between the belt and the pulley groove is negligible.
- 3. Since the V-belts are made endless and there is no joint trouble, therefore the drive is smooth.
- 4. It provides longer life, 3 to 5 years.
- 5. It can be easily installed and removed.
- 6. The operation of the belt and pulley is quiet.
- 7. The belts have the ability to cushion the shock when machines are started.
- 8. The high velocity ratio (maximum 10) may be obtained.
- 9. The wedging action of the belt in the groove gives high value of limiting *ratio of tensions. Therefore the power transmitted by V-belts is more than flat belts for the same coefficient of friction, arc of contact and allowable tension in the belts.
- 10. The V-belt may be operated in either direction, with tight side of the belt at the top or bottom. The center line may be horizontal, vertical or inclined.

Disadvantages:-

- 1. The V-belt drive can not be used with large center distances, because of larger weight per unit length.
- 2. The V-belts are not so durable as flat belts.
- 3. The construction of pulleys for V-belts is more complicated than pulleys of flat belts.
- 4. Since the V-belts are subjected to certain amount of creep, therefore these are not suitable for constant speed applications such as synchronous machines and timing devices.
- 5. The belt life is greatly influenced with temperature changes, improper belt tension and mismatching of belt lengths.
- 6. The centrifugal tension prevents the use of V-belts at speeds below 5 m/s and above 50 m/s.
- 1.b) Crankshafts materials should be readily shaped, machined and heat-treated, and have adequate strength, toughness, hardness, and high fatigue strength. The crankshaft is manufactured from steel either by forging or casting. The main bearing and connecting rod bearing liners are made of Babbitt, a tin and lead alloy.

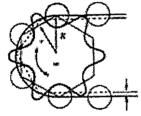
The proportions of the crankshaft are as follows:

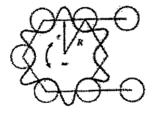
- Cylinder bore diameter = D.
- Cylinder Centre distance = 1.20 D.
- Big-end journals diameter = 0.65 D.
- Main-end journal diameter = 0.75 D.
- Big-end journal width = 0.35 D.
- Main-end journal width = 0.40 D.
- Web thickness = 0.25 D.
- Fillet radius of journal and webs = 0.04 D.

1.d) Straight beam carries only bending moment and shear force whereas curved beam has to carry bending moment, shear force and torsional moment. The torsional moment appears due to beam's geometry, its centroid of geometry (C.G) does not lie on its length.

 $\mathcal{I}_{ullet}(1.e)$ Polygonal effect

The chain passes around the sprocket as a series of chordal-links. This action is similar to a non-slipping belt wrapped around a rotating polygon.





• For smooth operation of chain drives higher number of teeth 19 or 21 gives better life expectancy to the chain with less noise during operation.

• It is preferable to use an odd number of teeth for the driving sprocket in combination with an even number of chain links for uniform wear and tear on the teeth and rollers. (In this case, a particular tooth of the sprocket wheel does not come in contact with a particular link of the chain for every rotation.)

1.f) Positive drive: A positive drive is one that can provide constant velocity ratio during operation. Such drives are free from slip, creep, polygonal effect, leakage, etc. A gear drive is one positive drive. On the other hand, friction drives (belt and rope drive) are affected by slip and creep

diameter 50 mm length 75 mm speed 1600 r.p.m, diametral clearance 0.001 mm ambient temperature 15.5 C oil SAE 10 is used and the film temperature is 60 C. Determine the heat generated and heat dissipated. Take absolute viscosity of SAE at 60 C = 0.014 kg/ms

Solution. Given: W = 3000 N: N = 1600 cp.m.; $t_0 = 60 ^{\circ}\text{C}$: Z = 0.014 kg/m-s: $t_0 = 28 ^{\circ}\text{C}$: : $C = 1232 \text{ W/m}^2/^9\text{C}$ (from data book)

The journal bearing is designed as discussed in the following steps:

I. First of all, let us find the length of the journal (I), the diameter of the journal (d)

as 50 mm.
$$l/d=1.5$$
 $l=1.5 d=75 \,\mathrm{mm}$

2. We know that bearing pressure,

$$p = \frac{W}{l.d} = \frac{3000}{75 \times 50} = 0.8 \,\text{N/mm}^2$$

Since the given bearing pressure for steam englies 1.5 N/mm², therefore the above value of p is safe and hence the dimensions of I and d are safe.

$$\frac{Z.N}{p} = \frac{0.014 \times 1600}{0.8} = 28$$

From Table 26 3, we find that the operating value of

$$\frac{Z.N}{p} = 28$$

We have discussed in Art. 26.14, that the minimum value of the bearing modulus at which the $3 K = \frac{ZN}{}$ oil film will break is given by

... Bearing modulus at the minimum point of friction,

$$K = \frac{1}{3} \left(\frac{Z.N}{p} \right) = \frac{1}{3} \times 28 = 9.33$$

Since the calculated value of bearing characteristic number $\left(\frac{Z.N}{p} = 28\right)$ is more than 9:33, therefore the bearing will operate under hydrodynamic conditions.

- 4. the clearance ratio (c/d) = 0.001
- 5. We know that coefficient of friction,

$$\mu = \frac{33}{10^8} \left(\frac{ZN}{P}\right) \left(\frac{d}{c}\right) + k = \frac{33}{10^8} \times 28 \times \frac{1}{0.001} + 0.002$$

$$= 0.01124 \times k = 0.002$$

6. Heat generated.

$$Q_{\pi} = \mu W V = \mu W \left(\frac{\pi d N}{60}\right) W \qquad ... \left(\forall V = \frac{\pi d N}{60}\right)$$
$$= 0.01124 \times 3000 \quad \left(\frac{\pi \times 0.1 \times 1600}{60}\right) = 211.869 W$$

... (d is taken in metres)

7. Heat dissipated.

$$Q_{d} = C.A (t_{b} - t_{a}) = C.Ld (t_{b} - t_{a}) W \qquad ... (: A = I \times d)$$
We know that $(t_{b} - t_{a}) = \frac{1}{2} (t_{0} - t_{a}) = \frac{1}{2} (.60^{\circ} - 28^{\circ}) = 16^{\circ}C$

$$Q_d = 1232 \times 0.075 \times 0.05 \times 16 = 73.98 \text{ W}$$
 ... (l and d are taken in metres)

2.b)

SKF Bearing Type-Spherical Roller Bearing Axial Load = 2KN Radial Load = 4KN Rotation (N)= 1000 rpm Expected Life L_h =10000 Hrs.= $(L_07)_h r$ = 10000 hrs Probability of Survival - 95 %

- 1) Life of Success $95\% L_{95} = 10000$ Hrs.

2) Life for % Probability of Survival
$$(L_{07})_{mr}$$

$$L_{93} = \frac{L_h x 60 \times N}{10^6} \qquad L_{93} = \frac{10000 \times 60 \times 1000}{10^6} = 324mr$$

$$\frac{L_{95}}{L_{10}} = \left\{\frac{\ln(\frac{1}{\rho_{95}})^{\frac{1}{B}}}{\ln(\frac{1}{\rho_{95}})}\right\}^{\frac{1}{B}}.$$

b = 1.17 for roller bearing

$$\therefore \frac{324}{L'_{1o}} = \left\{ \frac{ln\left(\frac{1}{0.95}\right)}{ln\left(\frac{1}{0.9}\right)} \right\}^{\frac{1}{1377}} = 445.58$$

$$(L'_{1o})_{mr} = 445.58$$

3) Equivalent Load (Pa) -

- $P_e = [V.X.F_r + Y.F_a.]S$ a) V=1 Assumed for inner race rotating
- b) X-Radial Factor & Y-Thrust Factor X & Y, Selecting Bearing Series 22205C to 22207C

e=0.32
Now,
$$\frac{F_0}{F_0} = \frac{1.2}{4} = 0.3$$
 c) $P_0 = [1 \times 1 \times 4 + 2.1 \times 1.2]1.1$
After Comparing with $P_0 = 7.18kN$
 $\therefore \frac{F_0}{F_0} \le e$ Select X & Y

4) Life in millions of revolution for Probability of Survival

$$L'_{10} = \left\{\frac{c}{r_s}\right\}^K$$

K = 10/3 for Roller bearing

From Above equation

find Dynamic Capacity (C) of Bearing -

$$\therefore 445.58 = \left\{ \frac{C}{7.18} \right\}^{10/3} \qquad C = 44.75kN$$

The capacity must be increased for temperature 125°C, using temperature factor Therefore, required capacity is C=1.05x44.75=46.98kN

$$C = 4698Kgf$$

∴ Selecting 22207 Spherical Bearing with C = 5200kgf

Now, Checking Life of Bearing.

$$L'_{10} = \left(\frac{G}{P_c}\right)^K \text{As per selected 'C'}$$

$$L'_{10} = \left\{ \frac{52}{7.55} \right\}^{10/3} = 621.61mr$$

Since $\overline{U_{10}} > (U_{10})_{exp}$

: Thus Bearing Selection will work satisfactorily

Design a connecting rod for an I.C. engine running at 1800 r.p.m. and developing a maximum pressure of 3.15 N/mm². The diameter of the pision is 100 mm; mass of the reciprocating parts percylinder 2.25 kg; length of connecting rod 380 mm; stroke of piston 190 mm and compression ratio 6 : 1. Take a factor of safety of 6 for the design. Take length to diameter ratio for big end bearing as 1,3 and small end bearing as 2 and the corresponding bearing pressures as 10 N/mm² and 15 N/mm². The density of material of the rod may be taken as 8000 kg/m² and the allowable stress in the bolts as 60 N/mm² and in cap as 80 N/mm². The rod is to be of 1-section for which you can choose your own proportions,

Draw a neat dimensioned sketch showing provision for lubrication. Use Rankine formula for which the numerator constant may be taken as 320 N/mm² and the denominator constant L/7500.

Solution. Given: N=1200 rp.m.; $p=5 \text{ N/mm}^2$; D=125 mm; $m_R=2 \text{ kg}$; I=300 mm; = 0.38 m; Stroke = 150 mm; *Compression ratio = 6:1; E S = 5

The connecting rod is designed as discussed below:

1. Dimension of I- section of the connecting rod

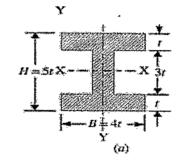
Let us consider an I-section of the connecting rod, as shown in Fig. 32.14 (a), with the following proportions:

Flange and web thickness of the section = tWidth of the section, B = 4t

and depth or height of the section.

$$H = 5$$

First of all, let us find whether the section chosen is satisfactory or not.



We have already discussed that the connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis. The connecting rod should be equally strong in buckling about both the axes. We know that in order to have a connecting rod equally strong about both the axes.

where

$$I_{xx} = 4I_{yy}$$
 $I_{xx} = Moment of inertia of the section about X-axis, and $I_{yy} = Moment of inertia of the section about Y-axis.$$

In actual practice, I_{∞} is kept slightly less than $4I_{\infty}$. It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about X-axis.

Now, for the section as shown in Fig. 32.14 (a), area of the section,

$$A = 2 (4 t \times t) + 3t \times t = 11 t^{2}$$

$$I_{xx} = \frac{1}{12} \left[4t(5t)^{3} - 3t \times (3t)^{3} \right] = \frac{419}{12} t^{4}$$

$$I_{yy} = 2 \times \frac{1}{12} \times t(4t)^{3} + \frac{1}{12} \times 3t \times t^{3} = \frac{131}{12} t^{4}$$

$$I_{yy} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

and

Since $\frac{I_{\text{LSF}}}{I_{\text{min}}}$ = 3.2, therefore the section chosen in quite satisfactory.

Now let us find the dimensions of this *I*-section. Since the connecting rod is designed by taking the force on the connecting rod (F_C) equal to the maximum force on the piston (F_C) due to gas pressure, therefore,

$$F_C = F_{\rm L} = \frac{\pi D^2}{4} \times p = 61359.3 \text{ N}$$

We know that the connecting rod is designed for buckling about X-axis (i.e. in the plane of motion of the connecting rod) assuming both ends hinged. Since a factor of safety is given as 6, therefore the buckling load,

$$W_{\rm B} = F_{\rm C} \times \tilde{F} \, S_{\rm c} = 306796 \, {\rm N}$$

We know that radius of gyration of the section about X-axis,

$$k_{\rm ge} = \sqrt{\frac{I_{\rm six}}{A}} = \sqrt{\frac{419t^4}{12} \times \frac{1}{11t^2}} = 1.78 t$$

Length of crank,

$$r = \frac{\text{Stroke of piston}}{2} = 75 \text{ mm}$$

...(Given)

... Equivalent length of the connecting rod for both ends hinged,

$$L = I = 300 \text{ mm}$$

Now according to Rankine's formula, we know that buckling load (W_n) .

$$306796 = \frac{\sigma_{e}A}{1 + a\left(\frac{L}{k_{xx}}\right)^{2}} = \frac{320 \times 11 r^{2}}{1 + \frac{1}{7500} \left(\frac{380}{1.78 \, r}\right)^{2}}$$

(It is given that $\sigma_c = 320$ MPa or N/mm² and a = 1.77500)

$$\frac{306796}{320} = \frac{11 \, t^2}{1 + \frac{6.1}{t^2}} = \frac{11 \, t^4}{t^2 + 6.1}$$

958
$$(r^2 + 6.1) = 1.1 r^4$$

 1 - 87 t^{2} - 531 \approx 0

$$t^{2} = \frac{\cdot 87 \pm \sqrt{(\cdot 87)^{2} + 4 \times 531}}{2} = \frac{87 \pm 98.4}{2}$$
...(Taking +ye sign)

or

1

or

Thus, the dimensions of I-section of the connecting rod are:

Thickness of flange and web of the section

R = 4 r = 36 mmWidth of the section.

and depth or height of the section,

$$H = 5 t = 45 \, \text{mm}$$

These dimensions are at the middle of the connecting rod. The width (B) is kept constant throughout the length of the rod, but the depth (B) varies. The depth was the big and or coast and is kept as 1.1Bto 1.25H and the depth near the small end or piston end is kept as 0.75H to 0.9H. Let us take

Depth near the big end,

$$H_1 = 1.2H = 54 \text{ mm}$$

and depth near the small end,

$$H_2 = 0.85 H = 38 \text{-mm}$$

. Dimensions of the section near the big end

and dimensions of the section near the small end

Since the connecting rod is manufactured by forging, therefore the sharp corners of I-section are rounded off, as shown in Fig. 32.14 (b), for easy removal of the section from the dies.

2. Dimensions of the crankpin or the big end bearing and piston pin or small end bearing

Lei

 $d_o =$ Diameter of the crankpin or big end bearing.

 $I_{\rm c} = {\rm length}$ of the crankpin or big end bearing = 1.3 $d_{\rm c}$

 $p_{bar} = \text{Bearing pressure} = 10 \text{ N/mm}^2$

We know that load on the crankpin or big end bearings.

$$= d_e J_e \cdot p_{bc} = d_e \times 1.3 d_e \times 10 = 13 (d_e)^2$$

Since the crankpin or the big end bearing is designed for the maximum gas force (F_1) , therefore, equating the load on the crankpin or big end bearing to the maximum gas force, Le.

13
$$(d_c)^2 = F_L = 61359 \text{ N}$$

 $d_c = 68 \text{ mm}$
 $I_c = 1.3 d_c = 89 \text{ mm}$

$$d = 68 \,\mathrm{mm}$$

and

$$I_{\rm c} = 1.3 \, d_{\rm c} = 89 \, {\rm mm}$$

The big end has removable precision bearing shells of brass or bronze or steel with a thin lining (Imm or less) of bearing metal such as babbit.

Again, let

 d_p = Diameter of the piston pin or small end bearing.

 \tilde{t}_n = Length of the piston pin or small end bearing = $2d_p$...(Given)

 $p_{bn} = \text{Bearing pressure} = 15 \text{ N/mm}^2$..(Given)

We know that the load on the piston pin or small end bearing

= Project area × Bearing pressure =
$$d_n$$
, I_n , $p_{bn} = d_n \times 2$, $d_n \times 15 = 30$ $(d_n)^T$

 $M_{\rm C} = \frac{F_1 \times x}{6}$

where

* *

x =Distance between the bolt centres

Dia. of crank pin or big end bearing + 2 × Thickness of bearing liner + Nominal dia. of bolt + Clearance

=
$$(d_c + 2 \times 3 + d_b + 3)$$
 mm = $44 + 6 + 12 + 3 = 65$ mm

... Maximum bending moment acting on the cap,

$$M_{\rm C} = \frac{F_{\rm f} \times x}{6} = \frac{3327 \times 65}{6} = 36042 \text{ N-mm}$$

Section modulus for the cap
$$Z_{\rm C} = \frac{b_{\rm c}(t_{\rm c})^2}{6} = \frac{89 (t_{\rm c})^2}{6} = 14.8 (t_{\rm c})^2$$

We know that bending stress (σ_h),

$$80 = \frac{M_{\rm C}}{Z_{\rm C}} = \frac{36042}{14.8(t_{\rm c})^2}$$

$$t_{\rm c} = 5.15 \,\text{mm}$$

Let us now check the design for the induced bending stress due to inertia bending forces on the connecting rod (i.e. whipping stress).

We know that mass of the connecting rod per metre length,

$$m_1$$
 = Volume × density = Area × length × density
= $A \times I \times P$ = $11t^2 \times I \times P$...(:: $A = 11t^2$)
= $11(0.009)^2 (0.30) 8000 = 2.13$ kg

.: Maximum bending moment,

$$M_{max} = m \cdot \omega^{2} \cdot r \times \frac{l}{9\sqrt{3}} = m_{1} \cdot \omega^{2} \cdot r \times \frac{l^{2}}{9\sqrt{3}}$$

$$= 2.13 \left(\frac{2\pi \times 1200}{60}\right)^{2} \left(0.075\right) \frac{(0.30)^{2}}{9\sqrt{3}} = 14.450 \text{N-mm}$$

$$= 14450 \text{ N-mm}$$

and section modulus

$$Z_{xx} = \frac{I_{xx}}{5t/2} = \frac{419 t^4}{12} \times \frac{2}{5 t} = 13.97 t^3 = 13.97 \times 9^3 = 10184 \text{ mm}^3$$

Amazimum bending stress (induced') due to inertia bending forces or whipping stress,

$$\sigma_{b(max)} = \frac{M_{max}}{Z_{xx}} = 1.41 \text{ N/mm}^2$$

Since the maximum bending stress induced is less than the allowable bending stress of 80 N/mm², therefore the design is safe.

Since the piston pin or the small end bearing is designed for the maximum gas force (F_t) , therefore, equating the load on the piston pin or the small end bearing to the maximum gas force,

i.e.

30
$$(d_p)^2 = 61359 \text{ N}$$

 $d_p = 45 \text{ mm}$
 $I_n = 2 d_n = 90 \text{ mm}$

and

 $l_p=2\ d_p=90\ {\rm mm}$ The small end bearing is usually a phosphor bronze bush of about 3 mm thickness.

3. Size of bolts for securing the big end cap

 d_{ch} = Core diameter of the bolts.

 $\bar{\sigma}_{\cdot}$ = Allowable tensile stress for the material of the bolts $= 60 \text{ N/mm}^2$

...(Given)

and

 n_h = Number of bolts. Generally two bolts are used.

We know that force on the bolts

$$= \frac{\pi}{4} (d_{cb})^2 \sigma_t \times \eta_b = \frac{\pi}{4} (d_{cb})^2 60 \times 2 = 94.26 (d_{cb})^2$$

The bolts and the big end cap are subjected to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead centre on the exhaust stroke. We know that inertia force of the reciprocating parts.

$$F_1 = m_R \cdot \omega^2 \cdot r \left(\cos \theta + \frac{\cos 2\theta}{Ur} \right)$$

We also know that at top dead centre on the exhaust stroke, $\theta = 0$.

$$F_1 = m_R \cdot \omega^2 \cdot r \left(1 + \frac{r}{I} \right) = 2.25 \left(\frac{2\pi \times 1200}{60} \right)^2 0.075 \left(1 + \frac{0.075}{0.3} \right) N$$

$$= 3327 \text{ N}$$

Equating the inertia force to the force on the bolts, we have

3327 = 94.26
$$(d_{ci})^2$$

 $d_{ch} = 5.94 \text{ say 6mm}$

and nominal diameter of the bolt,

$$d_b = \frac{d_{vb}}{0.84} = 7 \text{ mm}$$

4. Thickness of the big end cap

**

 t_a = Thickness of the big end cap,

b. = Width of the big end cap. It is taken equal to the length of the crankpin or big end bearing (l_i)

= 89 mm (calculated above)

 σ_b = Allowable bending stress for the material of the cap

 $= 80 \, \text{N/mm}^2$...(Given)

The big end cap is designed as a beam freely supported at the cap bolt centres and loaded by the inertia force at the top dead centre on the exhaust stroke (i.e. $F_{\rm I}$ when $\theta = 0$). Since the load is assumed to act in between the uniformly distributed load and the centrally concentrated load, therefore, maximum bending moment is taken as

Design Procedure for crank shaft:

3.b) The following procedure may be adopted for designing a crankshaft.

- 1. First of all, find the magnitude of the various loads on the crankshaft.
- 2. Determine the distances between the supports and their position with respect to the loads.
- 3. For the sake of simplicity and also for safety, the shaft is considered to be supported at the centres of the bearings and all the forces and reactions to be acting at these points. The distances between the supports depend on the length of the bearings, which in turn depend on the diameter of the shaft because of the allowable bearing pressures.
- 4. The thickness of the cheeks or webs is assumed to be from 0.4 ds to 0.6 ds, where ds is the diameter of the shaft. It may also be taken as 0.22D to 0.32 D, where D is the bore of cylinder in mm.
- 5. Now calculate the distances between the supports.
- 6. Assuming the allowable bending and shear stresses, determine the main dimensions of the crankshaft.

4.h Piston Rings

The piston rings are used to impart the necessary radial pressure to maintain the seal between the piston and the cylinder bore. These are usually made of grey cast iron or alloy cast iron because of their good wearing properties and also they retain spring characteristics even at high temperatures. The piston rings are of the following two types:

1. Compression rings or pressure rings, and 2. Oil control rings or oil scraper.

The compression rings or pressure rings are inserted in the grooves at the top portion of the piston and may be three to seven in number. These rings also transfer heat from the piston to the cylinder liner and absorb some part of the piston fluctuation due to the side thrust.

The oil control rings or oil scrapers are provided below the compression rings. These rings provide proper lubrication to the liner by allowing sufficient oil to move up during upward stroke and at the same time scraps the lubricating oil from the surface of the liner in order to minimise the flow of the oil to the combustion chamber.

Cylinder bare = 100 mm; Stroke = 125 mm; Maximum gas pressure = 5 N/mm²; Indicated mean affective pressure = 0.75 N/mm²; Mechanical/efficiency = 80%; Fuel consumption = 0.15 kg per brake power per hour; Higher calorific value of fuel = 42×10^8 kH/kg; Specd = 2000 ip/m.

Any other data required for the design may be assumed.

Solution. Given: D = 100 mm; L = 125 mm = 0.125 m; $p = 5 \text{ N/mm}^2$; $p_m = 0.75 \text{ N/mm}^2$; $\eta_m = 80\% = 0.8$; $m = 0.15 \text{ kg} / \text{BP/h} = 41.7 \times 10^{-6} \text{ kg} / \text{BP/s}$; $HCV = 42 \times 10^3 \text{ kJ/kg}$; N = 2000 r.p.m.

The dimensions for various components of the piston are determined as follows:

1. Piston head or crown

The thickness of the piston head or crown is determined on the basis of strength as well as on the basis of heat dissipation and the larger of the two values is adopted.

We know that the thickness of piston head on the basis of strength,

$$t_{\rm H} = \sqrt{\frac{3p.D^2}{16 \sigma_t}} = \sqrt{\frac{3 \times 5(100)^2}{16 \times 38}} = 15.7 \,\text{sny } 16 \,\text{imm}$$

...(Taking G, for cast iron = 38 MPa = 38 N/mm²)

Since the engine is a four stroke engine, therefore, the number of working strokes per minute,

$$n = N/2 = 2000/2 = 1000$$

and cross-sectional area of the cylinder,

$$A = \frac{\pi D^2}{4} = \frac{\pi (100)^2}{4} = 7855 \text{ mm}^3$$

We know that indicated power,

$$IP = \frac{P_{m} \cdot L \cdot A \cdot n}{60} = \frac{0.75 \times 0.125 \times 7855 \times 1000}{60} = 12.27 \text{ W}$$

= 12.27 kW

. Broke power,

$$BP = IP \times \eta_{ee} = 12.27 \times 0.8 = 9.8 \text{ kW}$$

$$...(::\eta_m=BP/IP)$$

We know that the heat flowing through the piston head,

$$H = C \times HCV \times m \times BP$$

= 0.05 \times 42 \times 10^3 \times 41.7 \times 10^{-6} \times 9.8 = 0.86 kW = 860 W
....(Taking $C = 0.05$)

. Thickness of the piston head on the basis of heat dissipation.

$$t_{\rm H} = \frac{H}{12.56 \, k \, (T_{\rm C} - T_{\rm E})} = \frac{860}{12.56 \times 46.6 \times 220} = 0.0067 \, \text{m} = 6.7 \, \text{mm}$$

 $T_{\rm c} = T_{\rm c} = 100 \, \text{GeV/m/°C}$, and $T_{\rm c} = 7_{\rm fg} = 220 \, \text{°C}$

Taking the larger of the two values, we shall adopt $i_{\rm H}=16\,{\rm mm}$ Ans.

Since the ratio of L/D is 1.25, therefore a cup in the top of the piston head with a radius equal to 0.7 D (i.e. 70 mm) is provided.

2. Radial ribs

The radial ribs may be four in number. The thickness of the ribs varies from $t_{\rm H}/3$ to $t_{\rm H}/2$.

z. Thickness of the ribs. $t_R = 16/3$ to 16/2 = 5.33 to 8 mm

Letus adopt

$$t_{\rm R} = 7 \, \rm mm \, Ans,$$

3. Piston rhigs

Let us assume that there are total four rings (i.e. $n_r = 4$) out of which three are compression rings and one is an oil ring.

We know that the radial thickness of the piston rings

$$r_t = D\sqrt{\frac{3 p_w}{\sigma_t}} = 100\sqrt{\frac{3 \times 0.035}{90}} = 3.4 \text{ mm}$$

...(Taking $p_w = 0.035 \text{ N/mm}^3$, and $\sigma_t = 90 \text{ MPa}$)

and axial thickness of the piston rings

$$t_2 = 0.7 t_1$$
 to $t_1 = 0.7 \times 3.4$ to 3.4 mm = 2.38 to 3.4 mm

Let us adopt

$$t_{\rm s} = 3 \, \mathrm{mm}$$

We also know that the minimum axial thickness of the pistion ring,

$$r_2 = \frac{D}{10 n_r} = \frac{100}{10 \times 4} = 2.5 \text{ mm}$$

Thus the axial thickness of the piston ring as already calculated (i.e. $t_2 = 3$ mm) is satisfactory. Ans.

The distance from the top of the piston to the first ring groove, i.e. the width of the top land,

$$b_1 = t_{\rm H}$$
 to 1.2 $t_{\rm H} = 16$ to 1.2 × 16 mm = 16 to 19.2 mm

and width of other ring lands.

$$b_2 = 0.75 t_2$$
 to $t_2 = 0.75 \times 3$ to 3 mm = 2.25 to 3 mm

Let us adopt

 $b_1 = 18 \text{ mm}$; and $b_2 = 2.5 \text{ mm Ans}$.

We know that the gap between the free ends of the ring.

$$G_1 = 3.5 r_1$$
 to $4 r_1 = 3.5 \times 3.4$ to 4×3.4 mm = 11.9 to 13.6 mm

and the gap when the ring is in the cylinder,

$$G_2 = 0.002 D$$
 to 0.004 $D = 0.002 \times 100$ to 0.004 × 100 mm
= 0.2 to 0.4 mm

Let us adopt

$$G_1 = 12.8 \text{ mm}$$
; and $G_2 = 0.3 \text{ mm Ans}$.

4. Piston barrel

Since the radial depth of the piston ring grooves (b) is about 0.4 mm more than the radial thickness of the piston rings (t_1) , therefore, $b = t_1 + 0.4 = 3.4 + 0.4 = 3.8$ mm.

We know that the maximum thickness of barrel,

$$t_3 = 0.03 D + b + 4.5 \text{ mm} = 0.03 \times 100 + 3.8 + 4.5 = 11.3 \text{ mm}$$

and piston wall thickness towards the open end,

$$t_4 = 0.25 t_2$$
 to $0.35 t_3 = 0.25 \times 11.3$ to $0.35 \times 11.3 = 2.8$ to 3.9 mm

Let us adopt

$$r_s = 3.4 \, \text{mm}$$

5. Piston skirt

Let

I = Length of the skirt in mm.

We know that the maximum side thrust on the cylinder due to gas pressure (p).

$$R = \mu \times \frac{\pi D^2}{4} \times p = 0.1 \times \frac{\pi (100)^2}{4} \times 5 = 3928 \text{ N}$$

...(Taking $\mu = 0.1$)

We also know that the side thrust due to bearing pressure on the piston barrel (p_{ij}),

$$R = P_b \times D \times l = 0.45 \times 100 \times l = 45 l \text{ N}$$

...(Taking $p_k \approx 0.45 \text{ N/mm}^2$)

From above, we find that

$$45 l = 3928$$
 or $l = 3928 / 45 = 87.3$ say 90 mm Ans.

.. Total length of the piston.

L = Length of the skirt + Length of the ring section + Top land
=
$$I + (4 t_2 + 3b_2) + b_1$$

= $90 + (4 \times 3 + 3 \times 3) + 18 = 129$ say 130 mm Aus.

6. Pixton min

Let

 $d_0 = \text{Outside diameter of the pin in jum.}$

 $I_1 =$ Length of pin in the bush of the small end of the connecting rod in

 p_{p_1} = Bearing pressure at the small end of the connecting rod bushing in N/mm². It value for bronze bushing is taken as 25 N/mm².

We know that fond on the pin due to bearing pressure

= Bearing pressure × Bearing area =
$$p_{b1} \times d_0 \times l_1$$
 = 25 × d_0 × 0.45 × 100 = 1125 d_0 N ... (Taking l_1 = 0.45 D)

We also know that maximum load on the piston due to gas pressure or maximum gas load

$$= \frac{\pi D^2}{4} \times p = \frac{\pi (100)^2}{4} \times 5 = 39 \ 275 \ \text{N}$$

From above, we find that

1125 $d_0 = 39$ 275 or $d_0 = 39$ 275 / 1125 = 34.9 say 35 mm Ans. $A_0 = 0.6 \times 35 = 21$ mm Ans.

The inside diameter of the pin (d_i) is usually taken as $0.6 d_{02}$

Let the piston pin be made of heat treated alloy steel for which the bending stress (σ_k) may be taken as 140 MPa. Now let us check the induced bending stress in the pin.

We know that maximum bending moment at the centre of the pin.

$$M = \frac{P.D}{8} = \frac{39275 \times 100}{8} = 491 \times 10^3 \text{ N-mm}$$
 We also know that maximum bending moment (M).

$$491 \times 10^{3} = \frac{\pi}{32} \left[\frac{(d_{0})^{4} - (d_{b})^{4}}{d_{0}} \right] \sigma_{b} = \frac{\pi}{32} \left[\frac{(35)^{4} - (21)^{4}}{35} \right] \sigma_{b} = 3664 \sigma_{b}$$

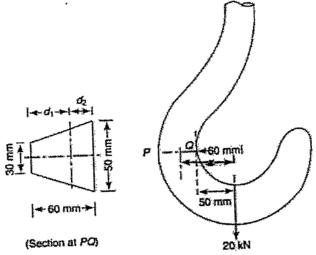
 $\sigma_{tr} = 491 \times 10^3 / 3664 = 134 \text{ N/mm}^2 \text{ or MPa}$

Since the induced bending stress in the pin is less than the permissible value of 140 MPa (Le. 140 N/mm²), therefore, the dimensions for the pin as calculated above (i.e. $d_0 = 35$ mm and $d_i = 21$ mm) are satisfactory.

A crane hook having a trapezoidal horizontal cross-section is 50 mm wide inside and 30 mm wide outside. Thickness of the section is 60 mm. The crane hook carries a vertical load of 20 kN action is 50 mm from the inside edge of the section. The centre of curvature is 60 mm from the inside edge. Determine the maximum tensile and compressive stresses in the section.

Solution

Given A crane hook of trapezoidal horizontal cross-section and loaded as shown in Fig. 9.15 To find Maximum tensile and compressive stresses



$$A = \frac{50 + 30}{2} \times 60 = 2400 \text{ mm}^2; d = 60 \text{ mm};$$

$$d_2 = \frac{d}{3} \left(\frac{2b_1 + b_2}{b_1 + b_2} \right) = \frac{60}{3} \left(\frac{2 \times 30 + 50}{30 + 50} \right) = 27.5 \text{ mm};$$

$$d_1 = 60 - 27.5 = 32.5 \text{ mm};$$

$$R = 60 + 27.5 = 87.5$$

Calculating p2

$$p^{2} = \frac{R^{3}}{A} \left[\left(b_{1} + \frac{(b_{2} - b_{1})}{d} (R + d_{1}) \right) \ln \frac{R + d_{1}}{R - d_{2}} - (b_{2} - b_{1}) \right] - R^{2}$$

$$= \frac{87.5^{3}}{2400} \left[\left(30 + \frac{(50 - 30)}{60} (87.5 + 32.5) \right) \ln \frac{87.5 + 32.5}{87.5 - 27.5} - (50 - 30) \right] - 87.5^{2}$$

$$= 279.1 \left[(30 + 40) \ln 2 - 20 \right] - 87.5^{2} = 303.77 \text{ mm}^{2}$$

Stress at outside face (P)

at ourside face (1)

$$x = 50 + d_2 = 50 + 27.5 = 77.5 \text{ mm}; y = d_1 = 32.5 \text{ mm}$$

$$\sigma_o = \frac{-Wx}{AR} \left[1 + \frac{R^2}{p^2} \frac{y}{R + y} \right] = \frac{20.000 \times 77.5}{2400 \times 87.5} \left[1 + \frac{87.5^3}{303.77} \cdot \frac{32.5}{87.5 + 32.5} \right] = -57.76 \text{ MPa}$$

Stress at inside face (Q)

$$v = d_s = 27.5 \text{ mm}$$

$$\sigma_a = \frac{-Wx}{AR} \left[1 - \frac{R^2}{p^2} \frac{y}{R - y} \right] = -\frac{20\ 000 \times 77.5}{2400 \times 87.5} \left[1 - \frac{87.5^2}{303.77} \cdot \frac{27.5}{87.5 - 27.5} \right] = 77.88 \text{ MPa}$$

Direct stress

Direct stress =
$$\frac{20000}{2400}$$
 = 8.33 MPa

Maximum stresses

At the outside edge, $\sigma_0 = -57.76 + 8.33 = -49.43$ MPa At the inside edge, $\sigma_i = 77.88 + 8.33 = 86.21$ MPa

5.b) Bending stress: Bending stress is the normal stress induced in the beams due to the applied static load or dynamic load. The study of bending stress in beams will be different for the straight beams and curved beams.

Bending stress in straight beams

Consider a straight beam which is subjected to a bending moment M.

- While bending in the beam, the fibres in the upper side in the beam tend to compress (shortened) whereas the fibres in the lower side tend to elongate.
- In between somewhere these upper fibres and the lower fibres, few fibres neither elongate nor shortened. This part of the surface is known as the neutral surface.
- The intersection of these neutral surfaces with any normal cross-section of the beam is known as the Neutral Axis.

Considerations in the Bending Stress in the Straight Beams

There are some considerations has to made while finding the bending stress for the straight beams. They are as follows

- It has to consider that the material throughout the beam is same (Homogeneous material)
- It should obey the Hooke's law (Stress is directly proportional to the strain in the beam)
- Each layer in the beam has to expand or contract freely and independently.
- The load should be applied in the plane of bending
- The Young's modulus is to be same for both the tension and the compression.
- The neutral axis has considered to always pass thru the centroid of the beam.
- For the Symmetrical section(Circle, square, rectangle) the neutral axis passes thru the geometric centre

Curved Beams

- In Straight beam, it is considered that the centroidal axis and the neutral axis coincide. But in curved beams, it is considered that the neutral axis and the centroidal axis do not coincide rather the neutral axis will be shifted towards the centre of curvature.
- Due to the shifting of the neutral axis towards the centre of curvature, the stress distribution in the curved beam will be non-linear.

Bending stress in Curved Beams

Consider an initially curved beam which is subjected to the bending moment M. The sassumptions are made as same as the straight beams (Mentioned at the end of the article). The following are the notable things that we will be observed while finding the bending stress in the curved beams.

- In curved beams, the bending stress induced in the inside fibres tends to tensile stress whereas the outside fibre tends to compress.
- If the section is symmetric (Circular or the rectangle I-beam sections) the maximum bending stress will be induced in the inside fibre
- If the section is unsymmetric then the maximum bending stress may be observed either in inside fibres or outside fibres.

Considerations in the Bending Stress in Curved Beams

There are some considerations has to made while finding the bending stress for the curved beams. They are as follows

- It has to consider that the material throughout the beam is same (Homogeneous material)
- It should obey the Hooke's law (Stress is directly proportional to the strain in the beam)
- Each layer in the beam has to expand or contract freely and independently.
- The load should be applied in the plane(Plane of curvature) of bending
- The Young's modulus is to be same for both the tension and the compression.

6.a) A belt 120 mm wide and .8 mm thick is transmitting power at . 800 metres/min. The net driving tension is 1.6 times the tension on the slack side. If the safe permissible stress on the belt section in 1.8 MPa, calculate the maximum power, that can be transmitted at this speed. Assume density of the leather as 1000 kg/m³.

Calculate the absolute maximum power that can be transmitted by this belt and the speed at which this can be transmitted.

Solution. Given : b = 120 mm = 0.12 m; t = 8 mm = 0.008 m; v = 800 m/min = 13.3 m/s; $T_1 - T_2 = 1.6 T_3$; $\sigma = 1.8 \text{ MPa} = 1.8 \text{ N/mm}^2$; $\rho = 1000 \text{ kg/m}^3$

Power transmitted

Let

 T_1 = Tension in the tight side of the belt, and

 T_a = Tension in the slack side of the belt.

We know that the maximum tension in the belt,

$$T = \sigma.b.j = 1728 \text{ N}$$

Mass of the belt per metre length,

$$m = Area \times Iength \times density = b \times t \times l \times p = 0.8 \text{ Kg/m}$$

... Centrifugal tension,

$$T_{co} = m_e v^2 = 176 \text{ N}$$

We know that

$$T_1 = T - T_{cr} = 1552 \text{ N}$$

and

$$T_1 - T_2 = 1.6 T_2$$

$$T_3 = \frac{T_1}{2.6} = \frac{1552}{2.6} = 596.92 \text{ N}$$

...(Given)

We know that the power transmitted.

$$P = (T_1 - T_2) v = 12702.8 W = 12.702 KW$$

Speed at which absolute maximum power can be transmitted

We know that the speed of the belt for maximum power,

$$v = \sqrt{\frac{T}{3m}} = \sqrt{\frac{1728}{3 \times 1}} = 24 \text{ m/s Ans.}$$

Absolute maximum power

We know that for absolute maximum power, the centrifugal tension,

$$T_{\rm C} = T/3 = 576\,{\rm N}$$

.. Tension in the tight side,

$$T_1 = .T - T_C = 1152 \text{ N}$$

and tension in the slack side,

$$T_2 = \frac{T_1}{2.8} = \frac{1152}{2.8} = 411.4 \text{ N}$$

: Absolute maximum power transmitted,

$$P = (T_1 - T_2) v = 17774.4 W = 17.77 KW$$

The following data is given for an open-type V-belt drive: diameter of driving pulley = 275mm

6.b diameter of driven pulley = 275mm

diameter of driven pulley = 820 mm

groove angle for sheaves = 36°

mass of belt = 0.523 kg/m

maximum permissible tension in belt = 920N

coefficient of friction = 0.24

contact angle for smaller pulley = 140°

speed of smaller pulley = 1150 rev/min

power to be transmitted = 30 kW

How many V-bells should be used, assuming each belt takes its proportional part of the load?

Solution

Given kW = 30 n = 1150 rev/min D = 820mm d = 275mm $\theta = 36^{\circ} m = 0.523 \text{ kg/m}$ f = 0.24 $\alpha_s = 140^{\circ}$ allowable belt tension = 920 N

Step I Belt tensions

$$\frac{f\alpha}{\sin\left(\frac{\theta}{2}\right)} = \frac{0.24\left(\frac{140}{180}\right)\pi}{\sin\left(\frac{36}{2}\right)} = -0.780$$

$$e^{\int d\sin{(\theta/2)}} = e^{\int .874} = 2.18$$

$$v = \frac{\pi dn}{60 \times 10^3} = \frac{\pi (275)(1150)}{60 \times 10^3} = 16.55 \text{ m/s}$$

$$mv^2 = 0.523 (16.55 \text{ s})^2 = 143$$

From Eq. (13.7),
$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha/\sin(\theta/2)}$$

$$\frac{\bar{P}_1 - 143}{P_2 - 143} = 2.18$$

$$P_1 - 2.18P_2 + 168.74 = 0$$
(1)

From Eq. (13.8),

$$\widehat{KW} = \frac{(P_1 - P_2)\widehat{v}}{1000}$$
 or $30 = \frac{(P_1 - P_2)(16.55)}{1000}$

$$P_1 - P_2 - 1812 = 0$$

Solving Eq. (i) and (ii),

$$P_1 = 3490$$
 N and $P_2 = 1678$ N

Step II Number of belts

Number of belts =
$$\frac{\text{Maximum tension in belt}}{\text{Allowable belt load}}$$
$$= \frac{3490}{920} = 3.79 \text{ say 4 belts}$$

A spur gear drive is used to transmit a power of 5 kW. The pinion rotates at 900 rpm; and has diameter of 50 mm. The profile of teeth is 20° full depth. Take material for pinion and gear having a permissible static stress of 200 MPa and 150 MPa respectively. Calculate the module and width of the teeth using Lewis equation. Take Gear ratio = 3.

(a) First step is to find the weaker of two gears. In the problem directly the allowable stress 's' is given, so comparing sy for gear and pinion. [Generally $s_{\omega}y$ is compared, but since $s = s_{\omega}\left(\frac{3}{3+V}\right)$ or $s_{\omega}\left(\frac{6}{6+V}\right)$ or $s_{\omega}\left(\frac{5.6}{5.6+\sqrt{V}}\right)$ as may be the case and V

is constant for gear pair, hence $s = s_o$. The number of teeth on gears are also not given, so considering largest number of teeth on pinion i.e. using least standard module of 1 mm.

 $mN_p = D_p \Rightarrow N_p = 80$ and $N_g = 80 \times 3 = 240$ corresponding $y_p = 0.139$ and $y_g = 0.148$; $(sy)_{pinion} = 200 \times 0.139 = 27.8$ and $(sy)_{gear} = 150 \times 0.148 = 22.2$. Hence, gear is probably weaker.

(b) Since diameters are known, using the Lewis equation, $\frac{1}{m^2 y} = sk\pi^2/F$ = 150 × 10⁶ × 4 × $\pi^2/1326.25$

[Assuming
$$k = 4$$
, $M_t = \frac{5 \times 9550}{900} = 53.05$

Nm,
$$F = \frac{53.05}{(80/2) \times 10^{-3}} = 1326.25 \text{ N}$$

= 4465042,519

(c) Using y = 0.148 in $\frac{1}{m^2 y}$ equation, 'm'

can be calculated as 1.23 mm. Considering standard module of 1.25 m m $N_p = 80/1.25 = 64$ teeth, corresponding $y_p = 0.135$ and $N_p = 64 \times 3 = 192$ teeth, corresponding

$$y_g = 0.147 \cdot \frac{1}{m^2 y}$$
 for gcar = $\frac{1}{\left(\frac{1.25}{1000}\right)^2 \times 0.147}$

= 4353741.97 < 4465042.519, so design is strong.

 $b = k \times \pi m = 4(\pi)(1.25) = 15.7$ taking 15.5-mm

[k can be changed to 4×4353741.971 4465042.519 = 3.9]

- (d) Checking strength of gear and pinion. (sy)_{pinion} = 200 × 0.135 = 27 and (sy)_{gear} = 150 × 0.147 = 22.05; we have also started with weak gear criteria in Step (a).
- (e) Checking gear in permissible stress.

$$s_{\text{grar}} = \frac{F}{b p_c y_x}$$

$$= \frac{1326.25}{0.0155 \times \pi \times 0.00125 \times 0.147}$$

≈ 148.22 MPa

So, induced stress (148.22) < permissible stress (150), gear is safe in strength.

т в такник тон арриканоп, the tool holder is pulled by means of an operating

nut mounted on a screw. The tool holder travels at a speed of 5 m/min. The screw has single-start square threads of 48 mm nominal diameter and 8 mm pitch. The operating nut exerts a force of 500 N to drive the tool holder. The mean radius of the friction collar is 40 mm. The coefficient of friction at thread and collar surfaces is 0.13. Calculate

- (i) power required to drive the screw; and
- (ii) the efficiency of the mechanism.

From Eq. (6.18),

Solution

Given V = 500 N

For screw, d = 48 mm l = p = 8 mm $\mu = 0.15$

For collar, $r_{\rm eff} = 40 \, \text{mm}$ $\mu = 0.15$

Tool holder speed = 5 m/min

Step I Power required to drive the screw

From Eq. (6.2),

$$d_m = d - 0.5p = 48 - 0.5$$
 (8) = 44 mm

$$\tan \alpha = \frac{I}{\pi d_m} = \frac{8}{\pi (44)} = 0.0579 \text{ or } \alpha = 3.312^{\circ}$$

 $\tan \phi = \mu = 0.15 \text{ or } \phi = 8.531^{\circ}$

= 2306.64 N-mm.

From Eq. (6.6),

$$M_i = \frac{Wd_{in}}{2} \tan (\phi + \omega)$$

= $\frac{(500)(44)}{2} \tan (8.531 + 3.312)$

$$(M_l)_c = \frac{\mu_c W}{4} \cdot (D_0 + D_l) = \mu_0 W r_m$$

or $(M_t)_c = \mu_c W r_m = (0.15)(500)(40) = 3000 \text{ N-mm}.$

 $(M_i)_i = M_i + (M_i)_c = 2306.64 + 3000$ = 5306.64 N·mm.

The nut travels at a speed of 5 m/min along with the tool holder. If n is the rpm of the screw, the relationship between the pitch, speed and rpm can be written as,

$$n = \frac{(5 \times 10^3)}{p} = \frac{(5 \times 10^3)}{8} = 625 \text{ rpm}$$

Power required to drive the screw is given by,

$$kW = \frac{2\pi n(M_t)_t}{60 \times 10^6} = \frac{2\pi (625)(5306.64)}{60 \times 10^6} = 0.35$$
 (i)

Step II Efficiency of mechanism

From Eq. (6.20),

$$\eta_0 = \frac{WI}{2\pi (M_c)_c} = \frac{(500)(8)}{2\pi (5306.64)} = 0.12 \text{ or } 12\% (ii)$$

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Subject Code: R16ME3202

III B.Tech II Semester Regular&Supple Examinations, November-2020 DESIGN OF MECHANICAL COMPONENTS

(ME)

Time: 3 hours

Max Marks: 60

Question Paper Consists of Part-A and Part-B.

Answering the question in Part-A is Compulsory & Four Questions should be answered from Part-B All questions carry equal marks of 12.

PART-A

- 1. (a) Why liners are provided in the cylinder? How distortion of liner can be reduced?
 - (b) List any two advantages of rolling contact bearings over sliding contact bearings
 - (c) What is the function of cup shaped cavity in piston head?
 - (d) Write down the assumptions in derivation of Winkler Bach formula.
 - (e) Define pitch and offset of chain drives
 - (f) Different causes of gear tooth failures and suggest possible remedies to avoid such failures

[2+2+2+2+2+2]

PART-B

 $4 \times 12 = 48$

- 2. A rolling contact bearing is subjected to the following work cycle: (a) Radial load of 6000 N at 150 r.p.m. for 25% of the time; (b) Radial load of 7500 N at 600 r.p.m. for 20% of the time; and (c) Radial load of 2000 N at 300 r.p.m. for 55% of the time. The inner ring rotates and loads are steady. Select a bearing for an expected average life of 2500 hours.
- 3. Design a side or overhung crank shaft for a single cylinder petrol engine having following specifications Size = 500 mm by 600 mm, maximum pressure on piston = 2.1 N/mm², l/r = 4.5, shaft material = 40C8 for crankpin bearing assume l/d = 1 and pressure should not exceed 10 N/mm². For main bearing assume l/d = 1 and pressure should not exceed 9 N/mm². The side crank carries a flywheel whose weight is 40 KN and is in between the two journal bearings which act as main bearing. The distance between the bearing may be taken as 600 mm. The cylinder of the engine is horizontal. The maximum torque on the crank occurs at an angle of 30°. The pressure at maximum torque is about half of the maximum pressure. Assume any other data.
- 4. Design a trunk type cast iron piston for a 4-stroke diesel engine with the following specifications: cylinder bore = 250 mm, stroke length = 375 mm, speed = 600 rpm, maximum gas pressure = 5 MPa, indicated mean effective pressure = 0.8 MPa, rate of fuel consumption = 0.3 Kg/BP/Hr, higher calorific value of fuel = 44 MJ/Kg, mechanical efficiency = 80%. Sate clearly the design decision taken.
- 5. a) Find the expression of R_N for rectangular cross-section.
- b) Find the expression for bending stress produced in a curved bar which is subjected to bending moment.

- 6. The following data are given for a V-belt drive: design power = 20 KW, diameter of driving pulley = 200mm, speed of smaller pulley = 1120 rpm, velocity ratio = 3, groove angle = 38°, centrifugal force = 125 N, permissible tension in the belt = 550 N, coefficient of friction = 0.3, angle of contact at smaller pulley = 156°, centre distance = 2m. Determine the number of belts and the life of the belt. The belt is to be used for 8 hours per day.
- 7. A gear drive is required to transmit a maximum power of 22.5 kW. The velocity ratio is 1:2 and r.p.m. of the pinion is 200. The approximate centre distance between the shafts may be taken as 600 mm. The teeth has 20° stub involute profiles. The static stress for the gear material (which is cast iron) may be taken as 60 MPa and face width as 10 times the module. Find the module, face width and number of teeth on each gear. Check the design for dynamic and wear loads. The deformation or dynamic factor in the Buckingham equation may be taken as 80 and the material combination factor for the wear as 1.4.

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1. The cylinder liner prevents the compressed gas and combustion gas from escaping outside. It is necessary that a cylinder liner which is hard to transform by high pressure and high temperature in the cylinder. For that reason, cast iron cylinder liners are used in most cases

The cylinder liner wear due to corrosion is caused due to these reasons: — Burning of heavy fuel oil in the combustion space: This happens because heavy fuel oil contains high sulfur content.

Pretightening process is required to reduce such distortion.

2. Advantages of Rolling Contact Bearings over Sliding Contact Bearings

- Low starting and running friction except at very high speeds.
- Ability to withstand momentary shock loads.
- Accuracy of shaft alignment.
- Low cost of maintenance, as no lubrication is required while in service.
- Small overall dimensions.
- · Reliability of service.
- Easy to mount and erect.
- Cleanliness
- 3. The piston has a bowl in its crown. As the piston approaches the top of the cylinder the rotating air is forced into the piston bowl. The rotational force is magnified by the reduced diameter of the piston bowl. Thin, deep bowls have a higher swirl rate.

At the same time air squish is initiated as the air is forced from above the piston crown in towards the centre of the piston. The squish forces meet at the centre of the cylinder and oppose each other being forced downwards into the bowl where they follow the bowl profile being led into a horizontal swirl.

4. The following assumptions are made:

- Transverse sections which are plane before bending remain plane after bending.
- Hook's law is applicable. This means the working stresses are below the limit of proportionality.
- The longitudinal fibres of the bar, parallel to centroidal axis exert no pressure on each other. This means the distance between any longitudinal fibre from centroidal axis is same before and after bending.
- Each layer of the beam is free to expand or contract, independently of the layer, above or below it.
- 5. **Pitch of chain.** It is the distance between the hinge centre of a link and the corresponding hinge centre of the adjacent link. It is usually denoted by p. When the chain has an odd number of links, an additional link, called 'offset'

6. Causes of Gear Tooth Failure

The different modes of failure of gear teeth and their possible remedies to avoid the failure, are as

1. Bending failure. Every gear tooth acts as a cantilever. If the total repetitive dynamic load acting on the gear tooth is greater than the beam strength of the gear tooth, then the gear tooth will fail in bending, i.e. the gear tooth will break.

200

In order to avoid such failure, the module and face width of the gear is adjusted so that the beam strength is greater than the dynamic load.

2. *Pitting*. It is the surface fatigue failure which occurs due to many repetition of Hertz contact stresses. The failure occurs when the surface contact stresses are higher than the endurance limit of the material. The failure starts with the formation of pits which continue to grow resulting in the rupture of the tooth surface.

In order to avoid the pitting, the dynamic load between the gear tooth should be less than the wear strength of the gear tooth.

3. *Scoring*. The excessive heat is generated when there is an excessive surface pressure, high speed or supply of lubricant fails. It is a stick-slip phenomenon in which alternate shearing and welding takes place rapidly at high spots.

This type of failure can be avoided by properly designing the parameters such as speed, pressure and proper flow of the lubricant, so that the temperature at the rubbing faces is within the permissible flimits.

- 4. Abrasive wear. The foreign particles in the lubricants such as dirt, dust or burr enter between the tooth and damage the form of tooth. This type of failure can be avoided by providing filters for the lubricating oil or by using high viscosity lubricant oil which enables the formation of thicker oil film and hence permits easy passage of such particles without damaging the gear surface.
- 5. Corrosive wear. The corrosion of the tooth surfaces is mainly caused due to the presence of corrosive elements such as additives present in the lubricating oils. In order to avoid this type of wear, proper anti-corrosive additives should be used.

1					
(Soi)	Sl. No	Radial hall	Element	(rpm)	
	0	6000	ବହ୍ୟି: = ୦:ସହ	150	0.25 × 150 = 37.5 m (N)
4 4	(R)	7500	㺾. = 0 ∙30	600	0-20×600 = 120 rev (Nz)
1	(<u>3</u>)	\$000 ·	55·1. = 0·55	300	0.55 × 300 = 165 xer (N3)
Ac	Equivale Cording L10 =	60 N Froh 106 load capacity C	13+ No We3 + Na + N. 120 (7500)3+ 15+120+165 relationship 60 (3225) (19) 106 106 106 107 107 10,806N	(500) = 48.	= 5710.17010
3	Si- Cylin Cram Man Shal Flye	wheel weight Wo	500 mm, 5 500 mm, 5 2 = 600 = 300 0 c 8, 41 = 10kp = 40; 2 + 0kp = 40; 2 + 0kp = 40;	1., Masim x10310.	

I Design of crankshoft when the ownk is at dead Gate: -Load Fp= 1 Dxp= 1 (500)x2.1 1Fp = 412 × 103 N.

a Design of comp pin: de = dia of cante pin L= length of Grank pin = 0.8de Fp=delePb 412×103= 9 (6.89 c) × (10)

:: 9 = 326.6 ~ 331 mm Lc = 0.8 de = 0.8 (997)

1. Ye = 181mm Bending monat at the carsh pin, M = 3 x Fpx/c = 3 x412x163x181 M = 55929000

section modules Z= Tolde)3 Z = TT (227)3 = 1147776.26

Bending stors = M

= 48.728 D/mm?. smake induced stres is less than perhabile limite of 600/mm. Therefore the Lesign is sale.

: - de = 227 mm, · le = 18 mm

(1) Design of Begings: let d, - Dia of bearing 1). Thickness of Crank web, t=0.6dc=0.6x227=136.2mm Length of the beging 1,=1.7de li=1.7(227)= 386.9= 386 mm. Banding moment at the centre of the

M=Fp(0.75xlc+++0.5l,) = 412×103 (0.75×181+136.2+0.5(386) M = 191.55 x 106 p-mm. But, bending moment, W= II (91)30P TT (d,)30 = 191.55 × 106 [: Assur & 500) d, = 3/191.55 × 106 × 32 d, = 319.23 mm. @ Dealgn of Grank Web Let w- width of andcoreb in mm.

Bending moment on Crunk web. M=Fp(0.75 lc+0.5t) = 412×103 (0.75×181 +0.5(136.8)) = 83.9×106 = 84×106 pmm. Section modulus, Z= 1/6 xxx 12 $=\frac{1}{6}\times \omega\times (136.8)^{2}$ Z = 3091.74 W mm3. : Bending stress = M

б = 27.136 × 103 Direct Compressive stress,

o] = Fp = 412 x103 = 3402 x103 .. Total stres on Grank web.

$$= \frac{n}{30.538 \times 10^{3}}$$

$$= \frac{n}{3.0136 \times 10^{3}} + \frac{n}{3.10.5 \times 10^{3}}$$

$$= \frac{P + 2}{10.5 \times 10^{3}}$$

= \$0.1 × (005) × 1.05 Pieton gue bad. Fp = 1 Dang nateig is of an angle of maximum tuding @ Design of Gran bakel when the curile me.526 = P 20 x = 0 x = 001 x T.37 Jox 26x TE = 9M forman gribus - min-0,01 × L.9L = 8W 8(301× 1-12)+8(201× 2.31) /= 8 = 1 (MH) = BH - harmon pribned forbluesh .. mm-dgalxL. n= athx falxah = ~W My = Makes Waxb = Axaxb = White Apos Sandagh The vertical bending moneral due to the m-0901x 25.25 = 1M = 2M +1M = 4M . Loren pribused beforered betat. 0=gM =: 1/4q Had self at sub in those lest constrion or sel Sinc, the is no belt pull, the will mn-0391 x 759.9L= E OLH YEAR SEL - (BED+ BLE) EDIX GIM = M, = Fp (a+ b2) - H, b2. It so froman pribrial betrastion sil Since her is no tension. He havisoned -reaction due to the belt is inglocked.

NE SOXOB

W = Slaxed = solu = eV

Thusk in Connecting rod. Fa = FP [: = 4.5] d=sin (0.111)= 6.379°. : Fa = Fp = 206062.5 Cas (6.379°) = Fa = 207097.9 = 207098N Tangental for a acting on the annicabet. FT = FQ (Sin0+0) = 207098.5in (20+637) FT = 200684 = 200 x103 10. Radial force FR = Facces (0+4) FR = 207098 Cas (30°+6.379°) = 503×10310. Due to tangential form 'FT' the horizontal reaction at bearing - 1, HT, = FT (a+b) 200×10-x (3719+470) = 358.5×1030. The horizontal reaction at bearing - 2 HTE = FT XQ = 200 X 103 x 372 HT2 = 158.2 × 103N Die to radial force 'FR' the horizonal reaction at bearing "1" HR, = FR (9+b) = 50×103×(371.9+479) HR, = 89.57 x 103 N The horizontal reaction at bearing - 2, HRe = FR(a) = SOX103(372) HRL = 39.57 × 103 N.

@ Design of cumle useb: BM due to tangential force. Mbr = Fr[8-4] = 300 ×103 300 - 350 MbT = 28 × 106 10-mm. ". Bending stess duto tangeted force "bT = MbT = 6 x 98 x 106 0 Pet = 1.86 = 20/mm [: 5=6tw] Bn dut radial fora, MbR = Fr (0.75/c+0.5t) = 503×103 (0.75(181)+0.5(136)) MPB = 10 5 × 10 pmm. Bending stresdu to radial Porce PB = 2 = 0 + 3 = 2007 × 136 5R = 17.71 RHAN /mm2. Direct Compressive stress, of = FR = 502×103 = 7.330/mm?. .. Total Compressive stres 0 = bT + bR + T = 5+17+7 = 29 N/mm3. :. Tuisting moment du to Tongetel Porce T= FT (0.75 lc+0.5t) = 20x10 (0.75(181) +0.5(136)) T= 40.75 × 106 10- mm. : Shear stres . I = Tp [: Zp= Lis] = 4.5T = 4.5 × 40.7 × 10 E= 19.64N/mm3. " Total or maximum stres, max = = = + = V(E) 3+4t2 = 29 + 1 (29) 2+4(19.6)2 = 38.8 = 39 p/mm2.

11 to 416/12 to 010 - 126 441 Design of shoft of Jundand Cank: Merter, Corolde de = 297,3mm. ib nothrossal 2 is be souther aft . wm E.LbB = 5P 79.61× (9) 11 = 901× E.101 JE (26) TE = 31 02/A 901× 59E.101 = 3] 8/31×09)+6(81×18) = FT+8AM = 2T Jasman Britabist Indurings mm-4201x00 = 00 Ex Edix00 = 5 x - 7 = 7 . Hade all no troman pritains! W = 81.83×10g 12- mm. 8(301×1.18) + 8(301×1.18) V = · framon pribred trishess ..

FUM + FHM = AM SEGX SEGX EDIXON TOG 1900 W Vertical bending manont due to Ayushad MM-M BOIXT-18 = BOIXE-ED-BOIXER = HM MH = Pp (0+b2) - (4/R) 3+ (HT) 3 b2 brod eop rateig at mb Hode of hadreyt when Hale to rejest a slabermay mall sool or water still DMP2. Assess of GON/my though its design is sold

As a meles of 30H my to LOD hom? NOE less then the permissible limit of 2) rests the Induced share stees 19 1. T = 12.88 phones.

J > E(P. PLE) II = BOIX 28.88 --- EB. P = 15 P : 500 1357 × (15p) II = 901x 56.68] - 1 = 3] = 3] drawon Pritainst tedorings Jul · mm-01 901 X 56.68 = 31 = ((901 x 09) + (901 x 8 L & - 9 5) / = 8T+FM V= = T Jamon Endered Indulys .. . mu - 01,01×09 =

8x TA = T, Framon Posseul

((251) + (181) SL.O) & 60LOB =

BM of the junction of Charoli

701 × 816.95 = W

W= Fer (0.75/c+t)

MC18.82 = 8.0x P2.ET = IP = 13593 M = 13.59km. · 8/4-0 05L. E 65EL = 3/mm-N 05L/EB5EL = d.I. -: = 4.7 Corsessory x215 x8.0 = M4 1.mg = 9.1 Indicated home power. Form 2.630PH = F(029) IT = FO IT = A Area of cross section of the cylinder (37-5T) 12561 = HJ The Michness of piston head, (b) Thornal Consideration: -:. FH = 39.00mm. (i. For CI, of = 380/mm) 8Ex91 / = 709E / H7. Lg mile ai bosh matery to result show the - : menon to had mitely () For a 4-state Engine.

4. MED-19 to 80.E1 = 9+ 76.95 7 76.95 = 87 中一种 = 4 = 4 = 4 ~ ~ The Michorage of the pickers cool FH = 30.3 Emm. Considering large of the rate mugLE80.0 = H7 10.788×103 [>0/m/009.24=4 [: For CT (TE-TE)= 2202 and 上ば = 19.56k(nc.TE) . The clemes of pisters head. · m = 10.788 km = 10x 188 × 103 m. 18.85x5580000.0 x501x24 x 20.0 = H 20.0= > -m2=A H = Cx Hcn xmx B.b given by ei board rated Aquard and F. Jast Bb= 28.818km.

Higher cabilitie value of Rich (H.C.V) = LHMJ/18 = HHX103KS/18. Fied concerptions 0.3 kg (Bp/HV = 0.000833 kg/Bp/sec Indicated mem afterly present of P. P. = 0.8 mm bolos that Glinde bur D=250mm, Speed N=600rpm.

Glinde bur D=250mm Max grees preser P=5mpa=50/m² (rated rai teas) engine (rated .: 14.2)

Mechanial efficiency much =-801.

(1.0=1) (1×1.0= (1.0=1) (1×1.0= (1×0=1×8)= (R= Mx TT x D3 x P The colinder, The maximum side though on - : Jrida malei9 @ murros. d = wit Considering an average ratue, . musine. L of sell.5= (69.00) ST. 0 07 (69.00) 56.0= dder.o at ater.o = wt Pister wall thickness township mun69.00 = 97 (5.4+698+(056) 60.0)= mm(9.7+9+080.0)=97 Asnod to sursbart municoM 69.8=4.0+66.8=4.0+7=9 . Escarp Brir madeig to Algod - Marie mater (mmgr.0 2 mm 1 2 5.0 = en .. (038) 200.0 of (038) 800.0 = ۵،00،0 مل ۱۹۵۵۰۵ = وی , bd mark 21 warmy with my maker after of all oals mm80.15 ~ mm91.55 9 510.68 = (68.8) 4 of (68.8) 2.5 == サカササラをもり Bring & bed and all mosusted good mm L of 56.5= L of (r) 51.0 = 69

of 2) 01200 = ed rebod sprir sho to Abia (38.96)8.1 \$ 36.86= H7 5-1 7 H7 = 19 bool got ill & Abien (mm = of = 1) masiles des Thus, the arich thickness of The , muss. 9 = 97 m (H= 14 Buy p soquine (truben) 25.6 = 4×01 = 25 = 53 . The minimum asid hickoness of the · wungnar = of Condiding an areage value. , mmpg.8 2 508.5= すずないのこのすり egitinateig all to searshill lowA " + = 8.30 mm. (find Gop = 30 sends disna) dazemog 2 em/0250.0= 19 inserry Macy Labor galasseA) 4E D. V = 47 Ed coup 21 sprinded & sepressial bibor sit - leguir nateia (E) mm 55E.91= 47 adia to easodaily sport

Also, the side that on the piston barrel (Pb) R = Pb x Dyl = 0.45 x 956xl (: Bearing pressure Ph 10 limited to 0.45 10/mm?) R=112.51 -> 3 Equating Quation (1) and (3) 7=0-80-mm. 1= 218.055 mm .. Total highest the piston 1 = Length of the skirt + length of the ringsection + Top land L= l+ (4.ta+3ba)+b, = 218+(4(7)+3(6))+63) 1 = 307mm (6) Piston pin (or) Gudgeon pin:-Let Pb, - Bearing pressure at bushing for brown bush = 95N/m do - outer diameter of pin in li-length of pin in imm, Maximum load on the piston dueto gas pressure = TDOX P = 1 (990) 9 x 5=945,217.54 The load on the pin due to bearing

= Pb,x do *1,

= 25 x do x (0,45 x 250) (" 1=0.45D). = 2812.5 do -> @ Equating of () and (1) we got, 245312.5 = 2812.5 26 90= 81.22 mm * do = 87mm India dramater of the pin, d: = 0.6 x do = 0.6 x 87 d: = 52.2 mm checking the Induce I Bending store in Maximum BM at the conte of pin, Mmax = P.D = 34531.2 × 250 Mman = 766,600 D-mm. Also, the maximum B.M is grund Mmax = II do 4 - di 4 xon 0 = 39M [do] = 32×766,600 87 (87) 4- (599)4 = 7,812,484 \ L9865007 - 06 = 13.63052N/mm? sing the induced bending stress is well below the permissible = Booing Pressur & Bearing ones stress of 140 D/m? Therefore = Pb, x do xl, the design is Sak.

· (e) Cat). Orange borner b, = 30 km = 30 x1030. Diameter of thing pulley D, = 200mm, Grand of finallar pulley NI = 1120 pm, Velocity ratio = 3, greene angle 23 = 38°. Cartifingal Force To = 195N. Permissible temsion in the belt Ti = 550 D. Coefficient of Riction M=0.3. Angle of contact at smaller pulley = 1560. Centre distance & = 2m = 2000mm. Service = 8 hours per day Power P= Design power = $\frac{20 \times 10^3}{Fa} = \frac{20 \times 10^3}{1} = 20 \times 10^3$ The conaction Pactur according to service (8 hours/day) is 1. .. Pour P = 20×103 cm. Velocity voto $\frac{N_e}{N_1} = \frac{d_1}{d_2} = 3$ $\Rightarrow \frac{N_2}{N_1} = 3$, $\frac{d_1}{d_2} = 3$. N2 = 3 (N,) = 3(1180) = 3360 pm. de = di/3 = 200 = 66.66mm. Type of Goss-Section: According to designed power and speed of motor, the cross-sata of v-belt is to Pitch Dianete of pulleys: -From the standards of cross-section. He minimum pitch diameter for C-section belt is 200 mm. length of Belt dine:
1 - 9x | Tr(D+d) + (D-d)2

,

Fi = Courtie hat Roved contact. for election and 4450 mm pitch langh. Fe= 1.0 4. where Fe = consolies for Ported langh. North belt of PXFa. · morelog=x someth who houted .. [mmg106= x 8888 - 9567907 / 7 9106 (5069)(1)H - (3100) + (9106-) - = x · O= 668'6 + REL9. 9106 - 5x ** 55.1507 = 5.66661 + 3x8 & 55.1507 = 5.66661 + x8 5. blll + 59. 8ih + x8 = 0544 ×17 (99.99-006) + (99.99+006) IL + x6 = 05+H = (P-0) + (P+0) IL + x6 = 7 endaib who lated Lysomier) H628mm. Thus the pitch bugh of belt is askned 21 Also Losses and how cosection ball is Desirable pitch bongain-· ww. L8, 944 4 = 66,66 + 59.814 + 0004 = 7

44

6(99.99 - 006) + (99.99 + 006) IL + (0006) 6=7

An of Gotnot, et = 180-2515-1 [D-1 d - 180 - 3512 1 3(2015) of = 180 - 2012 (0.0330) > 9 = 180 - 3(4.804) 2-176.218 · [m 2=176.018 , Fd = 0.99. For c- section 1120 pm and some pulley, Pr = 6,25+0.99 = 7.24kw. V) = ROX 104 × 1 = 2.68 = 3 : Nonteer of belts , n= 3 belts.

Empression Per Rn Par Rectangular Cross Section:

Let b- width of rectangula cross section h-Depth of rectargular coses section

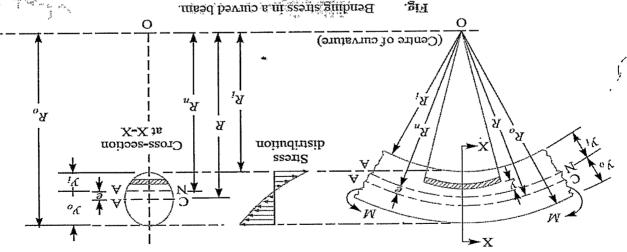
JA - Aca of coss section = bxh

R: - Radius of construct inner libre. Ro-Radius of construct of outer libre.

The neutral axis of this Goss section is detired by relation.

We have seen in the previous article that for the straight beams, the neutral axis of the section causing a non-linear (hyperbolic) distribution of stress, as shown in Fig. It may be noted that the neutral axis lies between the centroidal axis and the centre of curvature and always occurs within the neutral axis lies between the centroidal axis and the centre of curvature and always occurs within the neutral axis lies between the centroidal axis and the centre of curvature and always occurs within the neutral axis lies between the centroidal axis and the centre of curvature and always occurs within the curved beams. The application of curved beam principle is used in crane hooks, chain links and

frames of punches, presses, planers etc.



Consider a curved beam subjected to a bending moment M, as shown in Fig. 5.8. In finding the bending stress in curved beams, the same assumptions are used as for straight beams. The general expression for the bending stress (σ_h) in a curved beam at any fibre at a distance y from the neutral

 $\left(\frac{q}{q-n}\right)\frac{M}{a\cdot h}={}_{d}O$

M =Bending moment acting at the given section about the centroidal

axis.

A = Area of cross-section, a = Area of cross-section. $a = A - A_n$, $a = A - A_n$, $b = A_n$, $b = A_n$, b =

B = Pading of curvainte of the centroidal axis

R = Radius of curvature of the centroidal axis,

 $R_n = Radius$ of curvature of the neutral axis, and v = Pistance from the neutral axis to the fibre under consideration. It is

 $\overline{y}=$ Distance from the neutral axis to the fibre under consideration. It is positive for the distances towards the centre of curvature and negative for the distances away from the centre of curvature.

Notes: I. The bending stress in the curved beam is zero at a point other than at the centroidal axis.

2. If the section is symmetrical such as a circle, rectangle, I-beam with equal flanges, then the maximum

Dending stress will always occur at the inside fibre.

3. If the section is unsymmetrical, then the maximum bending stress may occur at either the inside fibre

or the outside fibre. The maximum bending stress at the inside fibre is given by

$$\sigma_{bi} = \frac{M \cdot N_i}{A \cdot a \cdot k} = \sigma_{bi}$$

 y_i = Distance from the neutral axis to the inside fibre = $R_n - R_i$, and $R_i = R_i$ and $R_i = R_i$

 $R_{\rm i}=R$ adius of curvature of the inside fibre. The maximum bending stress at the outside fibre is given by

$$\frac{o^{V_1} \cdot W_1}{o^{N_1} \cdot o \cdot A} = o_0$$

 $P_{o}=P_{o}=P_{o}+R_{n}$ and the neutral axis to the outside fibre = $R_{o}-R_{n}$, and

where
$$y_o = D$$
 is an are from the neutral axis to the out $A_o = R$ adius of curvature of the outside fibre.

MUSES

Where

axis, is given by

It may be noted that the bending stress at the inside fibre is tensile while the bending stress at the outside

Thre is compressive.

4. If the section has an axial load in addition to bending, then the axial or direct stress (σ_d) must be added algebraically to the bending stress, in order to obtain the resultant stress on the section. In other words,

7 Solution. Given: P = 22.5 kW = 22500 W; $VR = D_G/D_p = 2$; $N_p = 200 \text{ r.p.m.}$; L = 600 mm; C = 80;

Module 1

m = Module in nun,

 $D_{\rm p}$ = Pitch circle diameter of the pinion, and

 D_G = Pitch circle diameter of the gear.

We know that centre distance between the shafts (L).

$$600 = \frac{D_{\rm p}}{2} \div \frac{D_{\rm G}}{2} = \frac{D_{\rm p}}{2} + \frac{2D_{\rm p}}{2} = 1.5 D_{\rm p} \qquad \dots (\because D_{\rm G} = V.R. \times D_{\rm p})$$

$$D_{\rm p} = 600 / 1.5 = 400 \text{ mm} = 0.4 \text{ m}$$

$$D_{\rm G} = 2 D_{\rm p} = 2 \times 400 = 800 \text{ mm} = 0.8 \text{ m}$$

and

Since both the gears are made of the same material, therefore pinion is the weaker. Thus the design will be based upon the pinion.

We know that pitch line velocity of the pinion

$$v = \frac{\pi D_{\rm p} . N_{\rm p}}{60} = \frac{\pi \times 0.4 \times 200}{60} = 4.2 \text{ m/s}$$

Since ν is less than 12 m/s, therefore velocity factor,

$$C_v = \frac{3}{3+v} = \frac{3}{3+4.2} = 0.417$$

We know that number of teeth on the pinion, $T_{\rm p} = D_{\rm p}/m = 400 / m$

.. Tooth form factor for the pinion,

$$J_{\rm p} = 0.175 - \frac{0.841}{T_{\rm p}} = 0.175 - \frac{0.841 \times m}{400}$$
 ... (For 20° stub system)
= 0.175 - 0.0021 m ...(i)

Assuming steady load conditions and 8–10 hours of service per day, the service factor (C_s) from Table is given by $C_c = 1$

We know that design tangential tooth load,

$$W_{\rm T} = \frac{P}{V} \times C_{\rm S} = \frac{22\ 500}{4.2} \times 1 = 5357\ {\rm N}$$

We also know that tangential tooth load (W_T) ,

$$5357 = \sigma_{wP} \cdot b \cdot \pi m \cdot y_P = (\sigma_{OP} \times C_v) b \cdot \pi m \cdot y_P$$

$$= (60 \times 0.417) \ 10 \ m \times \pi m \ (0.175 - 0.0021 \ m)$$

$$= 137.6 \ m^2 - 1.65 \ m^3$$

Solving this equation by hit and trial method, we find that

$$m = 0.65$$
 say 8 mm Ans.

Face width $b = 10 m = 10 \times 8 = 80 \text{ mm Ans.}$

Number of teeth on the gears $T_p = D_p/m = 400/8 = 50$ Ans.

and number of teeth on the gear, $T_G = D_G/m = 800/8 = 100$ Ans.

Checking the gears for dynamic and wear load dynamic load,

$$W_{\rm D} = W_{\rm T} + \frac{21v \left(b.C \middle| + W_{\rm T}\right)}{21v + \sqrt{b.C + W_{\rm T}}} = 5357 + \frac{21 \times 4.2 \left(80 \times 80 + 5357\right)}{21 \times 4.2 + \sqrt{80 \times 80 + 5357}} = 10 630 \text{ N}$$

From equation (i), we find that tooth form factor for the pinion,

$$y_p = 0.175 - 0.0021 \text{ } m = 0.175 - 0.0021 \times 8 = 0.1582$$

From Table we find that flexural endurance limit (σ_a) for cast iron is 84 MPa or 84 N/mm².

.. Static tooth load or endurance strength of the tooth,

$$W_{\rm S} = \sigma_{\rm g} \cdot b \cdot \pi \, m \cdot y_{\rm p} = 84 \times 80 \times \pi \times 8 \times 0.1582 = 26722 \,\text{N}$$

We know that ratio factor, $Q = \frac{2 \times V.R.}{V.R. + 1} = \frac{2 \times 2}{2 + 1} = 1.33$

: Maximum or limiting load for wear. $W_w = D_p$. b. Q. $K = 400 \times 80 \times 1.33 \times 1.4 = 59584$ N

29 DESIGN OF LEVERS

29.1 Introduction

Lever is a simple mechanical device, in the form of a straight or curved link or a rigid rod, pivoted about the fulcrum. It works on the principle of moments and is used to get mechanical advantage and sometimes to facilitate the application of force in a desired direction. Examples of levers are: straight tommy bar used to operate screw jack, bell crank lever, rocker arm, lever of lever loaded safety valve etc. Figure 29.1 shows the construction of a simple lever. P is the applied effort required to overcome load, W.

Ratio of load to effort is called Mechanical Advantage and ratio of effort arm length to load arm length is called leverage.

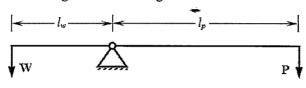


Figure 29.1 Lever

$$Mechanical Advantage = \frac{Load}{Effort} = \frac{W}{P}$$

$$Leverage = \frac{Effort Arm Length}{Load Arm Length} = \frac{l_p}{l_w}$$

29.2 Classes of Levers

Depending upon the position of load point, effort point and fulcrum, levers are classified into following classes:

	Class I Levers	Lever having the fulcrum located between the load point and effort point is called Class I lever. Examples are rocker arm, bell crank lever etc. Mechanical advantage of such levers is greater than one as effort arm is larger than the load arm.			
Class II Lever having load point located between the fulcrum and effort point is called Class II lever. Lever used in example of lever of this class. The effort arm is larger than the load arm; therefore the mechanical advantage one.					
	Class III Levers	Lever having effort point located between the fulcrum and load point is called Class III lever. The effort arm, in this case, is smaller than the load arm; therefore the mechanical advantage is less than one. Due to this, the use of such type of levers is not recommended. However a pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever.			

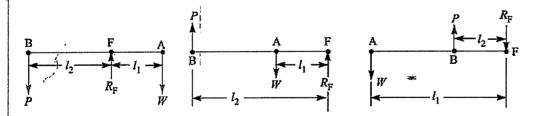


Figure 29.2 Class I, Class II and Class III Lever

Figure 29.2 shows levers of different classes.

29.3 Design of Lever

Design of lever involves determination of various dimensions of the lever for a specified load or output force required. For a specified load or output force desired, effort required can be calculated using principle of moments. Due to these forces, arms of the lever are subjected to bending and are designed based on that. Reaction force acting on the fulcrum can be calculated. Fulcrum of the lever is a pin joint and is designed based on bending and bearing considerations. Design procedure is discussed below.

ſ

If the load and effort are parallel to each other, as shown in figure 29.2, reaction on the fulcrum is the algebraic sum of these two forces. But if the load and effort are inclined to each other at an angle q, as shown in figure 29.3, reaction (R) at the fulcrum can be determined as:

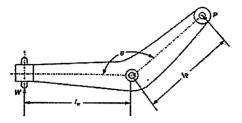


Figure 29.3 Angled Lever

$$R = \sqrt{W^2 P^2 - 2WP \cos \theta}$$

29.3.2 Design of Lever Arms

Arms are subjected to bending moment and their section is estimated from bending stress consideration. Figure 29.4 shows lever with fulcrum located between the load and the effort point. Bending moment is zero at the point of application of forces and is maximum at the fulcrum. Maximum Bending Moment is given by,

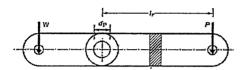


Figure 29.3 Lever Arms

Maximum Bending Stress is then given by, $M = Pl_p$

Most commonly used sections for lever arms are: rectangular, elliptical and I-section. Values of moment of inertia, I and distance of farthest fibre from neutral axis, y for these sections are given in table 29.1.

Table 29.1	Common	Sections	used fe	or Lever	Arms

Section	Rectangular	Elliptical	I-section
Shape	h	1	H 22 1 1 1 1 1 1 1 1
I	$\frac{bh^3}{12}$	$\frac{\pi a^3 b}{64}$	$\frac{BH^3 - bh^3}{12}$
у	h/2	a/2	H/2
Typical Proportions	h = (2 to 5)b	a = (2 to 3)b	H = (4 to 6)t B = (3 to 4)t

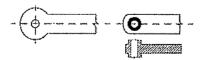


Figure 29.4 Section of Lever Arm at Fulcrum

Therefore using suitable values of I and y for selected section, its dimensions can be finalised so that the bending stress remains within the allowable limits. Often the arms are made with cross-section reducing from central portion to the point of application of load. This is done to save material using uniform strength condition. Critical section of the lever (section of maximum bending moment) becomes weak due to hole made for him. To compensate for the reduced strength, width of that section is increased or hose

Fulcrum of lever is a pin joint as shown in figure 29.5. Pin is designed based on bearing and bending considerations as discussed below.

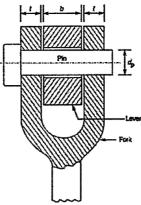


Figure 29.5 Lever Fulcrum

Bearing Failure

"The permissible bearing pressure ([P_{bearing}]) depends upon relative velocity, frequency of relative motion and the lubrication condition between the pin and the bush. The usual range of allowable bearing pressure for brass/bronze bush and steel pin is 10 25 N/mm². Lower values are used for high relative velocity, frequent motion and intermittent lubrication conditions. If d_p and l_p are diameter and length of the pin respectively, bearing pressure is given by,

$$P_{\text{bearing}} = \frac{|\text{load}|}{\text{projected area}} = \frac{R}{d_p l_p} \text{ should be } \leq [P_{\text{bearing}}]$$

Shear Failure

Pin is subjected to double shear and maximum shear stress is given by, $\tau = \frac{R}{2\left(\frac{\pi}{4}d_p^2\right)} = \frac{2R}{\pi d_p^2} \text{ should be}$ $\leq \lceil \tau \rceil$

Bending Failure

As discussed in the design of pin for knuckle joint, when the pin is loose in the eye, which is a desired condition here for relative motion, pin is subjected to bending moment. It is assumed that: Load acting on the pin is uniformly distributed in the eye and uniformly varying in the two parts of the fork. Maximum Bending Moment (at centre) is given by,

$$M = \frac{R}{2} \left(\frac{b}{2} + \frac{a}{3} \right) \left| -\frac{R}{2} \left(\frac{b}{4} \right) \right| = \frac{R}{2} \left(\frac{b}{4} + \frac{a}{3} \right)$$

Maximum Bending Stress in the pin, $\sigma_b = \frac{My}{I}$ should be $\leq [\sigma]$

$$I = \frac{\pi d_p^4}{64} \text{ and } y = \frac{d_p}{2}$$

29.4 Lever Material & Factor of Safety

Levers are generally forged or cast. It is difficult to forge curved levers with complicated cross-sections and have to be cast. As the levers are subjected to tensile stress due to bending, cast iron is not recommended to be used as material for levers. Aluminium alloys are generally used for levers. For severe loading and corrosive conditions, alloy steels are used. Suitable heat treatment processes are also often employed to improve wear and shock resistance of lever. Factor of safety of 2 to 3 on yield strength is generally used. For severe loading conditions or fatigue loading higher factor of safety is also taken.

- Design of Machine Elements by C.S. Sharma & K. Purohit
 Machine Design by P.C. Sharma & D.K. Aggarwal
 Machine Design by R.S. Khurmi