DEPARTMENT OF MECHANICAL ENGINEERING

Subject Name: Design of Machine Elements –II [302048]

(TE-A & B) AY 2018-19

UNIT 01: "DESIGN OF SPUR GEARS"

PROBLEMS:

1. A pair of spur gear with 20° full depth involute teeth consists of pinion having 21 teeth to be mesh with a gear having 60 teeth internal gear. The pinion shaft is directly coupled to 7.35 kW an Electric motor running at 1440 r.p.m. The gear shaft is transmitting power to machine. The application factor is 1.5. The pinion as well as gear are made of alloy steel (S_{ut}= 1500 N/mm²). The module & face width of gears are 3mm & 35mm. The gears are machined to specification of grade 8 & heat treated to surface hardness of 400 BHN. The deformation factor is 11500 e, N/mm. assuming the dynamic load is accounted by Bu Buckingham's equation. Calculate:

I. The factor of safety against bending failure
II. The factor of safety against pitting failure

Use following data:

Lewis form factor Y=0.484 - $\frac{2.87}{Z}$

Load-stress factor, k = $0.16\left(\frac{\mathrm{BHN}}{100}\right)^2$ $\mathrm{N/mm^2}$

For grade – 8, e = 16.00 + 1.25 (m + 0.25$\sqrt{\mathrm{d}}$) µm

Buckingham's equation for dynamic load: $F_{\mathrm{d}} = \frac{21\mathrm{V}\,(\mathrm{bC} + F_{\mathrm{t\,max}})}{(21\mathrm{V} + \sqrt{(\mathrm{bC} + F_{\mathrm{t\,max}})}}$,

$F_{\mathrm{t\,max}} = K_{\mathrm{a}} K_{\mathrm{m}} F_{\mathrm{t}}$

2. A spur gear pair with 20° full depth involute teeth consists of 20 teeth on pinion meshing with 41 teeth on gear. Face width 40mm & module is 3 mm. The material for pinion as well as gear is steel with an ultimate tensile strength 600 N/mm² with surface hardness of 400 BHN. The pinion rotates at 1450 rpm & service factor 1.75. Assume velocity factor accounts for dynamic. Determine rated power that the gear can transmit. Assume Factor of safety 1.5

3. Spur gear pair is used to transmit 7.5 KW power from electric motor running at 1440 rpm to machine tool at 370 rpm. Pinion & gear are made of carbon steel 40C8 (S_{ut}=720 N/mm²) & steel (S_{ut}= 600 N/mm²) respectively. Tooth system is 20° full depth involute & a tooth on pinion is 18. The service factor & load concentration factor are 1.25 & 1.2 respectively. Factor of safety is 1.2 while face width 12 times module. The gears are machined to specification of grade 7. Design the gear pair by using the velocity factor $C_v = \frac{3}{3+V}$ & Buckingham's equation for dynamic load.

Use following data:

Lewis form factor Y=0.484 - $\frac{2.87}{Z}$, Load-stress factor k = $0.16\left(\frac{\mathrm{BHN}}{100}\right)^2 \mathrm{N/mm^2}$

For grade – 7, e = 11.00 + 0.9 (m + 0.25$\sqrt{\mathrm{d}}$) μm

Buckingham's equation for dynamic load: $F_{\mathrm{d}} = \frac{21\mathrm{V}\,(\mathrm{bC}\ + F_{\mathrm{t\,max}})}{(21\mathrm{V} + \sqrt{(\mathrm{bC} + F_{\mathrm{t\,max}})}}$

Deformation factor C= 0.111e $\left[\frac{E_P.E_G}{E_P+.E_G}\right]$ N/mm

Modulus of elasticity for pinion & gear = 201 x 10³ N/mm²

Standard module (mm) = 1, 1.25, 1.5, 2, 3, 4, 5, 6, 8, 10, 12 & 16.

4. For spur gear pair, number of teeth on pinion is 18 & that of gear is 36. Using following data calculate the beam & wear strength of gear teeth, rated power the pair can transmit & static load on gear.

Ultimate tensile strength of pinion material is 660 N/mm²

Ultimate tensile strength of gear material is 510 N/mm²

Module = 5 mm & Surface hardness of pinion 330 BHN & that of gear is 280 BHN.

Velocity factor $C_v = \frac{5.6}{5.6 + \sqrt{V}}$, the service factor= 1.5, Factor of safety= 2.

Pinion speed is 1440 rpm & Lewis form factor Y=0.484 - $\frac{2.87}{Z}$

5. It is required to design a spur gear reducer for compressor running at 250 rpm driven by a 7.5 KW, 1000 rpm electric motor. The center distance between axes of the gear shaft should exactly 250mm. Starting torque of motor can be assumed to be 150% of rated torque. The gears are made of plain carbon steel 40C8 (S_{ut} = 700 N/mm²) .Pressure angle is 20˚. The Factor of safety is 2 for preliminary design based on use of velocity factor.

i. Design the gears & specify all dimensions.
ii. Assume that gears are manufactured to meet the requirement of Grade-6 & calculate the dynamic load by Buckingham's equation.
iii. Calculate Effective load & Factor of safety against bending failure.
iv. Using same Factor of safety against pitting failure, specify suitable Surface hardness for gears.

Use following data:

Lewis form factor Y=0.484 - $\frac{2.87}{Z}$, velocity factor $C_v = \frac{3}{3+V}$

Load-stress factor, k = $0.16 \left(\frac{\mathrm{BHN}}{100}\right)^2 \mathrm{N/mm^2}$

For Grade 6 e=8+0.63 [m+0.25$\sqrt{d}$] microns.

Deformation factor C= 11400 e N/mm.

Buckingham's equation for dynamic load: $F_{\mathrm{d}} = \frac{21\mathrm{V}\,(\mathrm{bC} + F_{\mathrm{t}})}{(21\mathrm{V} + \sqrt{(\mathrm{bC} + F_{\mathrm{t}})}}$ N

6. For spur gear pair with 20˚ Pressure angle, number of teeth on pinion is 25 mesh with & 60 teeth on gear. Module is 5 mm & face width is 45 mm. Pinion speed is 500 rpm. The gear pair is made of steel (S_{ut}= 600 N/mm²) with surface hardness of 220 BHN. Assume dynamic load accounted to velocity factor. Take service factor as 1.75, factor of safety as 2. & load deformation factor is unity. Calculate:
Beam strength, wear strength, static load that gear can transmit & rated power that can be transmitted by gear.

Use following data: Lewis form factor Y=0.484 - $\frac{2.87}{Z}$, velocity factor $C_v = \frac{3}{3+V}$

Load-stress factor, k = $0.16 \left(\frac{\mathrm{BHN}}{100}\right)^2 \mathrm{N/mm^2}$

7. A pair of spur gear with 20° full depth involute teeth consists of 25 teeth on pinion meshing with 60 teeth on gear. Electric motor transmits 7.5 KW power at 1440 rpm connected to pinion shaft. Both pinion & gear are made of carbon steel C40 with Ultimate tensile strength is 420 N/mm².

Use following data for design of gears based on Lewis equation,

Face width= 10 times module, Service factor as 1.75, factor of safety as 2,

Deformation factor C= 11400 e N/mm. Lewis form factor $Y_p = 0.340$; $Y_G = 0.421$,

Velocity factor $C_v = \frac{6}{6+v_P}$

For grade – 8, e = 16.00 + 1.25Ø where ø = (m + 0.25$\sqrt{\mathrm{d}}$) µm

Calculate:

a) Module & dimensions of pinion & gear.
b) Beam strength
c) Dynamic load using Buckingham's equation
d) Surface hardness for the gears & check the wear strength of the gear.

8. A pinion having 22 teeth is to mesh with a gear having 60 numbers of teeth. But pinion & gear made up of steel having Ultimate tensile strength of 600 MPa & 300 MPa respectively. The pinion is connected to a 10KW, 1440 rpm 3-ø induction motor. Design the gear pair & specify the surface hardness required on gear teeth using following data,

Starting torque of motor is 56 % greater than the rated torque.

Face width= 10 times module, Deformation factor C= 11400 N/mm²

Sum of errors between meshing teeth e= 7.3 microns, factor of safety = 1.75.

Velocity Factor $C_v = \frac{6}{6+V}$, Lewis form factor Y=0.484 - $\frac{2.87}{Z}$

Buckingham's equation for dynamic load: $F_{\mathrm{d}} = \frac{21\mathrm{V}\,(\mathrm{Ceb} + F_{\mathrm{t}})}{(21\mathrm{V} + \sqrt{(\mathrm{Ceb} + F_{\mathrm{t}})}}$ N

Standard module (mm) = 3, 4, 5, 6, 8, 10, 12 …….

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THEORY:

1) Discuss standard system of gear tooth
2) What are the effects of increasing or decreasing pressure angle in design of gear?
3) Define beam and wear strength of spur gear
4) State & explain the different types of gear tooth failures.
5) Discuss lubrication of gears
6) State and explain different types of gear tooth failures, their causes and remedies.

UNIT 02: "DESIGN OF HELICAL & BEVEL GEARS" (TE) AY 2017-18

PROBLEMS:

1. A pair of parallel **helical gear** with consists of pinion having 20 teeth to be mesh with a gear having 100 teeth gear. The pinion rotates at 720 rpm. The normal Pressure angle is 20˚ while helix angle 25˚. Face width 40mm & module is 4 mm. Pinion & gear are made of carbon steel 40C8 (S_{ut}=720 N/mm²) & steel (S_{ut}= 580 N/mm²) & heat treated to surface hardness of 300 BHN & 350 BHN respectively. The application factor is 1.5 & factor of safety is 2.0 Assume velocity factor accounts for dynamic load. Determine rated power that the gear can transmit. Use following data:

Lewis form factor Y'=0.484 - $\frac{2.87}{Z}$, Load-stress factor, k = $0.16\left(\frac{\mathrm{BHN}}{100}\right)^2 \mathrm{N/mm^2}$

Velocity factor $C_v = \frac{5.6}{5.6+\sqrt{V}}$ $\quad F_{\mathrm{t\,max}} = K_{\mathrm{a}} K_{\mathrm{m}} F_{\mathrm{t}}$

2. Helical gear pair is used to transmit 2.5 KW power from electric motor running at 5000 rpm. Speed reduction is 4:1. The normal Pressure angle is 20˚ while helix angle 23˚.The material for pinion as well as gear is steel with an ultimate tensile strength 750 N/mm². The service factor & factor of safety are 1.5 & 2 respectively. The gears are finish to specification for grade – 4.

i) In initial stage of gear design assume dynamic load accounted to velocity factor & Face width is 10 times module, pitch line velocity V= 10m/s. for estimating normal module.

ii) Select first preference module & find dimensions of gear.

iii) Calculate the dynamic load by Buckingham's equation also find factor of safety in bending.

iv) Specify suitable Surface hardness for gears at factor of safety is 2.0 Use following data:

$C_v = \frac{5.6}{5.6 + \sqrt{V}}$, Lewis form factor Y'=0.484$- \frac{2.87}{Z}$, Load-stress factor k = $0.16 \left(\frac{\mathrm{BHN}}{100}\right)^2 \mathrm{N/mm^2}$

For grade – 4, e = 3.20 + 0.25 (m_n + 0.25$\sqrt{\mathrm{d}}$) µm $\quad F_{\mathrm{t\,max}} = K_{\mathrm{a}} K_{\mathrm{m}} F_{\mathrm{t}}$

Buckingham's equation for dynamic load: $F_{\mathrm{d}} = \frac{21\mathrm{V}\left(\mathrm{bC}\, cos^2\alpha + F_{\mathrm{t\,max}}\right)\cos\alpha}{(21\mathrm{V} + \sqrt{(\mathrm{bC}\, cos^2\alpha + F_{\mathrm{t\,max}}})}$

Deformation factor C= 11500 e N/mm

First preference module (mm) = 1, 1.25, 1.5, 2, 3, 4, 5, 6, 8, 10, 12, 16 & 20.

3. A pair of **helical gear** with 20° full depth involute teeth consists of pinion having 18 teeth. The pinion shaft is directly coupled to 15 kW an Electric motor running at 720 rpm to machine at 360 rpm. The gear shaft is transmitting power to machine. The application factor is 1.25. Helix angle 26˚, Face width is 12 x normal module. Permissible bending stress for pinion & gear is 150 MPa. Combine tooth error is 40 microns. Deformation factor C=11600e N/mm. Assuming the dynamic load is accounted by Buckingham's equation. Calculate:

a) The *factor of safety* against **bending** failure.
b) The *factor of safety* against **pitting** failure.

Assume surface hardness of 350 BHN $\quad F_{\mathrm{t\,max}} = K_{\mathrm{a}} K_{\mathrm{m}} F_{\mathrm{t}}$

Lewis form factor Y'=0.484$- \frac{2.87}{Z}$

Buckingham's equation for dynamic load: $F_{\mathrm{d}} = \frac{21\mathrm{V}\left(\mathrm{bC}\, cos^2\alpha + F_{\mathrm{t\,max}}\right)\cos\alpha}{(21\mathrm{V} + \sqrt{(\mathrm{bC}\, cos^2\alpha + F_{\mathrm{t\,max}}})}$

THEORY:

7) Derive the relation for virtual number of teeth for helical gear.

8) What are the effects of increasing or decreasing helix angle in design of gear?
9) Derive expression for beam strength of bevel gear
10) State & explain the different mountings of bevel gear.
11) Compare straight bevel gear, spiral bevel gear & hypoid gear with sketch.

UNIT 03: "ROLLING CONTACT BEARING" (RCB)

PROBLEMS:

1) In particular application, the radial load acting on ball bearing is 5KN & expected life for 90% of bearing is 8000 hr. Calculate dynamic load carrying capacity of bearing, when shaft rotate at 1450 rpm.

2) A cylindrical roller bearing is subjected to 6000N. The desired life of bearing with 90% of reliability is 15000 hr. The application factor is 1.5. If shaft rotate at 1440 rpm, find dynamic load carrying capacity of bearing.

3) A taper roller bearing has dynamic load carrying capacity of 30 KN. The desired life for 90% of bearing is 8000 hr & speed of shaft is 300rpm, calculate equivalent radial load that the bearing can carry.

4) A 25KW, 1440rpm electric motor is directly coupled to shaft of 25mm diameter which is supported by two cylindrical roller bearings. The shaft transmits power to another line shaft through flat pulley of 300mm diameter, which place midway between two bearings. The coefficient of friction between belt & pulley is 0.3 while angle of lap is 180. The belt is horizontal. The load factor is 1.5. If expected life of bearing is 50,000hr, select bearing from manufacturer catalogue,

Use following data:

Bearing Number	**NU 2205**	**NU 2305**
Basic dynamic capacity	15.99	31.39

5) The radial load acting on ball bearing is 300N for 1^{st} five revolutions & reduces to 1500N for next ten revolutions. The load variation then repeats itself. The expected life of bearing is 30 million revolutions. Find dynamic load carrying capacity of bearing.

6) The deep groove ball bearing is to be selected. It is subjected to cyclic load as listed below. The expected life of bearing is 10,000 hrs. Assume radial load factor as 1 & axial load factor as 1.5. Find dynamic load carrying capacity of bearing.

Fraction of cycle	Type of load	Radial (N)	Axial (N)	Speed (rpm)	Shock & service factor
0.25	Heavy shock	3000	2000	550	3
0.25	moderate shock	2800	1500	650	2
Remaining	Light shock	2000	1400	750	1.5

7) A single row deep groove ball bearing operates with following work cycle. If the expected life of bearing is 25,000hrs, find dynamic load rating of bearing. work cycle is as follows,

Element time%	Radial load	Thrust load	Radial factor(X)	Thrust factor(Y)	Service factor	Speed (rpm)
50	2.8	1.2	0.56	1.4	1.5	720
20	2.8	0.8	0.56	1.6	2.0	1440
Remaining	Nil	Nil	-	-	-	720

8) A shaft with centrally mounted helical pinion is supported by deep groove ball Bearing at both ends. The center distance between the bearings is 100 mm. The shaft Transmits 5.50kW power at 3000r.p.m. The pitch circle diameter of the pinion is 80 mm. The normal pressure angle and helix angle are 20° and 19° of 95%. Calculate the dynamic basis capacity of the bearing which takes up the axial thrust, so that it can be selected from the manufactures catalogue based on a reliability of 90%. Assume: Shock load factor=1.25, Radial factor=0.56, Thrust factor=1.2

$$L=4.48\ L_{10}\left(\log_e\left[\frac{1}{R}\right]\right)^{\frac{1}{1.5}}$$

Where L is life with reliability R and L10 is rated life with reliability 90%.

9) A ball bearing operates with the following work cycle:

Element No.	Element Time %	Radial Load kN	Thrust Load kN	Radial Factor 'X'	Thrust Factor 'Y'	Race rotating	Service factor	Speed r.p.m.
1.	40	10	3	0.56	2	Inner	1	400
2.	30	5.5	1	1	0	Outer	1.25	800
3.	30	No Load	No Load	-	-	Inner	-	600

If the expected life of the bearing is 12000 hours with a reliability of 95% calculate the basic dynamic load rating of the bearing so that it can be selected from the manufacturers catalogue based on 90% reliability.

$$L=4.48\ L_{10}\left(\log_e\left[\frac{1}{R}\right]\right)^{\frac{1}{1.5}}$$

Where L is life with reliability R and L10 is rated life with reliability 90%.

10) A single row deep groove ball bearing operated with the following work cycle. If the expected late of the bearing is 13,000 hours with reliability of 90%. Calculate the dynamic load rating of the bearing and determine reliability of a system consisting of four such bearings. The work cycle is

Element No.	Element time	Fr (KN)	Fa (KN)	Radial factor	Thurst factor	Rate rotating	CS	Speed rpm
1	30%	5	1.5	0.56	1.1	Inner	1.25	960
2	40%	3.7	0.73	0.56	1.3	outer	1.4	1440
3	30%	–	–	–	–	outer	–	720

11) A ball bearing carries a radial load of 500N at 1760 rpm for 40% time, 650N at 880rpm for 30% time, 250N at l000rpm for 10% time and no load at 1500rpm for remaining period of the cycle. If the expected life of the bearing is 10,000 hours with 95% reliability, calculate: i) Basic dynamic load capacity of the bearing.

ii) Average speed of bearing operation.

Use following relation for reliability analysis:

$$\frac{L}{L_{10}} = \left[9.4911n\left(\frac{1}{R}\right)\right]^{\frac{1}{1.17}}$$

12) A single row deep groove ball bearing is used to support shaft of four Speed automobile gear box. It is subjected to following cycles.

Gear	Axial load 'N'	Radial load 'N'	Radial load factor X	Axial load factor Y	%Time engaged
First	3250	4000	0.56	1.176	1%
Second	500	2750	1	0	3%
Third	50	2750	1	0	21%
Fourth	Nil	Nil	1	0	75%

The shaft is connected to engine shaft & rotate at 1750 rpm. Calculate dynamic load carrying capacity, if expected life of bearing is 4000 hours.

THEORY:

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1) Discuss Load-life relationship for Rolling contact bearings.
2) Derive Stribeck's equation for basic static capacity of bearing with assumptions.
3) Short Note: a) Preloading of RCB b) Mountings of RCB.

UNIT 04: "WORM GEAR"

PROBLEMS:

1. A pair of worm & worm gear is designated as 2/30/10/8.
   Calculate, 1) The centre distance 2) The reduction ration 3) The dimensions of worm 4) The dimensions of worm gear.

2. A pair of worm gear designated as 2/52/10/4 transmits 10kW power at 7520rpm supplied to worm shaft. The coefficient of friction is 0.04 and pressure angle is 20^o. Assume worm is above the worm gear and rotates clockwise direction when viewed from left. If the worm is left hand determine: 1) Components of tooth forces acting on worm & worm gear. 2) Efficiency of worm.

3. A pair of worm gear designated as 4/40/10/4 transmits 10kW power. The worm is having left hand helix and is rotating at 1440rpm in CCW sense as viewed from RHS. The coefficient of friction is 0.04 and pressure angle is 20^o. Assume worm is located below the worm gear. 1) Determine and show the Components of tooth forces acting on worm & worm gear. 2) Efficiency of worm gear. 2) Power lost in friction.

4. A worm gear pair 2/30/10/8 consist of worm gear made of phosphor bronze (Ultimate tensile strength 245 Mpa) and worm made of hardened alloy steel (Ultimate tensile strength 700 Mpa). The coefficient of friction is 0.04 while the normal pressure angle is 20^o. The worm gear wear factor is 0.825Mpa. The fan is used for cooling purpose. for which overall heat transfer coefficient is 22W/m²°C. The permissible temperature rise for lubricating oil is above the atmospheric temperature which is 45°C.. The worm rotates at 720rpm. If the service factor is 1.25.
   Determine the input power rating based on,

1) Beam Strength

2) Wear strength

3) Thermal consideration.

Use the following Data: 1) Lewis Form factor Y = 0.484 - 2087/Zg 2) Velocity factor Cv = 6/(6+VG) 3) Area of housing A = $1.14*10^{-4}*a^{(1.7)}$ m². Where a is centre distance. 4) Efficiency = cosά – μ tanγ / (cosά + μtanγ). γ - Lead angle.

5. A worm gear box with an effective surface area of 1.5m is operating is still air with a heat transfer coefficient of 15W/m2°C. The temperature rise of lubricant is limited to 60° C. The worm gears are designated as 1/30/10/8. The worm shaft is rotating at 1440rpm and the normal pressure angle is 20°. Calculate the power rating based on thermal consideration for the drive.

6. A double start worm made of case hardened alloy steel 16 Ni 80 Cr 60 (Sut=700N/mm2) is to mesh with worm gear to be made of phosphor bronze (Sut= 240 N/mm2). The gear pair is required to transmit 5 kW power from an Electric motor running at 1500 rpm to a machine running at 75 rpm. The Service factor is 1.25 while the factor at safety required is 2.0. The face width of the worm gear is 0.73 times the pitch circle diameter of worm. The worm gear factor is 0.685 N/mm2 while the diametrical quotient is 10. The normal Pressure angle is 14.5°. If the coefficient of friction between worm and worm gear teeth is 0.03. Design the gear pair and find the power lost. Would you recommend a form for the gear box? Assume the permissible temperature rise is 50°C. Use following, Lewis form factor Y=0.484 - $\frac{2.87}{Zg}$, $C_v = \frac{6}{6+Vg}$, Area of housing A= 1.14 x 10-4 (a)1.7 m2; a= centre distance.

7. A worm gear drive is to be used to obtain a speed reduction of 20.5 from an input speed of 1450 r.p.m. and to transmit 20 kW. The material of worm gear is bronze with per. wear load factor K = 2.4 MPa when

   center distance is more than 200 mm and worm has hardness of 600 BHN and is ground. Use the data given below.

   Diameter factor = 10, Service factor = 2, No. of starts on the worm = 2, Coefficient of friction = 0.026

Per. Beam strength for worm gear material = 275MPa, Vel. Factor = 6/(6 + VG) where VG in m/s (assume 2.5 m/s initially), Form factor Y for normal pressure angle of 14.5° = 0.314, Worm gear width = 0.73 × worm P.C.D.

• Standard first preference values of module are = 1, 1025, 1.6, 2, 2.5, 3.15, 4, 5, 6, 8, 10, 12, 16, 20

• Factor of safety = 1, Design the worm & worm gear drive; Find also Heat losses, would you

Recommend blower for the gear box. If it is not possible to fit the blower

Then what will be the new value of module for worm gear will you suggest?

Use following, Lewis form factor Y=0.484 - $\frac{2.87}{Zg}$, $C_v = \frac{6}{6+Vg}$, Area of housing A= 1.14 x 10^{-4} (a)1.7 m^2; a= centre distance

THEORY:

1. Explain : i) Diameter quotient ii) Self locking worm iii) Rubbing velocity
2. Derive an expression for efficiency of worm gear drive.
3. Explain: Force analysis of worm gear drive.
4. Why the soft material like phosphor bronze is chosen for worm gear and alloy steel for worm.

UNIT 05: "BELT, ROPE, CHAIN DRIVE"

PROBLEMS:

1] A belt is required to transmit 18.5 kW from a pulley of 1.2 m diameter running at 250 rpm 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, μ = 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.

2] Select a V-belt drive to transmit 10 kW of power from a 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.

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3] Select a wire rope to lift a load of 10kN through a height of 600m from a mine. The weight of bucket is 2.5kN. The load should attain a maximum speed of 50m/min in 2 seconds.

4] 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 35 pitches.

5] A pump is driven by an electric motor through a open type flat belt drive. Determine the belt specifications for the following data. Motor pulley diameter(dS) = 300 mm, Pump pulley diameter(dL) = 600 mm Coefficient of friction (μS) for motor pulley = 0.25 Coefficient of friction (μL) for pump pulley = 0.20 Center distance between the pulleys=1000 mm; Rotational speed of the motor=1440 rpm; Power transmission = 20kW; density of belt material (ρ)= 1000 kg/m3 ; allowable stress for the belt material (σ) = 2 MPa; thickness of the belt = 5mm.

6] Find the power lost in friction assuming (1) uniform pressure and (ii) uniform wear when a vertical shaft of 100 mm diameter rotating at 150 r.p.m. rests on a flat end foot step bearing. The coefficient of friction is equal to 0.05 and shaft carries a vertical load of 15 kN.

7] Following are the details of a crossed belt drive. Calculate the length of the belt?

Diameter of the driver	:	200 mm
Diameter of the follower	:	400 mm
Center distance of the drive	:	2m
Speed of the driver	:	400 rpm
Angle of contact	:	197.3

Determine the length of the belt.

8] Select a flat belt to drive a mill of 250 rpm from a 10 kW, 730 rpm motor. Center distance is to be around 2 m. The mill shaft pulley is of 1 m diameter.

9] Design a flat belt drive for the following data: Power to be transmitted = 22.5 kW; driver speed = 740 rpm; speed ratio = 3; distance between the pulleys = 3 m; larger pulley diameter = 1.2 m.

10] A flat belt drive is to design to drive a flour mill. The driving power requirement of the mill is 22.5 kW at 750 rpm with a speed reduction of 3.0. The distance between the shaft is 3 m. Diameter of the mill pulley is 1.2 m. Design and make a neat sketch of the drive.

11] A motor of power 2 kW running at a speed of 1400 rpm transmits power to an air blower running at 560 rpm. The motor pulley diameter is 200 mm. The center distance may be 1000 mm. Design a suitable V-belt drive.

12] Design a V-belt drive to the following specifications: Power to be transmitted = 75 kW; Speed of driving wheel = 1440 rpm; Speed of driven wheel = 400 rpm; Diameter of driving wheel = 300 mm; Center distance = 2500 mm; Service = 16 hours/day.

13] A workshop crane carries a load of 30 kN using wire ropes and a hook. The hook weighs 15 kN. Diameter of the rope drum is 30 times the diameter of the rope. The load is lifted with an acceleration of 1 m/s2 . Find the diameter of the rope. Fs = 6; Er = 80 kN/mm2 , σU = 180 kN/mm2 ; cross section of thje rope = 0.4 x (dia of rope) 2 .

14] Design a chain drive to actuate a compressor from a 10 kW electric motor at 960 rpm. The compressor speed is to be 350 rpm. Minimum center distance should be 0.5 m. Compressor is to work for 8 hours/day.

15] Design a chain drive to actuate a compressor from a 10 kW electric motor. Speed of the motor shaft is 1050 rpm and the compressor speed is to be 350 rpm. Minimum center distance should be 600 mm. Compressor service required is 12 hours/day.

THEORY:

1] What are the advantages of a belt drive?

2] Why the slack side of the belt of a horizontal belt drive is preferable to place on the top side?

3] Which one should be the governing pulley to calculate tension ratio?

UNIT 06: "SLIDING CONTACT BEARINGS"

PROBLEMS:

1. The following data is given for 360˚hydrodynamic bearing:
Radial load=3.2KN
Journal speed=1490 rpm
Journal Diameter=50 mm
Bearing length=50 mm
Radial clearance=0.05 mm
Viscosity of lubricant=25cP
Assuming total heat generated in bearing is carried by total oil flow in the bearing calculates:

i. Coefficient of friction
ii. Power lost in friction
iii. Minimum oil film thickness
iv. Flow requirement in 1 liters/min

v. Temperature rise.

2. The following data is given for 360˚ hydrodynamic bearing used for electric motor:

External load= 9 KN

Length to diameter ratio=1

Journal speed=1350 rpm

Journal Diameter=100 mm

Diametric clearance=100 µm

The value of minimum oil film thickness variable is 0.3. Find the viscosity of oil that can be used.

3. Hydrodynamic bearing has diameter & length of 100mm. The radial load on bearing is 30KN.Journal speed is 1500 rpm & radial clearance is 100 microns. If the viscosity of oil is 25cP determine:

i. The minimum oil film thickness
ii. the coefficient of friction
iii. The power loss in friction
iv. Quantity of oil in circulation
v. Side leakage
vi. If makeup oil is supplied at 30˚C, Find average oil temperature.
vii. Assume specific gravity of oil as 0.86 & specific heat as 2.09 KJ/Kg˚C

4. The following data is given for 360˚hydrodynamic bearing:

Radial load=30 KN

Journal speed=3600 rpm

Journal Diameter=75 mm

Bearing length=75 mm

Radial clearance=0.15 mm

Inlet temperature=40˚C

The temperature & viscosity relationship is as follows:

T(˚C)	40	41	42	43	44	45	46	47	48	49	50
z(cP)	52.5	50	47.5	45	43	41	39	37.5	36	34	33

Assuming total heat produced in bearing is carried by total oil flow in the bearing. The specific gravity & specific heat of the lubricant are 0.86 & 1.76 KJ/Kg˚C respectively.

Calculate Power lost in friction & requirement of oil Flow.

5. Following data is given for a 360° hydrodynamic bearing.
Radial load = 10 kN
Journal speed = 1440 rpm
Unit bearing pressure = 1000 kpa
Clearance ratio (r/c) = 800
Viscosity of lubricant = 30 Mpa-s.
Assuming that the total heat generated in the bearing is carried by the total oil flow in the bearing. Calculate:

i. Dimensions Of Bearings
ii. Coefficient Of Friction
iii. Power List in Friction
iv. Total Flow of Oil
v. Side Leakage
vi. Temperature Rise
vii. Average Temperature (Inlet Temperature is 40°C) & Find maximum pressure (Pmax).

6. The following data refers to a 360° hydrodynamic journal bearing

- Radial load = 10kN
- Journal speed = 1450 rpm
- Length to Diameter ratio = 1
- Eccentricity = 15 microns
- Radial clearance = 20 microns
- Bearing length = 50mm
- Specific gravity of lubricant = 0.86
- Specific heat of lubricant = 2.09 kJ/ kg °C

Calculate:

i. The minimum oil film thickness
ii. the coefficient of friction
iii. The power loss in friction
iv. The viscosity of lubricant in cp
v. The total flow rate of lubricant in lit/min
vi. The side leakage
vii. The average temperature, if makeup oil is supplied at 30°C

7. Following data is given for 360° hydrodynamic bearing
Radial load =6.5 kN.
Journal diameter = 60 mm
Bearing length = 60 mm

Journal speed = 1200 rpm.
Minimum oil film thickness=0.009mm
The fit between the journal & bearing is normal fit H7e7 for which

Hole diameter = $60^{+0.00}_{+0.03}$ mm   Shaft diameter = $60^{-0.06}_{-0.09}$ mm

Specify the viscosity of the lubricating oil for the given journal bearing.

8. Following data is given for a full hydrodynamic bearing:
Journal diameter=100mm
Bearing length=50mm
Journal speed=1500rpm
Viscosity of lubricant=30cP
Minimum film thickness=15 microns
Specific gravity of lubricants=0.86
Specific heat of lubricants=2.09kJ/kg°C
The fit between the journal and bearing=H7e7
Calculate:

i. Load carrying capacity of the bearings,
ii. Side Leakage
iii. Temperature rise, considering the effect of side leakage.

Using the following data:

Diameter in mm	Tolerance, mm	
	H_7	e_7
100	+0.035	-0.072
	+0.000	-0.107

9. A 50mm diameter hardened and ground steel journal rotates at 1440 rpm in a lathe turned bronze bushing which is 50mm long. For hydrodynamic lubrication, the minimum oil film thickness should be five times the sum of surface roughness (clearance values) of journal and bearing. The data about machining methods is as follows Table 1

Table 1

Elements	Machining methods	Clearance values
Shaft	Grinding	1.6microns
Bearing	Turning/Boaring	0.8microns

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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. Also, calculate quantity of lubricating oil required.

THEORY:

1) What is infinitely long and short journal bearing? State conditions and write Reynolds equation for long and short journal bearing.
2) Write a note on- Additive for mineral oils.
3) Write a short note on selection of hydrodynamic bearing design variables
4) Discuss in detail the lubrication regimes
5) Derive from First principal Reynold's equation with usual notation.

   $$\partial/\partial x\left(h^3 \partial p/\partial x\right) + \partial/\partial y\left(h^3 \partial p/\partial y\right) = 6\mu u\, \partial h/\partial x$$
6) Write desirable properties of bearing material
7) State assumptions made in Petroff's equation and derive it.
8) State assumptions made in deriving 'Reynold's equation'.
9) Sketch pressure distribution in infinitely short hydrodynamic journal bearing.

All The Best