Unit 1: Spur and Helical Gears

Assignment

1. Discuss the design procedure for spur Gears.
2. Discuss the classification of gears.
3. Explain the different causes of gear tooth failure, and suggest the remedies to avoid such failure.
4. State two important reasons, why we are adopting involute tooth profile.
5. What condition must be satisfied that a pair of gear have constant velocity ratio.
6. What is a herringbone gear? Where they are used?
7. Explain the following terms used in helical gears :

   (a) Helix angle; (b) normal pitch; and (c) axial pitch.
8. Define formative or virtual number of teeth on a helical gear. Derive the expression used to obtain its value.
9. Write the expressions for static strength, limiting wear load and dynamic load for helical gears and explain the various terms used therein.

Numerical

1. A gear drive is required to transmit a maximum power of 22.5 KW. The velocity ratio is 1:2 and rpm of pinion is 200. The approximate centre distance between the shafts may be taken as 600mm. The teeth are 20o stub involute profiles. The static stress for the gear material (which is CI) may be taken as 60 MPa & face width as 10 times module. Find module, face width and number of teeth on each gear. Check the design for dynamic &wear load. The deformation of dynamic factor in the Buckinggham equation may be taken as 80 and the material combination factor for the wear as 1.4**.**
2. A spur gear made of bronze drives a mild steel pinion with angular velocity ratio of 3.5:1. The pressure angle is 14.5o. It transmits 5KW at 1800 rpm of pinion. Considering on strength design the smallest diameter gear and find necessary face width. The number of teeth should not be less than 15 teeth on either gear. The elastic strength of bronze may be taken as. 84 N/mm2 & steel as 105 N/mm^2
3. Design a spur gear drive transmitting 30 KW at 400rpm to another gear running approximately at 100 rpm. The load is steady and continuous. The material for pinion & gear are cast steel and cast iron respectively. Take module 10mm. Check the design for dynamic load and wear.
4. 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.Check the design for bending and wear.

5 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. Check the gears thus designed from the consideration of wear. The surface endurance limit may be taken as 700 MPa.

6 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 618 MPa.

7 A pair of helical gears with 30° helix angle is used to transmit 15 kW at 10 000 r.p.m. of the pinion. The velocity ratio is 4 : 1. Both the gears are to be made of hardened steel of static strength 100 N/mm2. The gears are 20° stub and the pinion is to have 24 teeth. The face width may be taken as 14 times the module. Find the module and face width from the standpoint of strength and check the gears for wear.

8 A helical cast steel gear with 30° helix angle has to transmit 35 kW at 1500 r.p.m. If the gear has 24 teeth, determine the necessary module, pitch diameter and face width for 20° full depth teeth. The static stress for cast steel may be taken as 56 MPa. The width of face may be taken as 3 times the normal pitch. What would be the end thrust on the gear? Check the gear drive for dynamic load & wear.

Unit 2: Bevel and Worm Gears

Numerical

1 Sketch neatly the working drawing of bevel gears in mesh.

2 For bevel gears, define with the help of neat sketch the following :

(i) Cone distance; (ii) Pitch angle; (iii) Face angle; (iv) Root angle; (v) Back cone distance; and (vi) Crown height.

3 What is Tredgold's approximation about the formative number of teeth on bevel gear? Derive the expression for equivalent number of teeth.

4 What are the various forces acting on a bevel gear ?

5 Write the procedure for the design of a shaft for bevel gears.

6 Define the following terms used in worm gearing :

(a) Lead; (b) Lead angle; (c) Normal pitch; and (d) Helix angle.

7 What are the various forces acting on worm and worm gears ?

8 Write the expression for centre distance in terms of axial lead, lead angle and velocity ratio.

9 Derive the the expression for efficiency of worm gear.

10 Write short note on worm gear.

Numerical

1 A pair of cast iron bevel gears connect two shafts at right angles. The pitch diameters of the pinion and gear are 80 mm and 100 mm respectively. The tooth profiles of the gears are of 14 1/2º composite form. The allowable static stress for both the gears is 55 MPa. If the pinion transmits 2.75 kW at 1100 r.p.m., find the module and number of teeth on each gear from the standpoint of strength and check the design from the standpoint of wear. Take surface endurance limit as 630 MPa and modulus of elasticity for cast iron as 84 kN/mm2.

2 A pair of bevel gears connect two shafts at right angles and transmits 9 kW. Determine the required module and gear diameters for the following specifications :

Particulars	Pinion	Gear
Number of teeth	21	60

Material	Semi-steel	Grey cast iron
Brinell hardness number	200	160
Allowable static stress	85 MPa	55 MPa
Speed	1200 r.p.m.	420 r.p.m.
Tooth profile	14.5 composite	14.5ºComposite

Check the gears for dynamic and wear loads.

3 A pair of straight bevel gears is required to transmit 10 kW at 500 r.p.m. from the motor shaft to another shaft at 250 r.p.m. The pinion has 24 teeth. The pressure angle is 20°. If the shaft axes are at right angles to each other, find the module, face width, addendum, outside diameter and slant height. The gears are capable of withstanding a static stress of 60 MPa. The tooth form factor may be taken as 0.154 – 0.912/TE, where TE is the equivalent number of teeth. Assume velocity factor as 4.5 /(4.5 +v), where v the pitch line speed in m/s. The face width may be taken as 0.25of the slant height of the pitch cone.

4 A triple threaded worm has teeth of 6 mm module and pitch circle diameter of 50 mm. If the worm gear has 30 teeth of 14½° and the coefficient of friction of the worm gearing is 0.05, find 1. the lead angle of the worm, 2. Velocity ratio, 3. centre distance, and 4. Efficiency of the worm gearing.

5 A double threaded worm drive is required for power transmission between two shafts having their axes at right angles to each other. The worm has 14½° involute teeth. The centre distance is approximately 200 mm. If the axial pitch of the worm is 30 mm and lead angle is 23°, find 1. lead; 2. Pitch circle diameters of worm and worm gear; 3. helix angle of the worm; and 4. efficiency of the drive if the coefficient of friction is 0.05.

6 Design a speed reducer unit of worm and worm wheel for an input of 1 kW with a transmission ratio of 25. The speed of the worm is 1600 r.p.m. The worm is made of hardened steel and wheel of phosphor bronze for which the material combination factor is 0.7 N/mm2. The static stress for the wheel material is 56 MPa. The worm is made of double start and the centre distance between the axis of the worm and wheel is 120 mm. The tooth form is to be 14½° involute. Check the design for strength, wear.

Unit-3 Bearings

Assignment

1 What is meant by hydrodynamic lubrication & hydrostatic lubrication.

2 List the basic assumptions used in the theory of hydrodynamic lubrication.

3 Explain wedge film and squeeze film lubrication.

4 Make sketches to show the pressure distribution in a journal bearing with thick film lubrication in axial and along the circumference.

5 Explain the following terms as applied to journal bearings :

(a) Bearing characteristic number ; and (b) Bearing modulus.

6 What are rolling contact bearings? Discuss their advantages over sliding contact bearings.

7 How do you express the life of a bearing? What is an average or median life?

8 Define the following terms as applied to rolling contact bearings: (a) Basic static load rating (b) Static equivalent load (c) Basic dynamic load rating (d) Dynamic equivalent load.

9 Write short note on classifications and different types of antifriction bearings.

Numerical

1 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. 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.

2 A full journal bearing of 50 mm diameter and 100 mm long has a bearing pressure of 1.4 N/mm. The speed of the journal is 900 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 : 1. The amount of artificial cooling required, and 2. The mass of the lubricating oil required, if the difference between the outlet and

inlet temperature of the oil is 10°C. Take specific heat of the oil as 1850 J / kg / °C.

3 The main bearing of a steam engine is 100 mm in diameter and 175 mm long. The bearing supports a load of 28 kN at 250 r.p.m. If the ratio of the diametral clearance to the diameter is 0.001 and the absolute viscosity of the lubricating oil is 0.015 kg/m-s, find : 1. The coefficient of friction ; and 2. The heat generated at the bearing due to friction.

4 150 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.

5 A shaft rotating at constant speed is subjected to variable load. The bearings supporting the shaft are subjected to stationary equivalent radial load of 3 kN for 10 per cent of time, 2 kN for 20 per cent of time, 1 kN for 30 per cent of time and no load for remaining time of cycle. If the total life expected for the bearing is 20 × 106 revolutions at 95 per cent reliability, calculate dynamic load rating of the ball bearing.

6 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.

7 A single row deep groove ball bearing operating at 2000 r.p.m. is acted by a 10 kN radial load and 8 kN thrust load. The bearing is subjected to a light shock load and the outer ring is rotating. Determine the rating life of the bearing.

Unit 4: Internal Combustion Engine components

Assignment

1. Explain the various types of cylinder liners.

2 State the function of the following for internal combustion engine piston:

(a) Ribs;(b) Piston rings;(c) Piston skirt (d) Piston pin (e) Piston head with neat sketch.

3 State the design consideration while designing piston for IC engine and Discuss piston material.

4 Discuss the design of piston for an internal combustion engine.

5 Explain the various stresses induced in the connecting rod?

6 Sketch a valve gear mechanism of an internal combustion engine and label its various parts.

Numerical

1 Following data refers to a 4 stroke petrol engine:

Brake power = 7.5 Kw

Engine speed = 1500 rpm

Indicated mean effective pressure = 0.45 MPa

Maximum explosion pressure = 3.2 MPa

Mechanical Efficiency = 80%

Allowable stress for C.I. Cylinder = 40 MPa

Allowable stress for Ni-steel Cylinder = 70 MPa

Find ,(i) Bore and stroke of engine taking L/D = 1.25

(ii) Thickness of cylinder wall and flange

(iii) Thickness of cylinder head

(iv) Size and number of bolts required to join the cylinder head.

2 Design a cast iron piston for a single acting four stroke engine for the following data: Cylinder bore = 100mm; Stroke = 125mm; Maximum gas pressure = 5N/mm2; Indicated mean effective pressure=0.75N/mm2; Mechanical efficiency = 80%; Fuel

consumption = 0.15kg per brake power per hour; Higher calorific value of fuel = 42×103 kJ/kg; Speed = 2000 r.p.m. Any other data required for the design may be assumed.

3 Design a connecting rod for four stroke petrol engine for following data.

Piston diameter = 0.1 m

Stroke=0.14 m,

Length of C.R. =0.315 m

Weight of reciprocating part =18.2 N

Speed = 1500 rpm with over speed 2500

Compression ratio 4:1

Maximum explosion pressure = 2.45 MPa.

F.O.S. =5 , For connecting rod YTS = 380 N/mm2 , UTS= 580 N/mm2

4 The conical valve of an I.C. engine is 60 mm in diameter and is subjected to a maximum gas pressure of 4 N/mm2. The safe stress in bending for the valve material is 46 MPa. The valve is made of steel for which k = 0.42. The angle at which the valve disc seat is tapered is 30°.Determine: 1. thickness of the valve head; 2. stem diameter; and 3. maximum lift of the valve.