M.V.S.R. ENGINEERING COLLEGE

Department of Mechanical Engineering

Nadergul, Hyderabad -501510.

QUESTION BANK

Machine Design

(Code: ME 351)

B.E. (Mechanical Engineering) III year-I Semester

Prepared by

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Assistant Professor

DPARTMENT OF MECHANICAL ENGINEERING

CONTENTS

ME 351

MACHINE DESIGN

Instruction	4	Periods per week
Duration of University Examination	3	Hours
University Examination	75	Marks
Sessional	25	Marks

UNIT-I

Mechanical Springs: Types of springs and materials used. Design of helical springs on stress, deflection and energy considerations. Design for fluctuating loads. Concentric springs. Leaf Springs: Stresses and Deflection. Nipping of Leaf springs.

UNIT-II

Gears: Types of gears and materials used. Standards for gear specifications. Design of Spur, Helical, Bevel and Worm Gears - Strength and Wear considerations. Types of failure of gear tooth and preventive measures.

UNIT-III

Bearings: Materials used for Bearings. Classification of Bearings. Viscosity of Lubricants. Theory of Hydrostatic and Hydrodynamic lubrication. Design of sliding contact bearings - for axial and thrust loads.

Rolling Contact Bearings: Different types of rolling element bearings and their constructional details. Static and Dynamic load carrying capacity, Load-life relationship. Design for cyclic loads.

UNIT-IV

I.C. Engine Parts: Design of piston, connecting rod and crank shafts (single throw and overhang). Design of Flywheels for I.C. Engines and presses.

UNIT-V

Theory of bending: Theory of bending of members with initial curvature - rectangular, circular and Trapezoidal sections. Design of crane Hooks, Machine flames and C-clamps.

Suggested Reading:

1. M.F. Spotts, "*Design of Machine Elements*", Pearson Edu, 7th Edn. 2003.
2. V. B. Bhandari, "*Machine Design*", Tata McGraw-Hill Publ, 2010.
3. P.C.Sharma & D.K. Aggarwal, "*Machine Design*", S.K. Kataria & Sons, 10th Edn, 2003.
4. P. Kannaiah, "*Machine Design*", Sci- Tech Publ., 2009.
5. J.E. Shigley & Charles R. Mischke, " *Mechanical Engineering Design*", Tata McGraw-Hill., 6th ed. 2003.

Department of Mechanical Engineering

Vision :

To provide educational opportunities that will prepare students for productive careers as competent professionals in Mechanical Engineering, and for higher studies and research.

Mission:

The department strives to provide the engineering foundation as well as professional, innovative and leadership skills to the students through the following activities:

1. Laying sound foundation in the areas of mechanics, design, thermal sciences and production processes, as well as allied engineering areas.
2. Enrich the undergraduate experience through experimental learning, and fostering a personalized and supportive environment that makes learning joyful and stimulating
3. Provide opportunities to design mechanical engineering components and systems to meet specific needs through select courses
4. Provide opportunities to develop good communication skills, and to encourage creativity and entrepreneurial skills
5. Create awareness in professional responsibility, ethics, global impact of engineering solutions, and of the need for life-long learning.
6. Providing opportunities for training in the latest automotive technologies and encourage product development.
7. Providing research and intellectual resources to address contemporary and complex problems of industry and to advance research and applications.

Course Outcomes:

At the end of this course, Students would be able to

S.No.	Course outcome	PO's mapped
ME351.1	Classify different types of springs and their applications, and how to analyze the springs for static and fluctuating loads according to working environment.	PO1, PO2, PO3, PO7
ME351.2	Distinguish different types of gears and materials used for making gears, and how to design spur, helical, bevel and worm gears under strength and wear considerations.	PO1, PO2, PO3, PO4, PO5, PO6
ME351.3	Know different types of tooth failures with their remedial measures and estimate complete design of suitable gear drive based on the application.	PO1, PO2, PO3, PO5, PO6, PSO1
ME351.4	Undesrstand the principle of hydrostatic lubrication and hydrodynamic lubrication, and estimate the load carrying capacity of bearings for axial and thrust loads, also correlate load –life relationship for static and cyclic loads.	PO1, PO2, PO3, PO4, PO7
ME351.5	List out the various components and materials used in IC Engine. Also, various types of pistons, crank shafts and flywheels and how to design those components under mechanical and thermal loads.	PO1, PO2, PO3, PO4, PO7, PSO2
ME351.6	Compare and contrast curvature bending and straight bending. And, know the values of radius of curvature of neutral axis and centroidal axis for various commonly used crosssections in curved beams. Also, understand design procedure for the crane hooks, C-clamp and machine frames.	PO1, PO2 PO3,

Program Outcomes:

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 the 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.

PSO's (Program Specific Outcome):

1. ***Research Potential***: Usage of advanced software packages commonly used in industry for modelling, assembly and to carry out Multiphysics analysis.
2. ***Competent areas***: Design and build components and systems related to mechanical and allied disciplines, using various manufacturing methods.

Unit-I
Mechanical Springs

Part-A

1) Classify different types of springs. Give one example for each.
2) What is the function of spring, what are the different materials used for manufacturing of springs and mention their applications?
3) Briefly discuss the importance of A.M Wahl's factor in the design of helical springs.
4) What is shot peening? Discuss its role in improving the fatigue strength of a spring Wire.
5) What is concentric spring? Enumerate the advantages.
6) How to avoid buckling in compression springs?
7) Define the terms spring index, stress factor, spring rate, active number of coils and free length in springs.
8) Explain in detail stresses induced in helical compression and tension springs?
9) Differentiate between closed coil and open coiled helical spring.
10) What properties of spring materials should have?
11) Wat is clash allowance in compression springs?
12) What is the effect of pre loading in tension springs?
13) When the use of non-circular section wire springs is recommended? Why this section of the wire for springs is not generally used?
14) What is surge in springs? How surge in springs is eliminated?
15) What is nipping in leaf spring? Explain with neat sketch.
16) Explain the utility of center bolt, U-clamp, rebound clip and camber in a leaf spring.
17) Differentiate between semi elliptical and elliptical leaf springs?
18) What is the stiffness of the connection when two springs of stiffness K_1 and K_2 are connected in series, and in parallel?

Part-B

19) A vertical spring loaded valve is required for compressed air receiver. The valve is to start opening at a pressure of 1 N/mm^2 gauge and must be fully open with a lift of 4mm at a pressure of 1.2 N/mm^2 gauge. The diameter of the port is 35mm. Assume the allowable shear stress in steel as 480 MPa and shear modulus as 80kN/mm^2. Design a suitable closed coiled round section helical spring having squared and ground ends. Also specify initial compression and free length of the spring. **(May/June 2017 O)**
20) A bumper consisting of two helical steel springs of circular in cross section brings to rest, a rail wagon of mass 1500 kg is moving with a velocity of 1.2m/s. while doing so, the springs are compressed by 150mm, the mean diameter of coils is 6 times the wire diameter. The permissible shear stress is 400 MPa. Determine: (a) Max. Force on each spring (b) wire diameter of the spring (c) Mean diameter of coils and (d) Number of active coils. Take G = 0.84 X 10^5 MPa. **(Dec 2016)**
21) A rail wagon of mass 20 tones is moving with a velocity of 2m/s. It is brought to rest by two buffers with springs of 300 mm diameter. The maximum deflection of springs is 250mm. the allowable shear stress in the spring material is 600 MPa. Design the spring for buffers. **(May 2016) (May/June 2012)**
22) A helical compression spring is required to deflect through 25mm when the external force acting on it varies from 500 to 1000N. The spring index is 8. The spring has square and ground ends.

There should be a gap of 2mm between adjacent coils when the spring is subjected to the maximum force of 1000N. The spring is made of cold drawn steel wire with ultimate tensile strength of 1000 MPa, and permissible shear strength is 500 MPa, G=81370 MPa. Design the spring the calculate : (i) wire diameter (ii) Mean coil diameter (iii) number of active coils (iv) total number of coils (v) solid length (vi) free length. **(April / May 2013)**

23) A helical compression spring is used to absorb the shock. The initial compression of the spring is 30mm, and it is further compressed by 50mm while absorbing the shock. The spring is to absorb 250kJ of energy during the process. The spring index can be taken as 6. The spring is made of patented and cold drawn steel wire with an ultimate tensile strength of 1500 N/mm^2 and modulus of rigidity of 81370 N/mm^2. The permissible shear stress for the spring wire should be taken as 30% of the ultimate tensile strength. Design the spring and calculate. (i) Wire diameter (ii) Mean coil Diameter (iii) Number of active turns (iv) Free length (v) Pitch of turns. **(December 2013/O)**

24) A safety value of 60 mm diameter is to blow off at a pressure of 1.2 N/mm^2. It is held on its seat by a close coiled helical spring. The maximum lift of the valve is 10 mm. Design a suitable compression spring of spring index 5 and providing an initial compression of 35 mm. The maximum shear stress in the material of the wire is limited to 500 MPa. The modulus of rigidity for the spring material is 80 kN/mm^2. Calculate (a) Diameter of the spring wire (b) Mean coil diameter (c) Number of active turns and (d) Pitch of the coil. **(April/May 2013/O)**

25) A safety value of 40 mm diameter is to blow off at a pressure of 1.2 N/mm^2. It is held on its seat by a close coiled helical spring, with initial compression of 20 mm. The maximum lift of the valve is 12 mm. Design a suitable compression spring of spring index 6. The ultimate strength of the wire is 1400 MPa. The permissible shear stress is 700 MPa and G is 81370 MPa. Calculate (a) Diameter of the spring wire (b) Mean coil diameter (c) Number of active turns. **(December 2013)**

26) A spring is made source wire of 1.25 mm diameter and 750 N/mm^2 as its yield strength. For a mean diameter of 1.25 mm and 14 active coils of the spring. Determine
(a) Static load corresponding to the yield point of the material and deflection corresponding to that. Take G = 0.85×10^5 N/mm^2.
(b) Solid height, assuming that the ends are square and ground. (c) Stiffness of the spring
(d) Pitch of the wire so that the stress will not exceed the yield point. **(January 2012)**

27) A helical- compression spring of a cam - mechanism is subjected to an initial preload of 50N. The maximum operating force during the load cycle is 150N. The wire diameter is 3mm, while the mean coil diameter is 18mm. The spring is made of oil-hardened and tempered valve spring wire of grade VW (Sut= 1430 N/ mm^2). Determine the factor of safety used in the design. **(June 2010)**

28) A truck with a weight of 20 kN is moving with a velocity of 1.5 m/s. it is brought to rest by a bumper consisting of two parallel helical springs with spring index as 6. The springs are brought to rest with a compression of 22cm. find the suitable diameter of spring wire, the mean coil diameter and the active number coils. Take permissible stress as 36 kN/cm^2. And G = 8×10^4 MPa service factor may be taken as 1.2. **(January 2010 O)**

29) A helical compression spring made of oil tampered carbon steel, is subjected to a load which varies form 400 N to 1000 N. the spring index is 6 and the design factor of safety is 1.25. If the yield stress in shear is 770 MPa and endurance stress in shear is 350 MPa, find (a) size of the spring wire (b) Diameter of the spring (c) Free length of the spring. The compression of the spring

at the maximum load is 30mm. the modulus of rigidity for the spring material may be taken as 80 kN/mm^2. **(December 2009)**

30) A helical valve spring is to be designed for an operating load range of 90 N to 140 N. The 90 N load acts when the valve is closed and 140 N force acts when the valve is open. The deflection of the spring is limited to 8 mm. Take G=84 GPa.
31) A helical compression spring is made of oil tempered carbon steel and is subjected to a load varying from 600 N to 1000 N. The spring index is 6 and the factor of safety is 1.5. If the yield stress & endurance stress of the material is 700 MPa, and 350 MPa respectively. Find the wire diameter of the spring.
32) It is required to design a helical compression spring for the valve mechanism. The axial force acting on the spring is 300N when the valve is open and 150N when the valve is closed. The length of the spring is 30mm when the valve is open and 35mmwhen the valve is closed. The spring is made of oil hardened and tempered valve spring wire and the ultimate tensile strength is $1370N/mm^2$. The permissible shear stress for spring wire should be taken as30% of the ultimate tensile strength. The modulus of rigidity is 81370N/mm2. The spring is to be fitted over a valve rod and the minimum inside diameter of the spring should be 20mm. Design the spring and Calculate i. Wire diameter; ii. Mean coil diameter; iii. Number of active coils; iv. Total number of coils; v. Free length of the spring; and vi. Pitch of the coil. Assume that the clearance between adjacent coils or clash allowance is 15% of the deflection under the maximum load.
33) A semi elliptic leaf spring 900mm long and 55mm wide is held together at the center by a band of 50mm wide. If the thickness of each leaf is 5mm, find the number of leaves required to carry a load of 4500 N. Assume a maximum working stress of 490 MPa. If the two of these leaves extend the full length of the spring, find the deflection of the spring. The young's modulus of the spring material ay be taken as 210 kN/mm^2. **(June 2017)**
34) A semi elliptic leaf spring used for automobile suspension consists of three extra full-length leaves and 15 graduated leaves including the master leaf. The centre to centre distance between two eyes of the spring is 1.5m. The maximum force acting on spring is 100 kN. For each leaf the ratio of width to thickness is 9:1. E=200 GPa, the leaves are pre-stressed in such a way that when the force is maximum, the stresses induced in all leaves are same and equal to 450MPa. Determine (i) the width and thickness of leaves, (ii) the initial nip; and (iii) the initial preload required to close the gap C between extra full-length leaves and graduated length leaves.

(Dec 2017/O) (April / May 2013)

35) A Semielliptical laminated spring is made of 5mm thick steel plate 50mm wide. The length between the supports is 665mm and the band is 65mm wide. The spring has two full length leaves and five graduated leaves. A central band of 1600N is applied. Determine
    (a) The maximum stress in each set of leaves for an initial condition of no stress in the leaves.
    (b) The maximum stress if initial stress is provided to cause equal stresses when loaded.
    (c) The deflection in above (a) and (b). **(May/June 2015) (Jan 2015)**
36) The spring of a small truck is to hold maximum load of 4000 N each and to have a deflection rate 50 N/mm. The engine develops a maximum torsional moment of 300 N-m. Where the rear axle ratio is 3:1 and the 4 ply tyres are sized 170x760 mm. The co efficient of friction between the tyre and ground is 0.6. The span length of the spring may be taken as 1.5 times the tyre diameter. Design the leaf spring and make a neat sketch showing all necessary dimensions. Tabulate a satisfactory combination of (a) thickness of leaf and width (b) number of leaves (c) Camber in free

position (d) Radius of curvature (e) Material used and heat treatment (f) Length of each leaf, and deflection. **(April/May 2014)**

37) Design a leaf spring for the following specification:
Total load = 140kN; Number of springs supporting the load = 4; maximum number of leaves = 10; span of the spring = 1000 mm; permissible deflections = 80mm, take young's modulus E = 200 kN/mm^2 and allowable stress in the spring material as 600 MPa. **(May/June 2012)**

38) A semi elliptical truck spring has 12 leaves, of which two are full length leaves. The spring supports are 0.7 m apart the width of the central band is 80mm, and the load on the spring is 20kN. The permissible stress is 460 MPa. The ratio of the total depth to width of the spring is 3. Determine the thickness and width of the spring leaves. Also, determine the deflection of the spring. Assume that the extra full length leaf is not pre stresses. Take E = 2.1×10^5 N/mm^2. **(December 2012 O)**

39) A semi elliptical laminated spring is to carry a load of 5000 N and consists 8 leaves 46 mm wide, two of the leaves being of full length. The spring is to be made 1000 mm between the eyes and is held at the centre by a 60mm wide band. Assume that the spring is initially stressed so as to induce an equal stress of 500 N/mm^2 when fully loaded. Design the spring giving (a) Thickness of leaves (b) Eye diameter (c) Length of leaves (d) Maximum deflection and camber. Assume E = 2.1 x 10^6 N/mm^2. **(January 2012)**

40) A semi elliptical leaf spring consists of two extra full-length leaves and eight graduated leaves including the master leaf. The center to center distance between the two eyes of the spring is 1 m. if the maximum force acting on the spring is 10kN and the width of the leaf is 50mm. the spring is initially preloaded in such a way that when the load is maximum, the stress induced in the leaves are equal to 350 N/mm^2. The modulus of elasticity of the leaf material is 2.07 x 10^5 N/mm^2 determine (a) the thickness of the leaves (b) the deflection of the spring at maximum load (c) initial nip (d) initial pre load required to close the nip. **(May/June 2011)**

41) A truck spring has 12 number of leaves, two of which are full length leaves. The spring supports are 1.05m apart and the central band is 85 mm wide. The central load is to be 5.4 KN with a permissible stress of 280 MPa. Determine the thickness and width of the steel spring leaves. The ratio of the total depth to the width of the spring is 3. Also determine the deflection of the spring. **(June 2010)**

42) Design a leaf spring for the following specifications: Total load = 140kN; No. of springs supporting the load = 4; Maximum number of leaves = 10; Span of the spring = 1000 mm; Permissible deflection = 80mm; Take E = 200 kN/mm^2 and allowable stress in the spring material as 600 MPa. **(December 2009)**

Unit-II
Gears

Part-A

1) Classify main four types of gears and discuss their applications.
2) Classify the gears based on the shape and relative positions of the shafts.
3) What is law of gearing?
4) Define the terms (i) Pitch circle (ii) Pressure angle (iii) Backlash (iv) Module (v) Addendum (vi) Dedendum (vii) Pitch in spur gears.
5) Distinguish between spur and helical gears.
6) What are the various types of failures in gear tooth?
7) List various types of materials and specify yield and ultimate strength.
8) Hat are the design considerations of gear drive?
9) Wat are preventive measures to avoid gear tooth failure.
10) What is the Lewis equation for strength of gear teeth?
11) Differentiate between cycloidal and involute tooth profiles used in gears.
12) Sketch the helical gear and shows the forces acting on the gear and write the wear load and dynamic tooth load.
13) Sketch the spur and bevel gears and show the forces and their analysis.
14) Sketch the worm gear and shows the forces acting on the gear and write the wear load and dynamic load capacity of the worm drive.
15) Write expressions for static and limiting wear loads for a helical gear.
16) Compare the beam strength of spur and helical gears.
17) What is herringbone gear? Where they are used, explain with neat sketch.
18) What are the limits on helix angle of helical gears?
19) What is interference in gears?
20) State two important reasons for adopting involute curve for gear tooth profile.
21) What is crown gear?
22) What is formative or virtual or equivalent number of teeth in case of gears?
23) What is *Tredgold's* approximation in bevel gears?
24) Explain briefly gear tooth modifications.
25) In a worm gear drive only wheel is designed, why?
26) What are the advantages of worm gear drives over other gear drives?
27) With the help of neat sketch define the following related to bevel gears. (i) Cone Center (ii) Pitch angle.

Part-B

28) A pair of 20° full depth involute profile spur gears are to transmit 24kW at a speed of 300 rpm of the pinion. The velocity ratio is 2:1. The pinion is made of cast steel having an allowable safe static stress σ = 120 MPa. While the gear is made of cast iron having allowable safe static stress σ = 60 MPa. The pinion has 18 teeth and its face width is 10 times the module. Determine the module, face width and pitch diameters of both pinion and gear from standard point of strength. Check

the design for beam strength and wear strength. Lewis form factor y = 0.154- (0.912/No. of Teeth), Velocity factor C_v = 3/(3+V), where V is peripheral velocity in m/sec. **(December 2017 O)**

29) A bronze spur pinion rotating at 600 r.p.m. drives a cast iron spur gear at a transmission ratio of 4: 1. The allowable static stresses for the bronze pinion and cast iron gear are 84 MPa and 105 MPa respectively. The pinion has 16 standard 20° full depth involute teeth of module 8 mm. The face width of both the gears is 90 mm. Find the power that can be transmitted from the standpoint of strength. **(May/June 2017) (May 2012)**
30) A pair of 20° Involute straight tooth spur gears to transmit 50kW and reduce the speed from 720 rpm to 180 rpm. The pinion and gear are made from phosphor bronze and cast steel with allowable static stresses 50 N/mm² and 70 N/mm² respectively. Assuming medium shock conditions design drive completely. **(May 2016) (December 2010) (May 2011)**
31) A pair of spur gears consists of 20 teeth pinion meshing with a 100 teeth gear. The pinion rotates at 720 rpm, the normal pressure angle is 20^0, the face width is 40mm and the module is 4mm. the pinion as well as gear is made of steel having ultimate strength of 600MPa, and heat treated to a surface hardness of 300 BHN. Taking factor of safety as 2.5, and assuming that the velocity factor accounts for the dynamic load, calculate the power transmitting capacity of the gears. **(December 2015) (May 2014)**
32) A steel pinion with 20^0 full depth involute teeth is transmitting 7.5 kW power at 100 rpm from an electric motor. The starting torque of the motor is twice the rated torque. Assuming that velocity factor accounts for the dynamic, calculate (i) Effective load on gear teeth; and (ii) The bending stresses in gear tooth. **(May 2013)**
33) A reciprocating compressor is to be connected to an electric motor with the help of spur gears. The distance between the shafts is to be 500 mm. The speed of the electric motor is 900 r.p.m. and the speed of the compressor shaft is desired to be 200 r.p.m. The torque, to be transmitted is 5000 N-m. Taking starting torque as 25% more than the normal torque, determine: 1. Module and face width of the gears using 20 degrees stub teeth, and 2. Number of teeth and pitch circle diameter of each gear. Assume suitable values of velocity factor and Lewis factor. **(May 2013 O)**
34) The following particulars refers to as spur gear drive; Central distance = 200mm; Velocity ratio = 4; Power = 50kW; Pinion speed = 1440 rpm; Tooth profile = 20°; full depth involute permissible normal load between the teeth = 160 N/mm of the face width. Design the drive. Also determine the load on the bearings, stating its nature. Assume that both the gears re mounted on overhanging shafts. **(December 2012)**
35) Determine the proper pitch, Module, face, number of teeth and outside diameters of a pair of 20° involute full depth spur gears to transmit 112.5kW, from a pinion at 75 rev/min, to a gear running at 140 rev/min. the service is intermittent with light shocks. **(January 2012)**
36) It is required to design a pair of spur gears with 20° full depth involute teeth based on lewi's equation. The velocity factor is to be used to account for dynamic load. The pinion shaft is connected to a 10kW, 1440 rpm motor. The starting torque of the motor is 150% of the rated torque. The speed reduction is 4:1. The pinion as well as the gear are made of plain carbon steel 40C8 (S_{ut} = 600 N/mm²). The factor of safety can be taken as 1.5. Design the gears, specify their dimensions and suggest suitable surface hardness for the gears. **(June 2010)**
37) A motor shaft rotating at 1500 rpm has to transmit 15kW to a low speed shaft with a speed reduction 3:1. The teeth are 14 ½ ^{0} involute with 25 teeth on the pinion. Both the pinion and gear are made of steel with maximum safe stress of 200MPa. A safe stress of 40MPa may be taken for

the shaft on which the gear is mounted and for the key. Design a spur gear drive to suit the above conditions. Assume starting torque to be 25% higher than running torque. **(January 2010 O)**

38) A compressor running at 300 rev/min is driven by 15kW, 1200 rev/min motor through a 14 ½ ^{0} full depth gears. The center distance is 0.375m, the motor pinion is to be of C30 forged steel hardened and temper, and the driven gear is to be cast steel. Assuming medium shock condition; (a) Determine module, the face width, and number of teeth on each gear. (b) Design the drive completely. **(May 2009)**

39) 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. **(December 2009)**

40) A 15 kW and 1200 r.p.m. motor drives a compressor at 300 r.p.m. through a pair of spur gears having 20° stub teeth. The centre to centre distance between the shafts is 400 mm. The motor pinion is made of forged steel having an allowable static stress as 210 MPa, while the gear is made of cast steel having allowable static stress as 140 MPa. Assuming that the drive operates 8 to 10 hours per day under light shock conditions, find from the standpoint of strength. 1. Module; 2. Face width and 3. Number of teeth and pitch circle diameter of each gear. **(December 2008)**

41) A pair of straight teeth spur gears, having 14 ½ ^{0} involute full depth teeth is to transmit 12 kW at 300 r.p.m. of the pinion. The speed ratio is 3:1. The allowable static stresses for gear of cast iron and pinion of steel are 60 MPa and 105 MPa respectively. Determine the module, face width and pitch circle diameters of the gears. Also check the gears for wear. Use the following Particulars: Number of teeth of pinion = 16; Face width = 10 times module, σ_{es} = 600 MPa; E_P = 200 kN/mm^2 and E_G = 100 kN/mm^2. Velocity factor $C_v = \frac{4.5}{4.5+V'}$. **(December 2008)**

42) A pair of helical gears 30^0 helix angle is used to transmit 15kW at 10,000rpm of the pinion. The velocity ratio is 4:1, both the gears are to be made of hardened steel of static strength 100 N/mm^2. The gears are 20° stub and pinion is to have 24 teeth. The face width may be taken as 14 times the module, find the module and ace width from standard point of strength. **(December 2017)**

43) A pair of helical gears consist of 18 teeth pinion meshing with a 45 teeth gear. An electric motor of 60kW running of 2000 rpm is supplying power to the pinion. The helix angle is 23° and the normal pressure angle is 20°. Determine the tangential, radial and axial loads between the meshing teeth's of the module is 8mm in the normal plane to the teeth.
(December 2016) (December 2010) (May 2011)

44) A pair of helical gears consist of a 20 teeth pinion meshing with a 100 teeth gear. The pinion rotates at 720 r.p.m. The normal pressure angle is 20° while the helix angle is 25°. The face width is 40 mm and the normal module is 4 mm. The pinion as well as gear are made of steel having ultimate strength of 600 MPa and heat treated to a surface hardness of 300 B.H.N. The service factor and factor of safety are 1.5 and 2 respectively. Assume that the velocity factor accounts for the dynamic load and calculate the power transmitting capacity of the gears.
(January 2015) (June 2015) (December 2013)

45) A pair of helical gears are to transmit 15 kW. The teeth are 20° stub in diametral plane and have a helix angle of 45°. The pinion runs at 10 000 r.p.m. and has 80 mm pitch diameter. The gear has

320 mm pitch diameter. If the gears are made of cast steel having allowable static strength of 100 MPa; determine a suitable module and face width from static strength considerations and check the gears for wear, given σ_{es} = 618 MPa. **(May 2013 O)** (**December 2009)**

46) Design a pair of equal diameter, 20° stub tooth helical gears to transmit 37.5kW with moderate shock at 1200 rev/min. The two shaft are parallel and 0.45m apart. Each gear is to be of steel. Find the module and face width of the teeth. **(January 2012)**

47) A pair of straight tooth bevel gears transmits 15kW at 1250 rpm of 120mm diameter case hardened steel pinion of 350 MPa, to a heat treated cast steel gear of 190MPa at a speed ratio of 3.5. use 14 ½ involute tooth system. The angle between shaft axes is 90^0 take service factor as 1.25. design the gears and suggest suitable hardness for the gears. **(December 2017 O)**

48) A pair of straight bevel gears consists of 30 teeth pinion meshing with a 45 teeth gear. The module and the face width are 60mm and 50mm respectively. The pinion as well as the gear is made of steel (S_{ut} = 600 N/mm^2). Calculate the beam strength of the tooth. **(December 2013)**

49) A pair of straight bevel gears consists of 24 teeth pinion meshing with a 48 teeth gear. The module at the outside diameter is 6mm, while face width is 50mm. the gears are made of grey cast iron from FG220 (S_{ut} = 220N/mm^2), pressure angle is 20°. The teeth's are generated and assumed that the velocity factor accounts for the dynamic load. The pinion rotates at 300 rpm and the service factor is 1.5, calculate (i) The beam strength of the tooth, (ii) The static load that the gears can, transmit with a factor of safety of 2 for bending consideration and (iii) Rated power that the gears can transmits. **(May 2013)**

50) A pair of straight bevel gears has a velocity ratio of 2:1. The pitch circle diameter of the pinion is 80mm at the large end of the tooth. A 5kW power is supplied to the pinion, which rotates at 800 rpm. The face width is 40mm and the pressure angle is 20°. Calculate the tangential, radial and axial components of the resultant tooth force acting on the pinion. **(June 2010)**

51) Design a bevel gear drive between two shafts whose axes are at right angles. Speed of the pinion shaft is 200 rev/min and that of the gear shaft is 120 rev/min. pinion is to have 21 teeth of involute profile with module of 20mm and pressure angle of 20° ad is to be of suitable material. Gear is made of cast iron. Power at gear shaft is 75kW. **(April 2009)**

52) A pair of bevel gears is required to transmit 12 kW at 500 r.p.m. from the motor shaft to another shaft, the speed reduction being 4:1. The shafts are inclined at 60°. The pinion is to have 24 teeth with a pressure angle of 20° and is to be made of cast steel having a static stress of 80 MPa. The gear is to be made of cast iron with a static stress of 55 MPa. The pinion is mounted mid-way on the shaft which is supported between two bearings having span of 200mm. design the gear pair. **(January 2010 O)**

53) A cast iron bevel gear has a module of 2.5mm and its pitch diameter is 0.6m. And the pitch angle is 30° and the teeth are 20° full depth. Determine the permissible endurance load. **(May 2015 DAC)**

54) A worm and gear speed reducer to transmit 22kW at a speed of 1440 rpm. The desired velocity ratio is 24:1. An efficiency of at least 85% is desired. Assume that the worm is made of hardened steel and the gear of phosphor bronze. **(June 2017)**

55) A worm gear drive transmits 15 kW to a machine. The worm speed and the gear speeds are 2000 rpm and 50 rpm respectively. The worm is triple threaded and has a pitch diameter of 65mm. The gear has 120 teeth of 6mm module. The tooth form is 20^0 full depth involute and co-efficient of

friction = 0.1. find (i) Tangential force acting on the worm (ii) Axial thrust acting on the worm (iii) Separating force on the worm (iv) Efficiency of the worm. **(May/June 2017 O)**

56) A worm drive transmits 15 kW at 2000 r.p.m. to a machine carriage at 75 r.p.m. The worm is triple threaded and has 65 mm pitch diameter. The worm gear has 90 teeth of 6 mm module. The tooth form is to be 20° full depth involute. The coefficient of friction between the mating teeth may be taken as 0.10. Calculate: (i) Tangential force acting on the worm; (ii) Axial thrust and separating force on worm; and (iii) Efficiency of the worm drive. **(May 2012)**

57) Design a worm gear drive for an input power of 1kW, with a transmission ratio of 2.5. The worm speed is 1600 rpm. The worm is made of hardened steel, and the gear of phosphorous bronze, for which the material combination factor is 0.7 N/mm2. The static strength of phosphorous bronze is 56MPa. The worm is of double start type, and the center distance of the drive is 120mm, the tooth form is 20° involute. Check the design for strength wear and heat dissipation. **(December 2015)**

58) A pair of worm and worm wheel is designated as 3/60/10/6. The worm is transmitting 5kW power at 440 rpm to the worm wheel. The co efficient of the friction is 0.1. And the normal pressure angle is 20°. Determine the components of the gear tooth force acting on the worm and worm wheel. **(December 2013)**

Unit-III
Bearings

Part-A

1) What is a bearing, how bearings are classified?
2) What are the four parts of a ball bearing?
3) Explain any method of bearing mounting?
4) Explain the details of the SKF designated bearing 6108.
5) Why the radial load carrying capacity of roller bearing is more than ball bearing?
6) Why the rigidity of roller bearing is more than ball bearing?
7) Why roller bearing generates more noise than ball bearing?
8) What types of bearing are used in applications where misalignment is likely to be present?
9) Why taper roller bearing takes thrust and radial load?
10) State any two advantages of thrust ball bearing.
11) What are the materials used for rolling contact bearings?
12) Suggest any two types of bearing to take up combined heavy axial and heavy radial loads.
13) What is an antifriction bearing?
14) Define static load carrying capacity and dynamic load carrying capacity of the bearing.
15) Name three factors on which static load carrying capacity depends.
16) What do you mean by failure of a rolling contact bearing? And State the types of failures in rolling contact bearings.
17) Write the expression for frictional torque transmitted in conical pivot bearing.
18) What is the life of an individual ball bearing?
19) What is median or average life of rolling contact bearings?
20) What is equivalent dynamic load in rolling contact bearing?
21) Write down the relationship between dynamic load carrying capacity, the equivalent dynamic load and bearing life.
22) What is the relationship between L50 and L10 life?
23) Enumerate the steps for selection of rolling contact bearings for a particular application.
24) How are rolling contact bearings designated?
25) State any two advantages of using oil as lubricant compared with grease for rolling contact bearings.
26) Why the coefficient of friction in needle roller bearing is higher than roller bearing?
27) Define lubrication. Name any two liquid lubricants, semi-solid lubricant. And any two solid lubricants.
28) Differentiate hydro static and hydro dynamic lubrication.
29) Enumerate the factors that form and maintain thick oil film in hydrodynamic journal bearings?
30) Draw a sketch showing the pressure distribution around the periphery of a hydrodynamic journal bearing.
31) What does journal-bearing mean?
32) What do you mean by self-contained bearing?
33) Differentiate between footstep bearing and collar bearing.
34) What is elasto hydrodynamic lubrication?

35) Distinguish between full and partial bearings. What is the preferred angle of contact for partial journal bearings?
36) What will happen when the viscosity of the lubricant is very low in a bearing?
37) What is the meaning of viscosity in Saybolt Universal Seconds (SUS)?
38) State the relationship between kinematic viscosity, absolute viscosity and density of lubricant.
39) What are the two assumptions of Petroff's equation?
40) State the two dimensionless performance parameters in Petroff's equation that govern the coefficient of friction.
41) State four important assumptions of Reynold's equation.
42) What are the two Sommerfeld's solutions of Reynold's equation?
43) What do you mean by Bearing characteristic number and bearing modulus?
44) Draw bearing characteristic curve and indicate the various regions.
45) Define eccentricity ratio in hydrodynamic journal bearing.
46) What are the advantages and disadvantages of short bearings over long bearings?
47) Define bearing pressure.
48) Give a list of materials for sliding contact bearing. What are the desired properties of a bearing material?
49) What is babbitt? Why babbitt is called white metal? Give a typical composition of tin based Babbitt and lead based babbitt.
50) What are sintered metal bearings? What are their two varieties?
51) Compare iron-base and copper-base sintered metal bearings.
52) What are the commonly used non-metallic bearing materials?
53) What is the approximate relationship between SAE number and viscosity of lubricating oil?
54) State any four advantages and disadvantages of mineral oil and vegetable oil.
55) State the types of bearing failure?

<u>Part-B</u>

56) Design a journal bearing for a centrifugal pump. Operating conditions are as follows: Load on the journal = 11.5 kN; Speed of the journal =1440 r.p.m.; Diameter if the journal is 75mm, oil film temperature is 70^0C, Ambient temperature of oil = 22°C; **(May/June2017 O)**
57) Design a journal bearing for a centrifugal pump running at 1440 rpm. The diameter of the journal is 100mm and the load on each bearing is 20kN. The factor ZN/P may be taken as 28 for centrifugal pump bearings. The bearing running at 75^0C temperature and the atmospheric temperature is 30^0C. the energy dissipation coefficient is 875 W/m2/^{0}C. Take diametral clearance as 0.1mm. **(December 2017)**
58) A bearing 50 mm diameter and 75 mm in length supports an overhanging shaft running at 900 rpm. The room temperature is 30^0C and the oil film temperature is 75^0C. The viscosity of the oil used is 0.012 kg/m-s at the operating temperature of 120^0C. The diameter clearance is 0.05 mm and the bearing is to operate in still air without any artificial cooling. Calculate the permissible load on the bearing and power loss in friction. The heat dissipation coefficient may be assumed as 300 W/m^2.^{0}C **(December 2016)**
59) A bearing 50 mm diameter and 75 mm in length supports an overhanging shaft running at 900 rpm. The room temperature is 30^0C and the oil film temperature is 75^0C. The viscosity of the oil

used is 0.012 kg/m-s, the diameter clearance is 0.05 mm and the bearing is to operate in still air without any artificial cooling. Determine (a) the permissible load on the bearing (b) power loss. **(December 2017 O)**

60) A full journal bearing of 50 mm diameter and 100 mm long has a bearing pressure of 1.5 N/mm^2. The speed of the journal is 1000 r.p.m. and 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 0.011 kg/m-s. The room temperature is 35°C. Find: (i) The amount of artificial cooling required, and (ii) The mass of the lubricating oil required, if the difference between the outlet and inlet temperature of the oil is 12°C. Take specific heat of the oil as 1900 J/ kg/°C. **(May 2016) (January 2015) (May/June2015) (December 2010)**

61) A 150 mm diameter shaft supporting a load of 10 kN has a speed of 1500 r.p.m. The shaft runs in a bearing whose length is 1.5 times the shaft diameter. If the diametral clearance of the bearing is 0.15 mm and the absolute viscosity of the oil at the operating temperature is 0.011 kg/m-s, find the power wasted in friction. **(April/May2013)**

62) A 80 mm long journal bearing supports a load of 2800 N on a 50 mm diameter shaft. The bearing has a radial clearance of 0.05 mm and the viscosity of the oil is 0.021 kg/m-s at the operating temperature. If the bearing is capable of dissipating 80 J/s, determine the maximum safe speed. **(April/May2013)**

63) The following data is given for a 3600 Hydrodynamic bearing: radial load = 10kN, Journal speed = 1440 rpm, unit bearing pressure = 1000kPa, clearance ratio (r/c) = 800, viscosity of lubricant = 30mPaS. Assuming the total heat generated in the bearing is carried by the total oil flow in the bearing. Calculate (i) dimensions of the bearing (ii) Coefficient of friction (iii) Power lost in friction (iv) Total flow of oil (v) Side leakage (vi) Temperature rise. **(December 2013)**

64) The following data is given for a Hydrostatic thrust bearing: Thrust load = 500kN; Shaft speed = 720 rpm; shaft diameter = 500mm; Recess diameter = 300mm; Film thickness = 0.15mm; Viscosity of the lubricant = 160 SUS; Specific gravity = 0.86; Calculate: (i) Supply pressure (ii) Flow requirement in Lit/min (iii) Power loss in pumping (iv) Frictional power loss. **(December 2013 O)**

65) A journal bearing, 100mm in diameter and 150mm long; carries a radial load of 7kN at 1200 rpm. The diametral clearance is 0.075mm. Find the viscosity of the oil being used at the operating temperature; if 1.2kW power is wasted in friction. **(December 2012)**

66) Design a journal bearing for a centrifugal pump from the following data: Load on the journal = 20000 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 / mm^2. 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/m^2/°C. **(May/June2012)**

67) A journal bearing of 50 mm diameter and 80 mm long has a bearing pressure of 6MPa. The speed of the journal is 1000 r.p.m. and the ratio of journal diameter to the diametral clearance is 800. 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. Find: (i) amount of heat generated (ii) Amount of heat dissipated through the bearing. The specific heat of the oil as 1900 J/kg /°C.Heat dissipation Coefficient = 490 W/m^2/°C **(May/June2011)**

68) A full journal bearing of diameter 0.075 m and a length of 0.125 m is to support a load of 20 x 10^3 N at the shaft speed of 1000 rev/min. the bearing temperature is limited to 77 ^{0}C in a room

temperature of 38 ^{0}C. The viscosity of the oil used is 0.0098 kg/m-s at 116 ^{0}C. Find the amount of artificial cooling required, by means of external oil cooler. **(April/May 2009)**

69) Following data are given for a 360^{o} 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: (i) Power lost in friction; (ii) The coefficient of friction; (iii) Minimum oil film thickness (iv) Flow requirement in 1/min; and (v) Temperature rise.

70) 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 ^{o}C and the ambient temperature is 32 ^{o}C. Select a suitable oil to suit the above conditions.

71) The rolling contact bearings are to be selected to support the overhang counter shaft. The shaft speed is 720rpm. The bearings are to have 99% reliability corresponding to a life of 24,000 hrs. the bearing is subjected to an equivalent radial load of 1kN. Consider life adjustment factors for operating condition and material as 0.9 and 0.85 respectively. Find the basic dynamic load rating of the bearing from manufacturers catalogue, specified at 90% reliability. **(June 2017)**

72) A 30 second work cycle consists of the following two parts

	Part-I	Part-II
Duration (Seconds)	10	20
Radial Load (kN)	50	20
Axial Load (kN)	10	5
Speed (RPM)	600	1200

For this application the static and dynamic load capacities of a single row deep groove ball bearing are 45kN and 60kN respectively. Calculate the life of the bearing in hours. **(June 2017 O)**

73) A shaft is mounted on two roller bearings, which are 350mm apart. The shaft carries a bevel gear at the middle at shaft speed of 900 rpm. The gear forces are radial load of 10kN and thrust load of 3.5 kN. Determine the rated dynamic capacity of the bearing, for thrust desired life 10,000 hrs. the service factors are 1.5, thrust factor is 3.7, and radial load factor is 0.67. **(Dec 2017)**

74) The radial load acting on ball bearing is 3000N for the first 10 revolutions and reduces to 2000N for the next 15 revolutions. The load variation repeats itself. The expected life of the bearing is 30 million of revolutions. Determine the load carrying capacity of the bearing. **(Dec 2017 O)**

75) A ball bearing is operating on a work cycle consists of three parts as follows: A radial load of 3000N at 1440 rpm for one quarter cycle, a radial load of 5000N at 720 rpm for one half cycle and radial load of 2500n at 1440 rpm for the remaining cycle. The expected life of the bearing is 10000hrs. Calculate the dynamic load carrying capacity of the bearing**.** **(May 2016) (Dec 2010)**

76) A ball bearing is subjected to a radial load force of 2500 N and an axial force of 1000N. the dynamic load carrying capacity of the bearing is 7000N. the values of X and Y factors are 0.58 and 1.8 respectively. The shaft is rotating at 800 rpm calculate the life of the baring in millions of revolutions and hours. **(Dec 2016)**

77) A shaft is mounted on two roller bearings, which are 350mm apart. The shaft carries a bevel gear at the middle at a shaft speed of 900 rpm; the gear forces are radial load = 10 kN, and thrust load = 3.5kN. Determine the rated dynamic capacity of the bearing, for desired life of 10,000 hours. The service factors are 1.5, thrust factor is 3.7, and radial load factor is 0.67. **(Dec 2015) (May 2014)**

78) A ball bearing with a dynamic load capacity of 22.8kN is subjected to a radial loads of 10kN. Calculate (i) The expected life in million revolutions that the 90% of the bearings will reach (ii) The corresponding life in hours, if the shat is rotating at 1500 rpm; and (iii) The life that 50% of the bearings will complete or exceed before fatigue failure. **(April/May 2013)**

79) A ball bearing is operating on a work cycle consists of three parts as follows. A radial load of 2500N at 1200 rpm for one quarter cycle and radial load of 2000N at 1440rpm for the remaining cycle. The expected life of the bearing is 12000 hrs. Calculate the dynamic load carrying capacity of the bearing. **(May/June 2011)**

80) A ball bearing subjected to a radial load of 3000 N is expected to have a satisfactory life of 10 000 hours at 720 r.p.m. with a reliability of 95%. Calculate the dynamic load carrying capacity of the bearing, so that it can be selected from manufacturer's catalogue based on 90% reliability. If there are four such bearings each with a reliability of 95% in a system, what is the reliability of the complete system? **(June 2010)**

81) A ball bearing is required to resist a radial load of 10kN and a thrust load of 5kN. The average life of the bearing is to be 5000 hours, with inner race rotation at 980 rpm. What basic dynamic load rating must be used in selecting the bearing? If this bearing is to have a life of 5000 hours at a reliability of 97%. What is the required basic dynamic load rating? **(December 2012)**

82) Design a deep grove ball bearing to support a vertical shaft for the following data: Radial load = 750kg; Axial load = 220kg; desired life = 160×10^6 revolutions; Speed = 300 rpm. **(January 2010 O)**

83) A single row deep grove ball bearing No.6002 is subjected to an axial thrust of 1000N and a radial load of 2200N. Find the expected life that 50% of the bearings will complete under this conditions. **(December 2009)**

84) 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.

85) An electric motor has a shaft of 60mm diameter and subjected to an axial thrust of 75kN. It is desired to provide 4 collars for the bearing of this shaft. Motor speed is 1200 rpm and co efficient of friction is 0.035. Find outside diameter of each collar if pressure is not to exceed 0.7 MPa. Also find the thickness of each collar if permissible shear stress is 30MPa. **(December 2008)**

Unit-IV
IC Engine Parts

Part-A

1) What are the various parts and design considerations for an IC Engine?
2) Discuss strength design and thermal design of piston head.
3) Explain the classification of piston rings.
4) Why piston is made light weight?
5) Discuss strength design and thermal design of piston head.
6) State the function of piston rings, piston skirt and piston pin for an IC Engine piston.
7) What is the effect of side thrust on IC Engine cylinder liner?
8) What are he functions of the following parts of a piston of an IC engine (i) skirt (ii) piston rings (iii) Piston pin
9) Give the design considerations of crank shaft.
10) Mention various types of forces and stresses used in crank shafts.
11) What are the methods and materials used in manufacturing of crank shafts?
12) Sketch the crank shaft and show the forces acting in the crank shaft.
13) Describe the whipping stresses in the connecting rod.
14) Mention the various considerations in the design of connecting rod.
15) Mention various types of forces and stresses induced in the connecting rod.
16) Why I-Section is preferred for the design of connecting rod?
17) Describe the materials used and their properties for piston and connecting rod.
18) What is the function of a flywheel and mention its two applications.
19) What are design considerations for the flywheel used in IC Engines and presses?
20) What are the functions of a valve spring in an IC engine?
21) What is interference angle between valve seating surface? Why it is provided?
22) Name the possible modes of failure, to be considered for the design of (i) Piston pin and (ii) Crank Pin.
23) Why the area of the inlet valve port is made larger than the area of exhaust valve port in an IC engine.
24) Define coefficient of fluctuation speed and fluctuation of energy.
25) What is the role of bearing pressure in design of crank shaft?
26) With the help of neat sketch explain differences between overhanging and center crankshafts.
27) What are design considerations for the flywheel used in IC engines?
28) What are the functions of a valve spring in an IC engine?

Part-B

29) Design a connecting rod for a petrol engine from the following data: Diameter of the piston = 100 mm, Mass of the reciprocating parts = 2.25 kg, Length of the connecting rod = 300 mm, Stroke length = 125 mm, Speed = 1500rpm, Maximum explosion pressure = 3.5 N/mm^2. Factor of safety is 5, Density of the rod material is 8000 kg/m^3, yield stress in compression is 330 MPa. Allowable bearing pressures at this small end and big end are 12 MPa and 8 MPa respectively. **(May 2017 O)**

30) Design a connecting rod for a petrol engine from the following data: Diameter of the piston = 120 mm, Mass of the reciprocating parts = 2 kg, Length of the connecting rod from centre to centre= 300 mm, Stroke length = 140 mm, Speed = 2000rpm, Maximum explosion pressure = 2.5 N/mm^2. The allowable stress for the material is 340 N/mm^2.**(May 2016) (May/June 2011) (December 2010)**
31) Why I sections are used in connecting rods? Design I section connecting rod for a diesel engine having 160mm bore and 200mm stroke running at 1440 rpm. The maximum explosion pressure is 5.25 N/mm^2. The allowable stress for the material is 340 N/mm^2. The L/R taio is4 and factor of safety is 5. Allowable bearing pressures at this small end and big end are 10 MPa and 6 MPa and weight of the reciprocating parts is 1.1 kg. **(December 2016)**
32) Determine the dimensions of an I-section connecting rod for a petrol engine from the following data: Diameter of the piston = 110 mm, Mass of the reciprocating parts = 2 kg, Length of the connecting rod from centre to centre= 325 mm, Stroke length = 150 mm, Speed = 1500rpm with permissible over speed of 2500 rpm, Compression ratio = 4 : 1, Maximum explosion pressure = 2.5 N/mm^2. **(May/June 2015)**
33) Design the connecting rod for a petrol engine from the following data: Diameter of the piston = 110 mm, Mass of the reciprocating parts = 2 kg, Length of the connecting rod = 325 mm, Stroke length = 150 mm, Speed = 1500rpm with possible over speed of 2500 rpm, Compression ratio = 4, Maximum explosion pressure = 2.5 N/mm^2. **(January 2015)**
34) Design the cross section of connecting rod of petrol engine, from the following data: Diameter of piston = 90mm, Length of connecting rod = 300mm, Maximum explosion pressure = 2.2 N/mm^2, Factor of safety = 5, the rod of I-section, with width 4t and depth 5t where 't' is the thickness of the web and flanges. Compare the values of 't' obtained in direct compression and buckling. **(January 2015 O)**
35) The following data is given for the cap and bolts of the big end of connecting rod; Engine speed = 1500 rpm, length of connecting rod = 320mm, length of stroke = 140mm, mass of reciprocating parts = 1.75kg, length of crank pin = 54mm, Permissible bending stress for cap = 120MPa, calculate the nominal diameter of bolts and thickness of cap of the big end. **(April/May 2013)**
36) Design a connecting rod for a high speed IC Engine using the following data: Cylinder bore = 125 mm, Length of the connecting rod = 300mm, Maximum explosion pressure = 3.5 MPa, length of Stroke = 125 mm, Mass of the reciprocating parts = 1.6 kg, Engine Speed = 2200rpm, Assume suitable data and state assumptions you made. **(December 2013 O)**
37) Design a connecting rod for a single cylinder four stroke diesel engine with following specifications: power = 7.5kW, Mechanical Efficiency = 80%, weight of reciprocating parts = 20N, length of connecting rod = 0.3m, speed = 1500 rpm with possible over speed of 2500 rpm. Assume suitable missing data. (**June 2011) (April/May 2009)**
38) Design a connecting rod for an IC Engine running at 1800rpm and developing a maximum pressure of 3.15 N/mm^2; The diameter of the piston is 100mm; mass of the reciprocating parts per cylinder is 2.25kg; length of the connecting rod is 380mm; stroke of the piston is 190mm. 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 material of the rod may be taken as 8000 kg/m^3. The allowable stress in the bolts and cap is 60 N/mm^2 and 80 N/mm^2 respectively. Use Rankine's formulae. Take σ_c = 320 N/mm^2 and const $\alpha = 1/7500$. **(December 2008)**

39) The following data is given for a connecting rod. Engine speed = 1800 rpm; Length of connecting rod = 350mm, Length of stroke = 8mm. assume the C/S to be 'I' and assume A= $11t^2$; $I_{xx} = \left(\frac{419}{12}\right) t^4$ and $y = \left(\frac{5t}{2}\right)$. Calculate whipping stress in the connecting rod. **(May/June 2015) (DAC)**

40) Design a cast iron piston for four stroke IC Engine, for the following specifications: Cylinder bore = 120mm, Stroke length = 150mm. Maximum gas pressure = 5MPa, Brake mean effective Pressure = 0.7MPa. Fuel consumption = 0.25 kg/kW/hr. Speed = 2400 rpm. Assume any other data if necessary for the design. **(December 2017 O) (December 2015) (April/May 2014)**

41) Design head and ring section of cast iron piston for a four stroke IC engine, for the following specifications. Cylinder bore = 200 mm, stroke length 150mm, Max gas Pressure is 5 MPa, fuel consumption 0.25 kg/kW/hr, speed 2400 rpm, assume any other data necessary for the design. **(December 2017 O)**

42) Determine the diameter of the piston rod for a steam engine. The diameter of the cylinder is 750mm. The greatest difference between steam pressures on the two sides of the piston is 0.25 N/mm^2. The rod is made of mild steel and is secured to the piston by a tapered rod and nut, and to the cross head by a cotter. Assume modulus of elasticity as 2 x 10^5 MPa, and factor of safety as 7. Length of the rod may be taken as 1.6m. **(December 2012)**

43) Determine the diameter of the piston rod for a steam engine. The diameter of the cylinder is 750mm, the greatest difference between steam pressure on the two sides of the piston is 0.25 N/mm^2. The rod is made of mild steel and is secured to the piston by a tapered rod and nut, and to the cross head by a cotter. Assume modulus of elasticity as 2 x 10^5 MPa, and factor of safety as 7. Length of the rod may be taken as 1.6m. **(December 2012)**

44) Design a trunk type piston for a single cylinder four stroke diesel engine running at 1000 rpm. **(December 2009)**

45) A four stroke diesel engine has the following specifications: Brake power = 5kW, speed = 1200 rpm, IMEP = 0.35 N/mm^2, η_{mech} = 80%. Determine (a) bore and length of the cylinder (b) Thickness of cylinder head (c) size of the studs for the cylinder head. **(April/May 2013)**

46) Design a plain carbon steel crank shaft for a single acting 4 stroke single cylinder engine for the following data: Bore = 400mm; stoke = 600mm; Engine speed = 200 rpm; Mean effective pressure = 0.5 N/mm^2; Maximum Combustible Pressure = 2.5 N/mm^2; Weight of flywheel a\used as pulley = 50kN; Total bet pull = 6.5 kN. When the crank angle is 35^0 from T.D.C, the pressure on the piston is 1 N/mm^2 and the torque on the crank is maximum. The ratio of connecting rod length to the crank radius is 5. Assume any other required for the designs. **(Dec 2017) (Dec 2009)**

47) Design an overhung crank shaft with two main bearings and a flywheel in between them for an IC Engine, single cylinder 0.25m x 0.3m. The flywheel weighs 27kN. The maximum pressure is 2.1MPa. The torsional moment is maximum when the crank is at 35^0 from the IDC, while the pressure is 1.05MPa. Assume missing data. **(January 2012)**

48) Design a side or overhanging crankshaft for a 250 mm X 300 mm gas engine. The weight of flywheel is 30kN and the explosion pressure is 2.1 N/mm^2. The gas pressure at the maximum torque is 0.9 N/mm^2, when the crank angle is 35^0 from I.D.C. The connecting rod is 4.5 times the crank radius. **(May/June 2012)**

49) The following data is given for the piston of a four stroke diesel engine. Cylinder bore = 100mm, material of piston rings = Grey cast iron, allowable tensile stress = 90MPa, allowable radial pressure on cylinder wall = 0.035MPa, thickness of the piston head = 16mm, number of piston

rings = 4, calculate (i) radial width of piston rings (ii) axial thickness of piston rings (iii) gaps between the free ends of the piston ring before assembly (iv) gaps 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 (viii) thickness of barrel at open end. **(December 2013)**

50) The cylinder of a four stroke diesel engine has the following specifications: Brake power = 3kN, Speed = 800 rpm, Indicated mean effective pressure = 0.3MPa, Mechanical efficiency = 80%, Determine the bore and length of the cylinder liner. **(December 2013 O)**

51) A single cylinder double acting steam engine delivers 185kW at 100 rpm. The maximum fluctuation of energy per revolution is 15 percent of the energy developed per revolution. The speed variation is limited to 1 percent either way from the mean. The mean diameter of the flywheel of the rim is 2.4m. design the flywheel. **(June 2017)**

52) An electric motor drives a punching machine. A flywheel is fitted to the machine has a radius of gyration of 0.6m, and runs at 300 rpm. The machine can punch 600 holes per hour. Each punching operation taking 2 seconds and requiring 20000N-m of work. Determine the power required to operate the machine and the mass of the flywheel; if the speed of the wheel should not drop below 220 rpm. **(January 2015) (December 2015) (April/May 2014)**

53) An electric motor drives a punching machine. A flywheel is fitted to the machine has a radius of gyration of 0.5m, and runs at 240 rpm. The machine can punch 600 holes per hour. Each punching operation taking 1.5 seconds and requiring 1500N-m of work. Determine the power required to operate the machine and the mass of the flywheel; if the speed of the wheel should not drop below 230 rpm. **(May 2015)**

Unit-V
Theory of Bending

Part-A

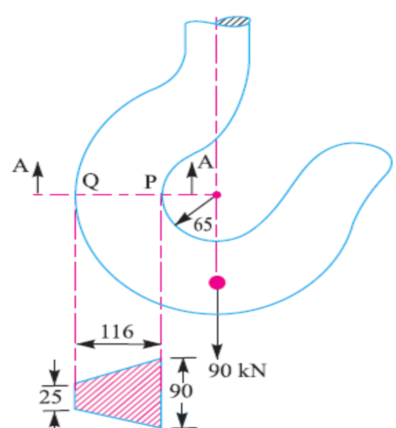
1) Explain Winkler Bach formulae used for curved beams.
2) Write the assumptions made by Winkler Bach wile deriving the theory of curved beams.
3) Write the relation between momenta and curvature for trapezoidal section and circular section.
4) Explain the design criteria for the machine frame (C-Clamp).
5) How curved beam theory of bending is different form straight beam? Explain.
6) Write short notes on theory of bending in different sections with sketches.
7) Mention the factors to be considered in the design of crane hooks.
8) Explain why forged steel is preferred materials for crane hook.
9) Briefly discuss the effect of initial curvature on the analysis of theory of bending of beams.
10) Draw the free hand good sketch for a crane hook with trapezoidal cross section.
11) Explain various stresses induced in curved beams.
12) Explain why unsymmetrical cross sections are preferred in curved beams?
13) What type of cross section is preferred for a crane hook and why?

Part-B

14) Find the load carrying capacity of a trapezoidal cross sectioned crane hook. The radius of curvature of the inner fiber is 50mm. Distance between parallel sides is 120 mm with sides of 30mm and 90mm. it is made of plain carbon steel of 45C8 (S_{yt} = 380 N/mm^2) and factor of safety is 3.5. **(December 2017) (June2017)**

15) A crane hook has a round cross section with diameter 90mm. The bed diameter is 120mm. (a) Determine the load which will produce a maximum stress of 125 N/mm^2 in the inner fibers; (b) Determine the load that will produced the corresponding stress in the outer fibers. **(Dec 2017)**

16) A crane hook has a trapezoidal section A-A as shown in figure. Find the maximum stresses at point P and Q Take σ_y= 350 MPa. **(Dec 2017 O)**



Section at A-A
All dimensions in mm.

17) Design a crane hook for a lifting capacity of 30kN with 50 percent over load. **(January 2015) (June2017 O) (May/June2015)**

18) A crane hook is having a circular cross section with diameter 100mm. the distance between the line of action of the load and centroidal axis of the cross section is 60mm. the material of the hook is 45C8 (σ_{yt} = 400 MPa) and factor of safety is 3.5. Determine the load carrying capacity of the crane hook. **(May 2016) (December 2010)**

19) A crane hook has a round cross section with diameter 95mm. The bed diameter is 125mm. (a) Determine the load which will produce a maximum stress of 125 N/mm^2 in the inner fibers; (b) Determine the load that will produced the corresponding stress in the outer fibers.

(December 2015) (May 2014)

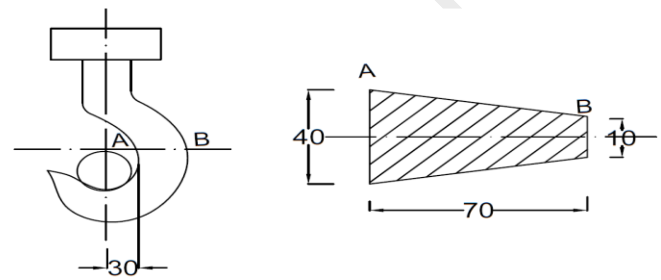
20) Design a crane hook with the lifting capacity of the crane as 250kW. The weight of the hook is 50kN. **(May/June 2009) (May/June 2010) (May/June 2012) (April/May 2013 O) (April/May 2013)**

21) (a) The width 'b' of rectangle be 100 mm and depth of section 'h' be 150mm, the value of radius R of the center of gravity is equal to 150mm for the curved beam of rectangular cross section, find the eccentricity, the stress concentration factor and the maximum stress.

(b) For the curved beam of circular cross section of diameter 200mm and the radius of curvatures is 40mm, then find the eccentricity, stress concentration factor and maximum stress.

(December 2013)

22) Design a crane hook of the firm shown in figure for a maximum load of 20kN. The radius of the inner fiber is 30mm and that of the outer fiber is 100mm from the line of action of load. Specify the position of the natural axis and stresses induced at inner and outer fibers. **(May/June 2011)**



All Dimensions are in mm

23) The bed diameter of a crane hook is 95mm. The section of the hook is trapezoidal with depth equal to 190mm. The width of the section at the larger end is 125mm and at the smaller end is 89mm. the load on the hook is 135kN. Determining the maximum unit stresses in tension and compression. **(January 2012)**

24) Write down the detail procedure for the design of crane hook of triangular section for given load. **(December 2009)**

25) Find the load carrying capacity of a trapezoidal cross sectioned crane hook. The radius of curvature of the inner fiber is 50mm. yield strength S_{yt} = 250 MPa. Use a factor of safety of 2. Distance between parallel sides are 40 mm and 80mm. Also find the working stress at a distance of 3mm on each side of the neutral axis with maximum load. **(December 2008)**

26) The bed diameter of a crane hook is 95mm. The section of the hook is trapezoidal with depth equal to 190mm. The width of the section at the larger end is 125mm and at the smaller end is 89mm. The load on the hook is 135kN. Determine the maximum unit stress in tension and compression. **(January 2012)**

27) Design a crane hook with the useful load lifting capacity of the crane as 50 kW. The weight of the hook with grabbing tongs is 10kN. **(Almost Every Year)**