ST.ANNE'S COLLEGE OF ENGINEERING AND TECHNOLOGY

(Approved by AICTE, New Delhi. Affiliated to Anna University, Chennai)
(An ISO 9001: 2015 Certified Institution)
ANGUCHETTYPALAYAM, PANRUTI – 607 106.

DEPARTMENT OF MECHANICAL ENGINEERING

ME 8651- DESIGN OF TRANSMISSION SYSTEM

THIRD YEAR - SIXTH SEMESTER

PREPARED BY

K.SARAVANAN. ASP/MECHANICAL

ME8651 DESIGN OF TRANSMISSION SYSTEMS L T P C

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OBJECTIVES:

- To gain knowledge on the principles and procedure for the design of Mechanical power Transmission components.
- To understand the standard procedure available for Design of Transmission of Mechanical elements
- To learn to use standard data and catalogues (Use of P S G Design Data Book permitted)

UNIT I DESIGN OF FLEXIBLE ELEMENTS 9

Design of Flat belts and pulleys - Selection of V belts and pulleys – Selection of hoisting wire ropes and pulleys – Design of Transmission chains and Sprockets.

UNIT II SPUR GEARS AND PARALLEL AXIS HELICAL GEARS 9

Speed ratios and number of teeth-Force analysis -Tooth stresses - Dynamic effects – Fatigue strength - Factor of safety - Gear materials – Design of straight tooth spur & helical gears based on strength and wear considerations – Pressure angle in the normal and transverse plane Equivalent number of teeth-forces for helical gears.

UNIT III BEVEL, WORM AND CROSS HELICAL GEARS 9

Straight bevel gear: Tooth terminology, tooth forces and stresses, equivalent number of teeth. Estimating the dimensions of pair of straight bevel gears. Worm Gear: Merits and demerits terminology. Thermal capacity, materials-forces and stresses, efficiency, estimating the size of the worm gear pair. Cross helical: Terminology-helix angles-Estimating the size of the pair of cross helical gears.

UNIT IV GEAR BOXES 9

Geometric progression - Standard step ratio - Ray diagram, kinematics layout -Design of sliding mesh gear box - Design of multi speed gear box for machine tool applications - Constant mesh gear box - Speed reducer unit. – Variable speed gear box, Fluid Couplings, Torque Converters for automotive applications.

UNIT V CAMS, CLUTCHES AND BRAKES 9

Cam Design: Types-pressure angle and under cutting base circle determination-forces and surface stresses. Design of plate clutches –axial clutches-cone clutches-internal expanding rim clutches Electromagnetic clutches. Band and Block brakes - external shoe brakes – Internal expanding shoe brake.

TOTAL: 45 PERIODS

OUTCOMES:

Upon the completion of this course the students will be able to:

CO1 apply the concepts of design to belts, chains and rope drives.

CO2 apply the concepts of design to spur, helical gears.

CO3 apply the concepts of design to worm and bevel gears.

CO4 apply the concepts of design to gear boxes.

CO5 apply the concepts of design to cams, brakes and clutches

TEXT BOOKS:

1. Bhandari V, “Design of Machine Elements”, 4 th Edition, Tata McGraw-Hill Book Co, 2016.

2. Joseph Shigley, Charles Mischke, Richard Budynas and Keith Nisbett “Mechanical Engineering Design”, 8 th Edition, Tata McGraw-Hill, 2008.

REFERENCES:

1. Merhyle F. Spotts, Terry E. Shoup and Lee E. Hornberger, “Design of Machine Elements” 8 th Edition, Printice Hall, 2003.

2. Orthwein W, “Machine Component Design”, Jaico Publishing Co, 2003.

3. Prabhu. T.J., “Design of Transmission Elements”, Mani Offset, Chennai, 2000.

4. Robert C. Juvinall and Kurt M. Marshek, “Fundamentals of Machine Design”, 4 th Edition, Wiley, 2005 5. Sundararajamoorthy T. V, Shanmugam .N, “Machine Design”, Anuradha Publications, Chennai, 2003.

DEPARTMENT OF MECHANICAL ENGINEERING

ME8651– DESIGN OF TRANSMISSION SYSTEMS

PART – A (2 MARKS)

UNIT – I

DESIGN OF TRANSMISSION SYSTEM FOR FLEXIBLE ELEMENTS

1. State the -Law of Belting' (May/June 2007)

The law of belting states that the centerline of the belt when it approaches the pulley must lie in the mid plane of that pulley which should be perpendicular to the axis of the pulley. Otherwise the belts will runoff the pulley.

2. Explain the term crowning of pulleys. (April/May2008) (Nov 19)

Pulleys are provided. a -slight conical shapes (or) convex shapes in their rim's r surface in order to prevent the belt from running off the pulley due centrifugal force. This is known as crowning, of pulley. Usually the crowning height t may be 1/96 of pulley face width.

3. Distinguish between open drive and cross drive of a belt drive. (May-11, Nov-04)

Open Belt drive: Used with shafts arranged parallel and rotating in same direction.

Cross Belt drive: Used with shafts arranged parallel and rotating in opposite direction.

4. What are the types of belts? (Nov 18)

(a) Flat Belts

(b) V Belts.

(ii) Multiple V belt. (iii) Ribbed Belt.

(c) Toothed or Timing

(d) Round, Belts.

5. What are the materials used for belt? (May/june 2013)

Leather, cotton fabrics, rubber, animal's hair, silk, rayon, woolen etc

6. Indicate some merits and demerits of belt-drive,

Merits

1. Belt drives are used for long distance power transmission.

2. Their operations are smooth and flexible.

3. Simple in design and their manufacturing cost is lower.

Demerits,

1. They need large space.

2. Loss of power due to friction is more.

7. What is meant by the ply of belt? (Apr 19)

Flat belts are made of thin strips and laminated one over the other in order to get thick belt. These thin strips or sheets are called as plies of belt. Usually flat belts are made of 11 ply, 4 ply, 5 ply, 6 ply and 8ply belt etc And 4 ply belt is thicker hand 3 ply belt and so-on.

8. Mention the different types of joints employed for joining flat-belts.

(i) Cemented joints (ii) Laced joints (iii) Crest joints. (iv) Hinged joints.

9. What is belt rating?

Flat-belts are made of different sizes such as 3 ply, 4 ply and V - belts are made of different grades such as A, B, C, D and E grade belts. Belt rating is defined as the power transmitting capacity of unit size flat belt or a particular grade single V belt.

10. Specify the purpose of crowning of belts.

To prevent slipping from pulley due to centrifugal force

11. Explain creep in belts.

Since the tensions produced by the belt on the two sides of the pulley are not equal, the belt moves with a very negligible velocity, due to the difference of two tensions. This slow movement of the belt over the pulley is known as creep of belt and it is generally neglected."

12. How is a V-belt designated?

V-belt is designated by a grade letter followed by its inside length in code number, year of coding. For example, D 3048: IS 2494: 1964. M belts are designated by the grade letter and inside length only such D - 3048. Sometimes, the inside length may be denoted in inches as D.

13. How is wire-ropes designated?

A wire-rope is designated by the number of strands and the number of wires in each strand. For example,

a wire rope having six strands, and each strand containing nineteen wires can be denoted as 6 x 19rope.

14. Give the relationship of ratio of tensions in a V-belt drive.(April/May 2008)

$$T_1/T_2 = e^{(\mu\theta / \sin\beta)}$$

T_1- Tensions on tight side

T_2-Tensions on slack side

15. Define maximum tension in a belt? (April/May 2008)

Maximum tension in a belt (T) = σ.b.t

σ =maximum safe stress

b=width of the belt

t= thickness of the belt

16. What is silent chain? In what situations, silent chains are preferred? (Nov/Dec 2007)

A silent chain consists of a series of toothed plates pinned together in rows across the width of the chain. The silent chains are mostly employed for high speed and high power applications.

17. Give any three applications of chain drive. What are their limitations? (May-11) (Apr 18) (Nov 19)

Chain drives are widely used in transportation industry, agricultural industry, metal and wood working machines.

Limitations: Chain drives cannot be used for velocity ratio more than 10.

18. In what ways the timing belts are superior to ordinary V-belts? (May-15) (Nov 17)

Flat belt and V-belt drives cannot provide a precise speed ratio, because slippage occurs at the sheaves. But certain applications required an exact output to input speed ratio. In such situations, timing belts are used.

19. Mention the losses in belt drives. (Nov-14)

Slip and creep of the belt on the pulleys,

Windage or air resistance to the movement of belt and pulleys,

Bending of the belt over the pulleys and Friction in the bearings of pulley.

20. What is meant by “chordal action of chain”? Also name a company that produces driving chains. (May -15)

When chain passes over a sprocket, it moves as a series of chords instead of a continuous are as in the case of a belt drive. It results in varying speed of the chain drive. This phenomenon is known as chordal action.

Some of the company names producing chains are: Roto mechanical equipments, Chennai; Monal chains limited, Mumbai; Innotech engineers ltd., New Delhi.

21. **Briefiy explain about friction and its applications (Apr 18)**
Friction is said to be a resisting force that is developed between two relatively -moving surfaces. For some machines, this frictional force may be, an unwanted force and hence it is to be reduced to the maximum level. For some other machines. Bearings brakes, clutches are the good examples.

22.When do you use stepped pulley drive? When do you use fast and loose pulley drive (Nov 18)

A stepped or cone pulley drive is used for changing the speed of the driven shaft while the driving shaft runs at constant speed.

A fast and loose pulley drive is used when the driven or machine shaft is to be started or stopped whenever desired without interfering with the driving shaft.

UNIT – II

SPUR GEARS AND PARALLEL AXIS HELICAL GEARS

1. Specify the types of gears-failures. (May/june 20013) (Apr 18)

a) Tooth breakage. b) Pitting of tooth surface.

c) Abrasive- wears. d) Seizing of teeth etc.

2. How are the following terms defined? ((May/june 2013)

a) Pressure angle (a) is the angle making by the line of action common- tangent to the pitch circles of mating pars.

b) Module m is the ratio of pitch circle diameter to the number d of gear teeth, and is usually represented in millimeters.

3. What factors influence backlash? (April/May 2008)

The factors like errors in tooth thickness, pitch, tooth spacing, mounting misalignment, etc influence the backlash.

4. Write short notes on backlash of gears. (April/May 2008)

Backlash can be defined as the play between a mating pair of gear assembled condition.

5. What is a herringbone gears? (Nov/Dec 2009)

A herring bone gear is made of two single helical gears attached other hence called as double helical gear in which the teeth of be set in the opposite direction to the teeth of another gear arrangement the axial thrust produced in one gear will be null', thrust produced in another gear, and the resultant thrust is improves the life of the gear. Sometimes, a single cylindrical L block is ova-played for making, herringbone Gear

6. What is the advantage of helical gear over spur gear? (May -08) (Apr 18) (Nov 19)

Helical gears produce less noise than spur gears.

Helical gears have a greater load capacity than equivalent spur gears.

7. What are the common forms of gear tooth profile? (May -10)

Involute tooth profile and Cycloidal tooth profile.

8. How does failure pitting happen in gears? (Nov-11, May-04)

Pitting is the process during which small pits are formed on the activate surfaces of gear tooth. It is a surface fatigue failure which occurs when the load on the gear tooth exceeds the surface endurance strength of the material.

9. Differentiate between circular pitch and diametral pitch. (Nov-13, Nov-08, May-11)

Circular pitch, $p_c = \pi \times m$

Diametral pitch, $p_d = \pi / p_c = 1/m$

10. What are the materials used for gear manufacturing? (May- 11, May-09)

<u>Metallic gears</u>: Steel, cast iron, and bronze.

<u>Non-metallic gears</u>: Wood, rowhide, compressed paper and synthetic resins.

11. Where do we use spiral gears? (Nov-13)

Spiral gears are used to connect and transmit motion between two non-parallel and non intersecting shafts.

12. Why is gear tooth subjected to dynamic load? (Nov-14, May-09)

Inaccuracies of tooth spacing

Elasticity of parts

Deflection of teeth under load

Irregularities in tooth profiles

Misalignment between bearings

Dynamic unbalance of rotating masses.

13. Laws of gearing (Apr 19)

The law of gearing states that for obtaining a constant velocity ratio, at any instant of teeth the common normal at each point of contact should always pass through a pitch point (fixed point), situated on the line joining the centre's of rotation of the pair mating gears.

UNIT – III

BEVEL, WORM AND HELICAL GEARS

1. In which gear-drive, self-locking is available? (Apr 18)

Self locking is available in worm-gear drive.

2. When do we use worm-gears? (May/June 2013) (Nov 19)

When we require to transmit power between nonparallel and non-intersecting shafts and very high velocity ratio, of about 100, worm gears, can be employed. Also worm-gears provide self-locking facility.

3. What are the forces acting on the bevel gears? (May / June 2013) (Nov/Dec 2009, 2018)

(i) Tangential force (ii) Axial force (iii) Radial force

4. What is the effect of increasing the pressure angle in gears? (Nov-11)

The increase of the pressure angle results in a stronger tooth, because the tooth acting as a beam is wider at the base.

5. What is working depth of a gear-tooth? (May-11)

Working depth is the radial distance from the addendum circle to the clearance circle. It is equal to the some of the addendum of the two meshing gears.

6. Name few gear materials. (May-11, May-12)

The materials used in the gear are

Steel, Cast iron and Bronze.

7. Mention the characteristics of hypoid gear. (May -10)

Hypoid gears are similar in appearance to spiral-bevel gears. They differ from spiral gears in that the axis of pinion is offset from the axis of gear. The other difference is that their pitch surfaces are hyperboloids rather than cones.

In general, hypoid gears are most desirable for those application involving large speed reduction ratios. They operate more smoothly and quietly than spiral bevel gears.

8. Calculate the angle between the shafts of a crossed helical gears made of two right handed helical gears of 15º helix angle each. (May-09)

Shaft angle = 2 (15°) = 30°

9. When is bevel gear preferred? (May-09, May-11) (Apr 18)

Bevel gears are used to transmit power between two intersecting shafts.

10. How can you specify a pair of worm gears? (May -08, May-09)

A pair of worm gears is specified as: (z_1/ z_2 / q / m_x)

Where z_1=Number of starts on the worm,

z_2 = Number of teeth on the worm wheel, q = Diameter factor = d_1/ m_x, and

m_x = Axial module.

11. Give the advantage of worm gear drive in weight lifting machines.(May-08) (Nov 17)

The worm gear drives are irreversible. It means that the motion cannot be transmitted from worm wheel to the worm. This property of irreversible is advantageous in load hoisting application like cranes and lifts.

12. State the advantages of herringbone gear. (May -15, Nov-08)

Herringbone gears eliminate the existence of axial thrust load in the helical gears. Because, in herringbone gears, the thrust force of the right hand is balanced by that of the left hand helix.

13. What is a zerol bevel gear? (May -15, Nov-07)

Spiral bevel gear with curved teeth but with a zero degree spiral angle is known as zerol bevel gear.

14. What is virtual number of teeth in bevel gears? (May-14, Nov-14)

An imaginary spur gear considered in a plane perpendicular to the tooth of the bevel gear at the larger end is known as virtual spur gear.

The number of teeth z_v on this imaginary spur gear is called virtual number of teeth in bevel gear. $z_v = z/\cos\delta$

Where z = actual number of teeth on the bevel gear and δ = pitch angle.

15. Define the following terms: a. Cone distance, b. Face angle. (May-14)

Back cone distance is the length of the back cone. Back cone is an imaginary cone, perpendicular to the pitch cone at the end of the tooth.

Face angle is the angle subtended by the face of the tooth at the cone centre.

16. What is the difference between an angular gear and a miter gear? (Nov-13, 17)(Apr 19)

When the bevel gears connect two shafts whose axes intersect at an angle other than a right angle, then they are known as angular bevel gears.

When equal bevel gears (having equal teeth and equal pitch angles) connect two shafts whose axes intersect at right angle, then they are known as mitre gears.

UNIT IV

DESIGN OF GEAR BOXES

1. What purpose does the housing of gear-box serve? (Apr 18)

Gear-box –housing or casing is used as container inside which, the gears, shafts, bearings and other components are “mounted.’ Also it prevents the entry of dust inside the housing and reduces noise of operation. That is, the housing Safe-guard the inner components.

2. What is step ratio in a gear box? (May-12, Nov-09)

Step ratio is the ratio of one speed of the shaft to its previous lower speed since the spindle speeds are arranged in geometric progression, the ratios adjacent speeds (i.e., step ratios) are constant. I f Nr is; the maximum speed and N, is the minimum speed, then, Nr (r-1) = (Step ratio).

3. What is step ratio? Name the series in which speeds of multi-speed gear box are arranged. (May-14)

When the spindle speeds are arranged in geometric progression, then the ratio between the two adjacent speeds is known as step ratio or progression ratio.

R20 and R40 series are used in the design of multi-speed gear boxes.

4. What are preferred numbers? (May/June 2013)

The preferred numbers are the conventionally rounded off values derived from geometric series. There are five basic series, denoted as R5, R20, R40 and R80 series.

5. What are the possible arrangements to achieve 12 speeds from a gear box? (May 2013)

(i) 3x2x 2 schemes (ii) 2x3x2 scheme (iii) 2x2x3 scheme

6. Differentiate ray diagram and structural diagram? (Nov/Dec 2009) (Apr 18) (Nov 19)

Ray diagram: The ray diagram (or) kinematic arrangement of gear box indicates the arrangement of various gears in various shafts of the gear box in order to obtain the different output speeds from the single speed of motor.

Structural diagram: speed diagram (or) structural diagram is the graphical representation of different speeds of output shaft, motor shaft and intermediate shaft.

7. Specify four types of gear boxes. (Nov-14, May-11) (Nov 19)

Sliding mesh gear box

Constant mesh gear box

Synchromesh gear box

Planetary gear box

8. List the ways by which the number of intermediate steps may be arranged in a gear box.

(May-10, May-12)

S.No.	Number of speeds	Preferred structural formula
1.	6 speeds	i. 3 (1) 2 (3) ii. 2 (1) 3 (2)
2.	8 speeds	i. 2 (1) 2 (2) 2 (4) ii. 4 (1) 2 (4)
3.	9 speeds	i. 3 (1) 3 (3)
4.	12 speeds	i. 3 (1) 2 (3) 2 (6) ii. 2 (1) 3 (2) 2 (6) iii. 2 (1) 2 (2) 3 (4)

9. Which type of gear is used in constant mesh gear box? Justify. (Nov-09, May-11)

Helical gears are used in constant mesh type gear boxes to provide quieter and smooth operation

10. What is speed reducer? (Nov-08, May-10)

Speed reducer is a gear mechanism with a constant speed ratio, to reduce the angular speed of output shaft as compared with that of input shaft.

11. What are the methods of lubrication in speed reducers? (Nov-11)

1. Splash or spray lubrication method

2. Pressure lubrication method

12. What is the function of spacers in a gear box? (May-12, Nov-04)

The function of spacers is to provide the necessary distance between the gears and the bearings.

13. Define progression ratio. [Nov 18]

In mathematics, a geometric progression, also known as a geometric sequence, is a sequence of numbers where each term after the first is found by multiplying the previous one by a fixed, non-zero number called the common ratio.

For example, the sequence 2, 6, 18, 54, ... is a geometric progression with common ratio 3. Similarly 10, 5, 2.5, 1.25 ... is a geometric sequence with common ratio 1/2.

Examples of a geometric sequence are powers r^k of a fixed number r, such as 2^k and 3^k. The general form of a geometric sequence is

UNIT V

DESIGN OF CAM CLUTCHES AND BRAKES

1. State about the profile of cam that gives no jerk and mention how jerk is eliminated. (May-12)

4-5-6-7 polynomial cam profile gives zero jerks. Profile smoothing techniques can remove the excessive jerks in a cam profile.

2. Why is it necessary to dissipate the heat generated during clutch operation? (May-13, Nov-11)

When clutch engages, most of the work done will be liberated as heat at the interface. Consequently the temperature of the rubbing surface will increase. This increased temperature may destroy the clutch. So heat dissipation is necessary in clutches.

3. What is self-locking in a brake? (Nov-11, May-13) (Nov 18)

When a frictional force is sufficient enough to apply the brake with no external force, then the brake is said to be self-locking brake.

4. What are the factors upon which the torque capacity of a clutch depends? (May-11, Nov-10)

Torque capacity of a clutch depends on

Number of pair of contact surfaces,

Coefficient of friction,

Axial thrust exerted by the spring, and

Mean radius of friction surface.

5. When do we use multiple disk clutches? (May -10)

A multiple clutch is used when large amount of torque is to be transmitted. In a multi plate clutch, the number frictional linings and the metal plates are increased which increases the capacity of the clutch to transmit torque.

6. What is the disadvantage of block brake with one short shoe? What is the remedy? (May-11)

If only one block is used for braking, then there will be side thrust on the bearing of wheels shaft. This drawback can be removed by providing two blocks on the two sides of the drum. This also doubles the braking torque.

7. Under what condition of a clutch, uniform rate of wear assumption is more valid? (May-09)

If the clutch is an old clutch, then uniform rate of wear assumption is more valid.

8. Name four profiles normally used in cams. (May-10)

The four profiles normally used in cams are

Uniform velocity, Simple harmonic motion, Uniform acceleration and retardation and Cycloidal motion.

9. How the "uniform rate of wear" assumption is valid for clutches? (May-08)

In clutches, the value of normal pressure, axial load for the given clutch is limited by the rate of wear that can be tolerated in the brake linings. Moreover, the assumption of uniform wear rate gives a lower calculated clutch capacity than the assumption of uniform pressure. Hence clutches are usually designed on the basis of uniform wear.

10. What are the significances of pressure angle in cam design? (Nov-07)

The pressure angle is very important in cam design as it represents steepness of the cam profile. If the pressure angle is too large, a reciprocating follower will jam in its bearing.

11. If a multidisc clutch has 6 discs in driving shaft and 7 discs in driven shaft then how many number of contact surfaces it will have? (May -15)

$n_1 = 6$, $n_2 = 7$

Number of pair of contact surface, $n = n_1+n_2-1$

$= 6+7-1 = 12$

12. Why in automobiles, braking action when traveling in reverse is not as effective as when moving forward? (May -15)

When an automobile moves forward, the braking force acts in the opposite direction to the direction of motion of the vehicle. Whereas in reverse travelling the braking force acts in the same direction to the direction of motion of the vehicle. So it requires more braking force to apply brake.

13. Differentiate between uniform pressure and uniform wear theories adopted in the design of Clutches. (Nov-14)

In clutches, the value of normal pressure, axial load for the given clutch is limited by the rate of wear that can be tolerated in the brake linings. Moreover, the assumption of uniform wear rate gives a lower calculated clutch capacity than assumption of uniform pressure. Hence clutches are usually designed on the basis of uniform wear.

14. Classify clutches based on the coupling methods. (May-14)

Positive contact clutches

Frictional clutches

Overrunning clutches, Magnetic clutches and Fluid couplings

15. What is meant by a self-energizing brake? (May-13) (Nov 17)

When the moment of applied force (F) and the moment of the frictional force ($\mu.R_N$) are in the same direction, then frictional force helps in applying the brake. This type of brake is known as a self- energizing brake.

16. Define pitch point in cam. (Nov-13)

The pitch point is the point on the pitch curve of the cam having maximum pressure angle.

17. Differentiate between clutch and a brake. (Nov-13)

A clutch connects two moving members of a machine, whereas a brake connects a moving member to a stationary member.

18 .In what ways, the clutches are different from brakes? (Nov/Dec 2011) (Apr 18) (Apr 19)

The clutch used to engage the driving and driven members and keep them moving (i.e., rotating) together, where as brakes are employed to stop a moving member or reduce its speed.

19. Differentiate brakes and dynamometer.[A/M -2017]

Brake is a mechanical device by means of which frictional force is applied to a moving machine member in order to slow or stop the motion.

The function of a dynamometer is to measure the forces or couples which tend to change the state of rest or of uniform motion of a body.

20. Distinguish between wet and dry operation of clutches. (Nov 17)

Sl. No	Wet clutch	Dry clutch
1	When the clutch operates proper lubrication it is said to be operating under wet condition.	When the clutch operates without any lubrication it is said to be operating under dry condition.
2	The coefficient of friction and torque transmitted are comparatively low	The coefficient of friction and torque transmitted are comparatively high
3	Due to the presence of the lubricant , considerable amount of heat is dissipated from the clutch.	Due to the absence of any medium the rate of heat dissipation is low.

21. What is torque converter? (Apr 19)

A torque converter is a type of fluid coupling which transfers rotating power from a prime mover, like an internal combustion engine, to a rotating driven load.

22. Why are cone clutches better than disc clutches. (Apr 19, Nov 19)

A cone clutch serves the same purpose as a disk or plate clutch. However, instead of mating two spinning disks, the cone clutch uses two conical surfaces to transmit torque by friction

UNIT - I

DESIGN PROCEDURE FOR FLAT BELTS :-

1. Selection of pulley diameter

2. calculation of design power in kW

Design power = Rated power x Service factor x arc of contact in kW

$= P \times K_s \times K_a$ in kW [Refer PSG 7.53 & 7.54]

3. Selection of Belting

4. Load rating

Load rating at V m/s = load rating at 10 m/s $\times \frac{V}{10}$

[Refer PSG 7.54]

5. Determination of Belt width (b) :

Total width of belt (b) = $\frac{\text{Design power}}{\text{load rating}}$ in mm

Refer PSG 7.52

6. Determination of pulley width (a):

Refer PSG 7.54

7. calculation of length (L)

For open belt drive $L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$ in mm

Refer PSG 7.53

8. Initial tension of Belt (or) Belt Tension :-

For based on plies 1% of length

eg: (6 plies)

Refer PSG 7.53

UNIT-I

Design of pulleys – Refer P.S.G. Data book 7.56

Design of flat belts:- using mfg data – Basic data – power, speed (i/p & o/p), dia (or) D & d of pulley, centre distance of pulley; using basic equations

1. Select a flat belt to drive a mill at 250 rpm from 10 KW, 730 rpm motor; centre distance is to be around 2 m. The mill shaft pulley is of 1m diameter

Given data:-

Rated power $P = 10KW$

Centre distance $C = 2m = 2000mm$

Speed N_2 (driver) $= 730$ rpm (max)

Speed N_1 (driven) $= 250$ rpm (min)

pulley diameter $D = 1m = 1000mm$

Solution:-

1. Selection of pulley diameter

$$D = 1m$$

2. calculation of design power in kW

Design power = Rated power × Service factor (or) Load correction factor × arc of contact factor

$$= P \times K_s \times K_a \text{ in kW}$$

Service factor $K_s = 1.3$ (Assume heavy duty intermittent load) Refer PSG : 7.53

Arc of contact factor $= 180^\circ - \left(\frac{D-d}{C} \times 60\right)$ PSG : 7.54

We know that relation

$$i = \frac{z_1}{z_2} = \frac{D}{d} = \frac{V_2}{V_1} = \frac{N_2}{N_1}$$

$$\therefore i = \frac{N_2}{N_1} = \frac{730}{250} = 2.92$$

Similarly $i = \frac{D}{d} \Rightarrow 2.92 = \frac{1000}{d}$ [D = 1m = 1000mm]

$\therefore d = 345\,mm$ (dia of smaller pulley)

arc of contact $= 180^\circ - \left(\frac{D-d}{C} \times 60\right)$

$= 180^\circ - \left(\frac{1000-345}{2000} \times 60\right)$ [C = 2m = 2000mm]

$= 160.4^\circ$

$\therefore K_a = 1.08$ Refer PSG 7.54

$\therefore$ Design power $= 10 \times 1.3 \times 1.08 = 14.04\,kW$

3. Selection of Belting

Assume [Refer PSG 7.52]

Dunlop "FORT" 949g fabric belting, using in the construction hard fabric having nominal weight of 949 g/m², recommended for heavy duty and for medium belt speeds.

4. Load rating :-

Load rating at v m/s = load rating at 10 m/s $\times \frac{V}{10}$

PSG : 7.54

load rating at 10 m/s = 0.0289 kW/mm/ply [For FORT 949]

WKT

$$V = \frac{\pi d N_2}{60} = \frac{\pi \times 345 \times 730}{60} = 13.2\,m/s = 13.2\,m/s$$

Load rating at v m/s = 0.0289 per mm width - per ply at 180° - of arc of contact and 10 m/s.

$$= \left(0.0289 \times \frac{13.2}{10} \times \frac{160.4^\circ}{180} \times \text{Belt ply}\right)$$

kw / mm width.

Belt ply

Based on $V = 13.2\,m/s$ so consider $15\,m/s$

$d = 350\,mm$

So No. of plies = 6 at PSG 7.52

∴ at standard widths table consider 6 PLY mm

$$\therefore = \left(0.0289 \times \frac{13.2}{10} \times \frac{160.4^\circ}{180} \times 6\right) \text{kw/mm width}$$

$$= 0.204 \text{ kw per mm width.}$$

5. ∴ Determination of belt width (b) :-

$$\text{Total width of belt (b)} = \frac{\text{Design power}}{\text{load rating}} = \frac{14.04 \text{ kw}}{0.204 \text{ kw/mm}}$$

$$(b) = 69 \text{ mm}$$

∴ Next higher standard belt width 112 mm. PSG 7.52

6. Determination of pulley width (a)

$$a = 112 + 13 = 125 \text{ mm}$$

PSG 7.54

7. Calculation of length

For open belt drive $L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$

PSG 7.53

$$= (2 \times 2000) + \frac{\pi}{2}(1000+345) + \frac{(1000-345)^2}{4 \times 2000}$$

$$\therefore L = 6166 \text{ mm}$$

8. Initial Tension of Belt:- (or) Belt Tension:-

For 6 plies 1% of Length PSG 7.53

$= 0.01 \times 6166 = 62 mm.$

$\therefore$ initial Tension $= 6166 - 62$
$= 6104$ mm.

$\therefore$ Result:-

1. pulley diameter D = 1m (or) 1000mm
2. Design power = 14.04 KW
3. Dunlop "FORT" 949 g fabric belting
4. velocity v = 13.2 m/s
5. Belt ply = 6
6. Belt width = 112 mm
7. Length L = 6166 mm
8. pulley width = 125 mm
9. Initial tension of belt = 6104 mm.

2. Design a flat belt drive to transmit 25 kw at 720 rpm to an aluminium rolling machine the speed reduction being 3.0. The distance between the shaft is 3 m. Diameter of rolling machine pulley is 1.2 m

(H.W)

DESIGN PROCEDURE FOR V BELTS

1. Selection of Belt cross Section:

   Based on power / load PSG 7.58

2. Calculation of pulley diameter(D):

   Based on ratio(i) PSG 7.54

3. Selection of center distance (c):

   Based on ratio (i) PSG 7.61

4. Nominal pitch length (L)

   $$L = 2c + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4c}$$ PSG 7.61

5. Minimum power Transmitting capacity

   Refer PSG 7.62

6. Number of Belts calculation

   $$\text{Number of Belts} = \frac{P \times F_a}{kW \times F_c \times F_d}$$

   PSG 7.70

7. Actual centre distance (c):

   $$C = A + \sqrt{A^2 - B}$$ mm Refer PSG 7.61

8. Dimensions of V-grooved pulley

   Based on Belt cross section

   Refer PSG 7.70

9. Sketch of V-Belt drive

Design of V Belts:-

3. A V Belt drive is to transmit 15 KW to a compressor. The motor runs at 1150 rpm and the compressor is to run at 400 rpm. Determine belt specification; number of belts; correct centre distance; drive pulley diameters

(N/D-14, 15, A/M-2018)

Given data:-

V belt drive

Power $P = 15\,KW$

Motor speed (driver) $N_2 = 1150\,rpm$

Compressor speed (driven) $N_1 = 400\,rpm$

Solutions:-

1. Selection of belt cross section:

For power / load of 15 KW

PSG 7.58

The cross section "B" is selected.

lly recommended minimum pulley pitch diameter $d = 125\,mm$

Nominal top width $W = 17\,mm$

Nominal thickness $T = 11\,mm$

2. Calculation of pulley diameter:-

wkt $i = \frac{D}{d} = \frac{N_2}{N_1}$

(large pulley) $D = \frac{N_2 \times d}{N_1} = \frac{1150 \times 125}{400}$

$D = 359.37\,mm$

PSG 7.54 Standard large pulley diameter = 400 mm.

3. Selection of center distance :- (C)

WKT speed ratio $i = \frac{D}{d} = \frac{400}{125} = 3.2$ PSG 7.61

For a speed ratio of 3; the recommended C/D ratio = 1.0 PSG 7.61

$$\frac{C}{D} = 1 \Rightarrow C = D \times 1 = C = 400 \times 1$$

$$C = 400\,mm$$

4. Nominal pitch length

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$ PSG 7.61

$$L = (2 \times 400) + \frac{\pi}{2}(400+125) + \frac{(400-125)^2}{4 \times 400}$$

$$L = 1671.93\,mm$$

PSG 7.59 & 7.60

For cross section "B" the next higher nominal pitch length = 1694 mm.

length of correction factor, $F_c = 0.94$

5. Minimum power transmitting capacity :-

From PSG 7.62

For "B" cross section

$$KW = \left(0.79S^{-0.09} - \frac{50.8}{d_e} - 1.32 \times 10^{-4} S^2\right)S$$

S - belt speed (or) velocity of Belt in m/s

$$S = \frac{\pi d N_1}{60} \text{ (or) } \frac{\pi D N_2}{60} \text{ in m/s}$$

$$S = \frac{\pi \times (0.125) \times 1150}{60} \quad [125 mm = 0.125 m]$$

$$S = 7.53 \text{ m/s}$$

Equivalent pitch diameter, $d_e = d_p \times F_b$

d_p - Smaller pulley diameter = 125 mm

F_b - Smaller diameter factor account of arc of contact.

Speed ratio range, $\frac{D}{d} = \frac{400}{125} = 3.2$

PSG 7.62

for D/d ratio 2.949 and over, $F_b = 1.14$ mm

$$\therefore d_e = 125 \times 1.14 = 142.5 mm$$

$$\therefore KW = \left(0.79 \times 7.53^{-0.09} - \frac{50.8}{142.5} - 1.32 \times 10^{-4} \times 7.53^2\right) 7.53$$

$$= 2.22 KW$$

6. Number of Belts calculation:-

For PSG 7.70

$$\text{Number of Belts} = \frac{P \times F_a}{KW \times F_c \times F_d}$$

P = drive power in kW = 15 kW

F_c = Correction factor for length = 0.94 (already found) PSG 7.69

F_d = Correction factor for arc of contact

from PSG 7.68

Arc of contact angle, $\theta = 180^\circ - 60^\circ \left(\frac{D-d}{C}\right)$

$$= 180 - 60 \times \left(\frac{400 - 125}{400} \right)$$

$$= 138.75^\circ$$

$$\simeq 139^\circ$$

PSG 7.68

$\therefore$ $F_d = 0.89$ (For V-V Belt)

$F_a = 1.3$

$$\therefore \text{Number of belts} = \frac{15 \times 1.3}{2.22 \times 0.94 \times 0.89}$$

$$= 10.5 \simeq 11 \text{ belts.}$$

7. Actual centre distance (C)

PSG 7.61

$$C = A + \sqrt{A^2 - B}$$

$$A = \frac{L}{4} - \frac{\pi(D+d)}{8} = \frac{1694}{4} - \frac{\pi(400+125)}{8}$$

$$A = 217.33$$

$$B = \frac{(D-d)^2}{8} = \frac{(400-125)^2}{8} = 9453.125$$

$$\therefore C = 217.33 + \sqrt{(217.33)^2 - (9453.125)}$$

$$C = 411.7 \text{ mm}$$

8. Dimensions of V-grooved pulley:-

PSG 7.70

For "B" crossection

pitch width, $l_p = 14$ mm

minimum distance of down to pitch line, $b = 4.2$ mm

Angle $A = 34^\circ$

minimum depth below pitch line $h = 10.8$ mm.

Centre to Centre distance of grooves $e = 19$ mm

Edge of pulley to first groove centre $f = 12.5$ mm

9. Sketch of V-belt drive :-

The number of belts are 11 for "B" section.

If we select "C" section the number of belts will be reduced (Refer PSG 7.70)

Results :-

1. Selection of Belt crosssection = "B"
2. pulley diameter (D) = 400mm (std)
3. Centre distance (C) = 400mm
4. Nominal pitch length $L = 1694$ mm
5. Maximum power transmitting capacity = 2.22 kW
6. Number of Belts calculation = 11 belts.
7. Actual Center distance $C = 411.7$ mm
8. Dimension of V. grooved pulley write l_p, b, A, h, e & f.
9. Sketch of V-belt drive.

(HW)

4. Two shafts whose centres are 1m apart are connected by a V-belt drive. The driving pulley is supplied with 100kW and has an effective diameter of 300mm. It runs at 1000rpm, while the driven pulley runs at 375 rpm. The angle of groove on the pulley is 40°. The permissible tension in 400 mm² crosssectional area of the belt is 2.1 Mpa. The density of the belt is 1100 kg/m³. Taking $\mu = 0.28$, estimate the number of belts required. Also calculate the length of each belt.

DESIGN PROCEDURE FOR WIRE ROPES

1. Selection of wire rope Types:- PSG 9.1
2. calculation of Design load

   Design load = W x maximum factor of safety

3. Selection of wire rope diameter (d):

   Refer PSG 9.5

4. calculation of Sheave (or) drum diameter (D)
5. Selection of cross-section Area (A):

   $$A = 0.4 \frac{\pi d^2}{4} \text{ in } mm^2 \quad \text{PSG } 9.1$$

6. calculation of wire diameter (d_w):-

   $$d_w = \frac{\text{Rope diameter (d)}}{1.5\sqrt{\text{No. of strands} \times \text{No. of wires per strand}}}$$

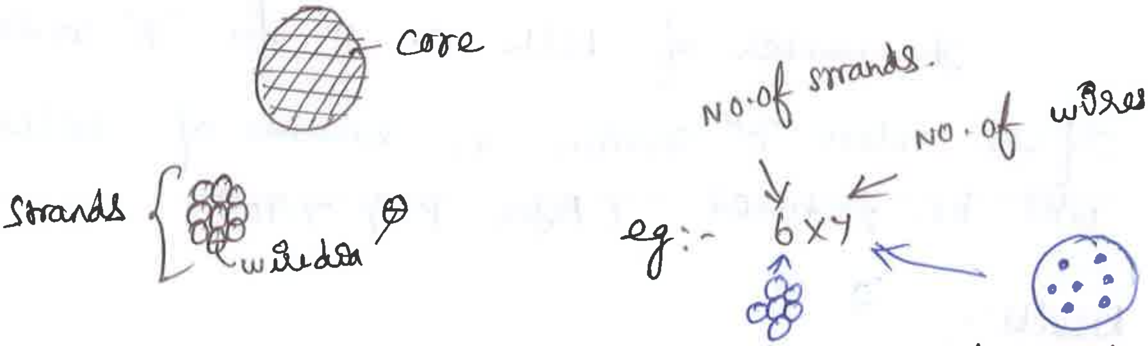
7. Effective (equivalent) load (W_{eq}):-

   $$W_{eq} = W_d + W_b + W_a \quad \text{Refer PSG } 9.1$$

8. Actual (or) working factor of safety

   $$= \frac{\text{Breaking load} \times \text{No. of ropes}}{\text{Equivalent / Effective load}}$$

Design of wire rope and pulleys :-



5. At the construction site, 1 tonne of steel is to be lifted up to a height of 20 m with the help of 2 wire ropes of 6 x 19 size, nominal diameter 12 mm, and breaking load 78 kN. Determine the factor of safety if the sheave diameter is 56d and if wire rope is suddenly stopped in 1 second when travelling at a speed of 1.2 m/s. What is the factor of safety if bending load is neglected. (NOV/DEC 14)

Given data :-

Load to be lifted $W = 1 \text{ tonne} = 1000 \text{ kg} = 1000 \times 9.81 = 9810 \text{ N}$

Height $h = 20 \text{ m}$

6 x 19 size 12 mm diameter rope = 2 no's

$D = 56d$; breaking load $= 78 \text{ kN} = 78000 \text{ N}$

Velocity $v = 1.2 \text{ m/s}$

Time $t = 1$ second ; bending load is neglected.

Soln:-

1. selection of wire rope type:- (not given in Question assume)

6x19 rope given — cranes & Hoists purpose

2. calculation of Design load:- (not reqd — because breaking strength is given) PSG 9.1

Design load = W x Maximum factor of safety

= 9810 N x 15

= 147150 N (or) 147.15 KN

3. selection of wire rope diameter (d):-

d = 12 (nominal diameter) — given

PSG 9.5 (6x19 group - page)

Note:- Breaking strength is not given consider Design load as a breaking load.

Approximate weight = 0.54 kgf/m

our given height = 20 m

∴ For 20M ⇒ W_r = 0.54 x 20

= 10.8 kgf

= 10.8 x 9.81

NOTE:- Hence to SI units.

[1 kgf = 9.81 N]
PSG 14.8

Approximate weight (W_r) = 106 N

4. calculation of sheave (or) drum diameter (D):-

D = 56 d (given)

∴ D = 56 x 12

= 672 mm

5. selection of cross-section Area (A):-

PSG 9.1

$$A = 0.4 \frac{\pi d^2}{4} = \frac{0.4 \times \pi \times (12)^2}{4} = 45.24 \text{ mm}^2$$

6. calculation of wire diameter (d_w):-

$$d_w = \frac{\text{Rope diameter (d)}}{1.5\sqrt{\text{no. of strands} \times \text{no. of wires per strand}}}$$

$$= \frac{12}{1.5\sqrt{6\times19}}$$

$$= 0.7493 \text{ mm}$$

7. Effective (equivalent) load, (W_{eq}):-

$$W_{eq} = W_d + W_b + W_a$$

Direct load (W_d) = $W + W_r$ = $9810 + 106$ = 9916 N.

Bending load (W_b) = $E' \times \frac{d_w}{D} \times A$

PSG 9.1

Corrected modulus of elasticity, $E' = 0.8\times10^6$ kgf/cm²

$$E' = 0.8\times10^6 \times 9.81\times10^{-2} = 78480 \text{ N/mm}^2$$

$$\therefore W_b = \frac{78480 \times 0.7493 \times 45.24}{672} = 3959 \text{ N}.$$

Accelaration load (W_a) = $\frac{W + W_r}{g} \times a$

Accelaration $a = \frac{\text{velocity}}{\text{time}} = \frac{V_2 - V_1}{t}$

Here initial velocity is not given [$\therefore V_1 = 0$] [$V_2 = 12$ m/s² given]

$$\therefore a = \frac{12 - 0}{1} = 12 \text{ m/s}^2$$

$$a = 1.2 \text{ m/s}^2$$

$$\therefore W_a = \frac{W + W_r}{g} \times a \Rightarrow \frac{(9810 + 106)1.2}{9.81} = 1213 \text{ N}$$

$\therefore$ Equivalent load, $W_{eq} = 9916 + 3959 + 1213$

$$= 15088 \text{ N}$$

8. Actual (or) working factor of safety:-

$$= \frac{\text{Breaking load} \times \text{No. of ropes}}{\text{Equivalent / effective load}}$$

$$= \frac{78000 \times 2}{15088}$$

$$= 10.34$$

Here the bending load is neglected (given)

$$\therefore W_{eq} = W_d + \cancel{W_b}^{\text{neglected}} + W_a$$

$$= 9916 + 1213$$

$$= 11129 \text{ N}$$

$$\therefore \text{working factor of safety} = \frac{\text{Breaking load} \times \text{No. of ropes}}{\text{Equivalent load (without bending load)}}$$

$$= \frac{78000 \times 2}{11129}$$

$$= 14.02$$

So our design is safe (Maximum Factor of safety = 15)

RESULTS:-

1. Selection of wire rope = 6 X 19 (Cranes & Hoists)
2. Calculation of Design load = 147.15 KW
3. Selection of wire rope diameter (d) = 12 mm
4. Calculation of Sheave (or) drum diameter (D) = 672 mm
5. Selection of cross sectional area (A) = 45.24 mm²
6. Calculation of wire diameter (dw) = 0.7493 mm
7. Effective load (W_{eq}) = 15088 N
8. Actual (or) working factor of safety = 10.34

DESIGN PROCEDURE FOR CHAIN AND SPROCKETS :-

1. Type of chain

2. preferred Transmission ratio (i)

$$i = \frac{N_2}{N_1} = \frac{Z_1}{Z_2} \qquad \text{PSG } 7.74$$

3. Number of Teeth (Z) :-
PSG 7.74

4. STANDARD PITCH (p)
Based on Centre distance (a) find the standard pitch (p) Refer PSG 7.74 & 7.72

5. Minimum factor of Safety (n)
Refer PSG 7.77

$$N = \frac{QV}{102\, n\, k_s} \text{ in kW}$$

$$Q = \frac{N \times 102 \times n \times k_s}{V} \text{ in kgf}$$

6. Selection of chain
Refer PSG 7.72

7. Check for actual factor of safety [n]

$$[n] = \frac{Q}{\Sigma p} \qquad \text{PSG } 7.78$$

8. Check for bearing stress (σ)

PSG 7.77

$$N = \frac{\sigma A V}{102\, K_s} \text{ in kW}$$

$$\sigma = \frac{N \cdot 102 \cdot K_s}{A V} \text{ in kgf/cm}^2$$

9. Actual length of chain (l):

PSG 7.75 $\quad l = l_p \cdot p$

where $$l_p = 2a_p + \frac{z_1+z_2}{2} + \frac{\left(\frac{z_2-z_1}{2\pi}\right)^2}{a_p}$$

$$a_p = \frac{a_0}{p}$$

10. Exact centre distance (a)

PSG 7.75

$$a = \frac{e + \sqrt{e^2 - 8m}}{4}\, p$$

$$e = l_p - \left(\frac{z_1+z_2}{2}\right) \; ; \quad m = \left(\frac{z_2-z_1}{2\pi}\right)^2$$

11. Chain wheel profile dimensions (diameter)

PSG 7.78

dia of small sprocket, $$d_1 = \frac{p}{\sin\left(\frac{180}{z_1}\right)} \text{ in mm}$$

dia of Big sprocket, $$d_2 = \frac{p}{\sin\left(\frac{180}{z_2}\right)} \text{ in mm}$$

Design of chain and sprockets:-

Types:- 1. Roller chain drive (problem comes)

2. Bushed " " (")

3. Silent " " (inverted tooth)

4. Link chain / welded chain drive.

6. A bucket elevator is to be driven by a geared motor and roller chain drive with the information given below:

(a) motor output = 3 KW

(b) Speed of motor shaft = 100 rpm

(c) Elevator drive shaft speed = 42 rpm

(d) load = even

(e) distance between centres of sprockets approximately = 1.2 m

(f) period of operation = 16 hours/day

(g) Geared motor is mounted on an auxiliary bed for centre distance adjustments.

(h) Design the chain drive [N/D - 2016]

Given data:-

power (P) = N = 3 KW

Motor (driver pinion) speed N_2 = 100 rpm.

Elevator drive (driven) N = 42 rpm.

Load evenly - 16 hours/day ; adjustment bed

Centre distance a = C = 1200 mm = 1.2 m

Solution:-

1. Type of chain

Roller chain

2. preferred transmission ratio (i):-

PSG 7.74

$$i = \frac{N_2}{N_1} = \frac{z_1}{z_2} \; ; \; i = \frac{100}{42} = 2.38 \simeq 2.4$$

3. Number of Teeth (z):-

PSG 7.74

For $i = 2.4$

No. of teeth on sprocket $z_1 = 27 - 25$

$\therefore z_1 = 25$ (selected)

$$\therefore i = \frac{z_2}{z_1} \Rightarrow z_2 = i z_1 = 25 \times 2.4 = 60$$

4. Standard pitch (p)

PSG 7.74

optimum centre distance, $a = (30 \text{ to } 50)p$

a - centre distance mm, p - pitch of chain mm,

[WKT $a = 1200$mm given]

$$a = 40p \Rightarrow p = \frac{a}{40} = \frac{1200}{40} = 30\text{mm}$$

$\therefore$ PSG 7.72 (in pitch column)

The next standard pitch value, $p = 31.75$mm.

5. Minimum factor of safety (n)

PSG 7.77

Power transmitted on the basis of breaking load.

$$N = \frac{QV}{102 \, n \, K_s} \text{ in kW}$$

$$Q = \frac{N \times 102 \times n \times K_s}{V} \text{ in kgf}$$

N – power = 3 Kw PSG 7.77 (Heading)
Service factor (K_s) PSG 7.77

$K_s = K_1 \cdot K_2 \cdot K_3 \cdot K_4 \cdot K_5 \cdot K_6$

PSG 7.76

$K_1 = 1.25$ (mild shocks)

$K_2 = 1.0$ (Adjustable supports)

$K_3 = 1.0$ ($a_p = (30 \text{ to } 50)P$)

PSG 7.77

$K_4 = 1.25$ (Inclination of sprocket Horizontal upto 60° & more than 60°)

$K_5 = 1.0$ (Drop lubrication)

$K_6 = 1.25$ (Double shift (or) 16 hours a day)

$\therefore K_s = 1.25 \times 1 \times 1 \times 1.25 \times 1 \times 1.25 = 1.95 \simeq 2$ (say)

velocity of chain (v) = $\dfrac{P Z_1 N_2}{60 \times 1000}$ (or) $\dfrac{P Z_2 N_1}{60 \times 1000}$ in m/s

$$v = \frac{31.75 \times 25 \times 100}{60 \times 1000} = 1.32 \text{ m/s}$$

PSG 7.77

Factor of safety $\therefore n = 7.8$ (For $z_1 = 25$; $N_1 = 100$ rpm)

$$\therefore Q = \frac{N \times 102 \times n \times K_s}{v} \text{ in kgf}$$

$$Q = \frac{3 \times 102 \times 7.8 \times 2}{1.32} = 3616 \text{ kgf}$$

6. Selection of chain :-

For pitch $p = 31.75$ mm and ~~Bearing~~ load (min) $= 3616$ kgf ;

PSG 7.72

Chain No 20A-1 R100 is selected.

Std (or) corresponding breaking load ; $Q = 8850$ kgf

weight per metre $= 3.8$ kgf

Bearing Area $A = 2.62$ cm^2

7. Check for actual factor of safety [n]

PSG 7.78

$$[n] = \frac{Q}{\Sigma p}$$

Q - breaking load (Std) $= 8850$ kgf

$$\Sigma p = P_t + P_c + P_s$$

Tangential force due to power transmission $P_t = \frac{102N}{V}$ if N in kW

$$= \frac{102 \times 3}{1.32} = 232 \text{ kgf}$$

centrifugal tension, $P_c = \frac{wv^2}{g} = \frac{3.8 \times (1.32)^2}{9.81} = 0.675$ kgf

Tension due to sagging of chain, $P_s = K.w.a$

PSG 7.78 co-efficient of sag $K = 2$ (assume more than $40°$)

Centre distance $a = 1.2$ m (given data)

$$\therefore P_s = K.w.a = 2 \times 3.8 \times 1.2$$

$$= 9.12 \text{ kgf}$$

$$\therefore \Sigma p = P_t + P_c + P_s = 232 + 0.675 + 9.12$$

$$= 242 \text{ kgf}$$

$$\therefore [n] = \frac{Q}{\Sigma p} = \frac{8850 \cancel{kgf}}{242 \cancel{kgf}} = 36.57$$

Actual factor of safety (36.57) is greater than assumed factor of safety (7.8)

The design is safe.

8. Check for bearing stress(σ)

PSG 7.77

power transmitted on the basis of allowable bearing stress.

$$N = \frac{\sigma A V}{102 K_s} \text{ in KW} ; \sigma = \frac{N \cdot 102 \cdot K_s}{A V} \text{ in kgf/cm}^2$$

$$\therefore \sigma = \frac{3 \times 102 \times 2}{2.62 \times 1.32} = 177 \text{ kgf/cm}^2 = \frac{177 \times 9.81}{100}$$

$$= 17.4 \text{ N/mm}^2$$

PSG 7.77

$[\text{For } K_s = 1 ; z_1 = 25 ; N_1 < 200]$

$\sigma = 3.15 \text{ N/mm}^2$

So the design is safe.

9. Actual length of chain l:

PSG 7.75 $\quad l = l_p \cdot p$

$$l_p = 2a_p + \frac{z_1+z_2}{2} + \frac{\left(\frac{z_2-z_1}{2\pi}\right)^2}{a_p}$$

$$a_p = \frac{a_0}{p} = \frac{1200}{31.75} = 38$$

(To be even number)

$$l_p = (2\times38) + \frac{25+60}{2} + \frac{\left(\frac{60-25}{2\pi}\right)}{38}$$

$$l_p = 76 + 42.5 + 0.147 = 118.6 \simeq 118 \text{ mm}$$

$$l = l_p \cdot p = 118 \times 31.75 = 3778 \text{ mm}$$

10. Exact centre distance (a) :-

PSG 7.75

$$a = \frac{e + \sqrt{e^2 - 8m}}{4}\, p$$

$$e = l_p - \left(\frac{z_1+z_2}{2}\right) = 118 - \left(\frac{25+60}{2}\right) = 76.5 \simeq 77$$

$$m = \left(\frac{z_2-z_1}{2\pi}\right)^2 = \left(\frac{60-25}{2\pi}\right)^2 = 31$$

$$a = \frac{77 + \sqrt{(77)^2 - (8\times31)}}{4} \times 31.75$$

$$a = 1209 \text{ mm (or) } 1.209 \text{ m}$$

11. Chain wheel profile dimensions (diameter)

PSG 7.78

$$\text{Diameter of small sprocket, } d_1 = \frac{p}{\sin\left(\frac{180}{z_1}\right)} = \frac{31.75}{\sin\left(\frac{180}{25}\right)}$$

$$d_1 = 253 \text{ mm.}$$

$$\text{Diameter of large sprocket, } d_2 = \frac{p}{\sin\left(\frac{180}{z_2}\right)} = \frac{31.75}{\sin\left(\frac{180}{60}\right)}$$

$$d_2 = 607 \text{ mm.}$$

PSG 7.72

chain NO : 20 A-1 R100

pitch $p = 31.75$ mm

Roller diameter $D_r = 19.05$ mm

Width between innerplate $W_{min} = 19.10$ mm

Results :-

1. Type of chain = Roller chain.
2. preferred Transmission ratio (i) = 2.4
3. Number of Teeth (z_1) = 25 ; $z_2 = 60$
4. Standard pitch (p) = 31.75 mm
5. Minimum factor of safety (n) = 7.8
6. Selection of chain = 20 A-1 R100
7. Check for actual factor of safety [n] = 36.57
8. Check for bearing stress (σ) = 3.15 kgf/mm² (safe)
9. Actual length of chain (l) = 3778 mm
10. Exact centre distance (a) = 1.209 m
11. Chain wheel profile dimension (diameter)

$d_1 = 253$ mm ; $d_2 = 607$ mm.

Homework

7. The transporter of a heat treatment furnace is driven by a 4.5 kW, 1440 rpm, induction motor through a chain drive with a speed reduction ratio of 2.4. The transmission is Horizontal with bat type of lubrication. rating is continuous with 3 shifts per day. Design the complete chain drive

(NOV/DEC 17) (15 marks)

UNIT–II

DESIGN PROCEDURE FOR SPUR GEAR

1. Gear ratio i

2. Material selection (pinion & wheel)

3. Gear life

4. Calculation of design torque (or) twisting moment (M_t)
   PSG 8.15

5. Calculation of E_q, σ_b & σ_c

Equivalent young's modulus, $E_{eq} = \frac{2E_1E_2}{E_1+E_2}$ PSG 8.13A

Other wise based on the material we take value 8.14 PSG

Bending stress $\sigma_b = \frac{1.4\,k_{bl}}{n \cdot k_\sigma}\sigma_{-1}$ PSG 8.18

(pinion & wheel)

otherwise based on the material we take value PSG 8.5

Compressive stress $\sigma_c = C_R\,HRC\,k_{cl}$ kgf/cm^2 PSG 8.16

(pinion & wheel) otherwise we take PSG 8.5

6. Calculation of center distance (min) a :-

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.74}{[\sigma_c]_{min}}\right)^2 \frac{E[M_t]}{i\psi}}$$ in cm PSG 8.13

7. Selection of z_1 & z_2

8. Calculation of module (m)

PSG. 8.22

$$m = \frac{2a}{z_1+z_2}$$ ~~PSG 8.13A~~

9. Revised center distance (a)

$$a = \frac{m(z_1+z_2)}{2}$$ PSG 8.22

10. calculation of diameter (d, D) & facewidth (b) :-

PSG 8.22

$d = z_1 m$ = pinion diameter for pitch

pitch diameter of wheel (D) = $z_2 m$ in mm

Face width (b) = $\psi . a$ in mm

11. check for compressive stress (σ_c)

PSG 8.13

$$\sigma_c = 0.74 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{ib} E [M_t]} \leq [\sigma_c]$$

12. check for Bending stress (σ_b) :-

PSG 8.13A

$$\sigma_b = \frac{i \pm 1}{a m b y} [M_t] \leq \sigma_b$$

13. calculation of other parameters.

UNIT–II

DESIGN OF SPUR GEAR:- (i) Normal Method ⊗

(ii) using Lewis and Buckingham equations.

$$\text{Dynamic load } (F_d) = F_t + F_i$$

$$= \frac{P}{V} + \frac{21V(Cb + F_t)}{21V + \sqrt{Cb + F_t}}$$

* power transfer pinion to Gear (wheel)
(N1) (N2)

1. Design a spur gear drive to transmit 22 kW at 900 rpm, speed reduction is 2.5, Materials for pinion and wheel are C15 steel and cast iron grade 30 respectively. Take pressure angle of 20° and working life of the gears as 10,000 hours.

Given data:-

Power (P) = 22 kW

Speed N_1 = 900 rpm

Speed ratio i = 2.5

pinion material = C15 steel

wheel material = Cast Iron (CI) grade 30

pressure angle $\alpha = 20°$

Life = 10,000 hours

Design a spur gear drive

Solution:-

1. Gear ratio (i)

$i = 2.5$

2. Material selection:-

pinion – C15 steel

wheel – CI 30

3. Gear life:

10,000 hours

4. Calculation of Design torque (or)

Design twisting moment (M_t)

PSG 8.15

$[M_t] = M_t \, k_d \, k$ kgf.cm

PSG 8.15

$M_t = 97420 \cdot \frac{kW}{n} \Rightarrow 97420 \times \frac{22}{900}$ (n- pinion speed $= N_1$)

PSG 8.15

$k_d \cdot K = 1.3$ (assume initially)

$\therefore [M_t] = 97420 \times \frac{22}{900} \times 1.3 = 3104$ kgf - cm

5. calculation of E_{eq}, σ_b and σ_c :-

pinion & wheel are different material

PSG 8.13A

Equivalent young's modulus, $E_{eq} = \frac{2 E_1 E_2}{E_1 + E_2}$

Note:- For σ_b & σ_c of the pinion & wheel material are not available in PSG 8.5. so we proceed the formula

PSG 8.14 $E_1 = 2.15 \times 10^6$ kgf/cm²

$E_2 = 1.4 \times 10^6$ kgf/cm²

$$E_{eq} = \frac{2 \times (2.15 \times 10^6)(1.4 \times 10^6)}{(2.15 \times 10^6) + (1.4 \times 10^6)} = 1.75 \times 10^6 \text{ kgf/cm}^2$$

Bending stress

PSG 8.18

$$[\sigma_b]_{pinion} = \frac{1.4 \, k_{bl}}{n \cdot k_\sigma} \sigma_{-1}$$

PSG 8.17

$= 60 \, nT$

Life cycle (N) $= 60 \times N_1 \times$ hours

$= 60 \times 900 \times 10000$

$= 54 \times 10^7$ cycles

PSG 8.20 $k_{bl} = 1$ [life cycle $\geq 10^7$]

PSG 8.19 $k_\sigma = 1.2$ [Assume steel case Hardened]

PSG 8.19 $n = 2.0$ [Tempered (or) Normalized]

" $\sigma_{-1} = 0.25 (\sigma_u + \sigma_y) + 500$ kgf/cm²

PSG 1.9 pinion C15 steel

Tensile $\sigma_u = 40$ kgf/mm² $= 4000$ kgf/cm²

Yield $\sigma_y = 24$ kgf/mm² $= 2400$ kgf/cm²

$\sigma_{-1} = 0.25(4000 + 2400) + 500 \quad kgf/cm^2$

$= 2100 \ kgf/cm^2$

$[\sigma_b]_{pinion} = \frac{1.4 K_{bl}}{n \cdot K_\sigma} \sigma_{-1} = \frac{1.4 \times 1}{2 \times 1.2} \times 2100$

$= 1225 \quad kgf/cm^2$

$||^{ly}$

$[\sigma_b]_{wheel} = \frac{1.4 K_{bl}}{n \cdot K_\sigma} \sigma_{-1} \quad kgf/cm^2$

cast IRON

PSG 8.20 $\quad K_{bl} = \sqrt[9]{\frac{10^7}{N}} = \sqrt[9]{\frac{10^7}{54 \times 10^7}} = 0.6 \simeq 1$

PSG 8.19 $\quad K_\sigma = 1.2$ (cast iron)

$n = 2.0$ $\quad$ PSG 1.5 ISI GRADE 30

$\sigma_{-1} = 0.45 \sigma_u$ $\quad [\sigma_u = 300 \ N/mm^2 = 3000 \ kgf/cm^2]$

$\sigma_{-1} = 0.45 \times 3000 = 1350 \ kgf/cm^2$

$\therefore [\sigma_b]_{wheel} = \frac{1.4 \times 1}{2 \times 1.2} \times 1350$

$= 788 \quad kgf/cm^2$

Compressive Stress

PSG 8.16

$[\sigma_c]_{pinion} = C_R \ HRC \ K_{cl} \quad kgf/cm^2$

PSG 8.16 $\quad C_R = 220; \ HRC = 60, \ K_{cl} = 0.585$

$= 220 \times 60 \times 0.585 = 7722 \ kgf/cm^2$

$[\sigma_c]_{wheel} = C_R \ HRC \ K_{cl} \quad kgf/cm^2$

PSG 8.16 $\quad C_B$ (or) $C_R = 23; \ HRC = 250;$

$K_{cl} = \sqrt[6]{\frac{10^7}{N}}$

PSG 8.17 $\quad N = 60 \ nT = 60 \times 900 \times 10000$

$= 54 \times 10^7$

$K_{cl} = \sqrt[6]{\frac{10^7}{54 \times 10^7}} = 0.82$

$[\sigma_c]$ wheel $= C_R \, HRC \, K_{cl}$ kgf/cm^2

$= 23 \times 250 \times 0.82$

$= 4715$ kgf/cm^2

6. Calculation of center distance (min) a :-
PSG 8.13 (For spur)

$$a \geq (i+1) \sqrt[3]{\left(\frac{0.74}{[\sigma_c]_{min}}\right)^2 \frac{E[M_t]}{i\psi}}$$

$$a \geq (2.5+1) \sqrt[3]{\left(\frac{0.74}{4715}\right)^2 \frac{1.75\times10^6[3104]}{2.5 \times 0.3}}$$

Note
For single material for pinion & wheel means σ_c only not min

PSG 8.14
$\psi = 0.3$

$$a \geq (3.5) \sqrt[3]{(2.46\times10^{-8})\times(72427\times10^5)} \Rightarrow a \geq 14\text{cm}$$

$a \geq 140$ mm

7. Selection of z_1 & z_2 :-

Assume $z_1 = 17$ WKT $i = \frac{z_2}{z_1}$ $\therefore z_2 = i z_1$

$z_2 = 2.5 \times 17 = 42.5 = 43$

8. Calculation of Module (m) PSG 8.22

$$m = \frac{2a}{z_1+z_2} = \frac{2\times140}{17+43} = \frac{280}{60}$$

$m = 5$ mm

9. Revised center distance (a) :- PSG 8.22

$$a = \frac{m(z_1+z_2)}{2} = \frac{5(17+43)}{2} = 150\text{mm}$$

10. Calculation of diameter (d, D) & face width (b) :-
PSG 8.22

pitch diameter of pinion (d) $= z_1 m = 17 \times 5$

$= 85$ mm.

" " " wheel (D) $= z_2 m = 43 \times 5 = 215$ mm

Face width (b) $= \psi . a$

$= 0.3 \times 150$

$= 45$ mm.

11. Check for compressive stress :- (σ_c)

PSG 8.13 (For spur gear)

$$\sigma_c = 0.74 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{i b} E [M_t]} \leq [\sigma_c]$$

[consider +ve]

$a = 150 \text{ mm} = 15 \text{ cm}$

$i = 2.5$

$b = 45 \text{ mm} = 4.5 \text{ cm}$

$E = 1.75 \times 10^6 \text{ kgf/cm}^2$

$[M_t] = 3104 \text{ kgf-cm}$

$$\sigma_c = 0.74 \frac{2.5+1}{15} \sqrt{\frac{2.5+1}{2.5 \times 4.5} \times 1.75 \times 10^6 \times 3104}$$

$$\sigma_c = 7118 \text{ kgf/cm}^2 \leq 7722 \text{ kgf/cm}^2$$

So our design is safe.

12. Check for Bending stress :- (σ_b)

PSG 8.13A (for spur gear)

$$\sigma_b = \frac{i \pm 1}{a m b y} [M_t] \leq \sigma_b$$ [consider +ve]

$$= \frac{2.5+1}{15 \times 0.5 \times 4.5 \times 1} [3104] \leq \sigma_b$$

[M = 5 mm = 0.5 cm]

[assume deflection y = 1]

$$= 322 \text{ kgf/cm}^2 \leq 1225 \text{ kgf/cm}^2$$

∴ So our design is safe

Tip circle dia of pinion $= (d + 2\,\text{module})$
$= (85 + (2 \times 5)) = 95\,mm$

Tip circle dia of gear $= (D + 2\,\text{module})$
$= (215 + (2 \times 5)) = 225\,mm$

Root circle dia of pinion $= (d - 2m)$
$= (85 - (2 \times 5)) = 75\,mm$

Root circle dia of gear $= (D - 2m)$
$= (215 - (2 \times 5)) = 205\,mm$

Results:-

1. Gear ratio $i = 2.5$
2. Material selection pinion – C15 steel
   wheel – CI 30
3. Gear life = 10,000 hrs
4. calculation of Design torque $(M_t) = 3104$ kgf·cm
5. calculation of E_{eq}, σ_b & σ_c ⇒ 4715 kgf/cm²
   1.75×10^6 kgf/cm², 788 kgf/cm²
6. calculation of center distance (min) $a \geqq 140\,mm$
7. Selection of $z_1 = 17$, $z_2 = 43$.
8. calculation of module $(m) = 5\,mm$.
9. Revised center distance $(a) = 150\,mm$
10. calculation of $d = 85\,mm$; $D = 215\,mm$; $b = 45\,mm$
11. check for σ_c is safe, 12. check for σ_b is safe.

Home work

(2) In a spur gear drive for a stone crusher the gear are made of C40 steel. The pinion is transmitting 30 kW at 1200 rpm. The gear ratio is 3. Gear is to work 8 hours per day, six days in a week and for 3 years. Design the drive.

Gear life $= 8 \times 52 \times 6 \times 3 = 7488$ hrs

DESIGN PROCEDURE FOR HELICAL GEAR

1. Gear ratio i
2. Material selection for pinion and gear
3. Gear life
4. Calculation of design torque (or) Twisting moment $[M_t]$

PSG 8.15

$$[M_t] = M_t \cdot k_d \cdot k \quad \text{kgf.cm}$$

5. Calculation of E_{eq}, σ_b, and σ_c:-

Equivalent young's modulus, $E_{eq} = \dfrac{2E_1 E_2}{E_1+E_2}$

Otherwise we take the value based on the material Refer Page.No: 8.14 at PSG

The procedure σ_b & σ_c is same as spur gear.

6. Calculation of center distance (a)

PSG 8.13

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.7}{[\sigma_c]}\right)^2 \frac{E\,[M_t]}{i\,\psi}} \quad \text{in cm}$$

7. Calculation of module (m_n) (or) (m)

PSG 8.13A

$$m = m_n = 1.15\cos\beta \sqrt[3]{\frac{[M_t]}{Y_v\,[\sigma_b]\,\psi_m\,z_1}} \quad \text{in cm}$$

8. Calculation of z_1 & z_2

$$z_1 = \frac{2a\cos\beta}{m_n(i+1)} \quad \text{PSG 8.22}$$

9. Calculation of diameter (d, D)

PSG 8.22

pitch diameter of pinion $d = \frac{m_n}{\cos\beta} z_1$ in cm

pitch diameter of wheel (D).

WKT $i = \frac{D}{d} \Rightarrow D = i d$ in cm.

10. Revised centre distance (a)

$$a = \frac{D+d}{2} \text{ in cm.}$$

11. Check for compressive stress (σ_c)

PSG 8.13

$$\sigma_c = 0.7 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{i b} E [M_t]} \leq [\sigma_c]$$

12. Check for Bending stress (σ_b)

PSG 8.13A

$$\sigma_b = 0.7 \frac{i \pm 1}{a b m_n y_v} [M_t] \leq [\sigma_b]$$

13. Calculation of other parameters.

Design of Helical gears

(i) Normal method (or) Based on life of gears

(ii) using lewis & Buckingham equations.

Dynamic load $(F_d) = F_t + F_i$

$$= \frac{P}{V} + \frac{21V(Cb+F_t)}{21V+\sqrt{Cb+F_t}}$$

Helix angle (spiral angle) (β):-

It is the angle between the tooth axis and the plane containing wheel axis

8 to 25° ⇒ Helical gears. (general take 15°)

25° to 40° ⇒ Herringbone gears. (general take 20°)

Note :- power given in hp ⇒ convert into kW

1 hp = 746 watts [PSG 14.8]

(3). Design a pair of helical gear drive to transmit the power of 10 kW at 1000 rpm of the pinion. reduction ratio of 5 is required, give details of the drive in the tabular column form.

Given data:-

power = 10 kW

pinion speed N_1 = 1000 rpm

Speed ratio $i = 5$

$\therefore i = \frac{N_1}{N_2}$; $N_2 = \frac{N_1}{i} = \frac{1000}{5} = 200$ rpm

Helix angle = 15° assume

Design a helical gear

1. Gear ratio

$i = 5$

2. Material selection:

PSG 8.5

Assume let the pinion and gear material as

40 Ni2 Cr1 MO 28 (steel)

$[\sigma_c] = 11000 \text{ kgf/cm}^2$

$[\sigma_b] = 4000 \text{ kgf/cm}^2$

3. Gear life

Not given.

4. Calculation of design torque (or) Design twisting moment (M_t)

PSG 8.15

$[M_t] = M_t \cdot K_d \cdot K$ kgf.cm

$M_t = 97420 \times \frac{kW}{n} = 97420 \times \frac{10}{1000}$ (n - pinion speed $= N_1$)

$K_d \cdot K = 1.3$ (assume initially)

$[M_t] = 97420 \times \frac{10}{100} \times 1.3 = 1266$ kgf.cm

5. Calculation of E_{eq}, σ_b and σ_c :-

we assumed pinion and wheel are same material ∴ 40Ni2 Cr1 MO 28 (steel)

∴ PSG 8.14 $E_{eq} = 2.15 \times 10^6 \text{ kgf/cm}^2$

PSG 8.5

$[\sigma_c] = 11000 \text{ kgf/cm}^2$

$[\sigma_b] = 4000 \text{ kgf/cm}^2$

6. Calculation of center distance (a):-

PSG 8.13 for helical gear

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.7}{[\sigma_c]}\right)^2 \frac{E\,[M_t]}{i\,\psi}}$$

PSG 8.14 ; $\psi = 0.5$ (Assume)

$$a \geq (5+1)\sqrt[3]{\left(\frac{0.7}{11000}\right)^2 \frac{2.15\times10^6 \times 1266}{5\times0.5}} \Rightarrow a \geq 9.84\,cm$$

$a \geq 98.4\,mm$ say $a \geq 98\,mm$

7. Calculation of Module (m_n) (or) (m)

PSG 8.13A for helical gears

$$m = m_n \geq 1.15\cos\beta\sqrt[3]{\frac{[M_t]}{y_v\,[\sigma_b]\,\psi_m\,z_1}} \text{ in cm}$$

Assume $z_1 = 20$

PSG 8.14 $\psi_m = \frac{b}{m_n (or)\, m} = 10$ (assumed) $[m = m_n]$

$y_v = 0.4$ (assume)

$$m_n \geq 1.15\cos 15\sqrt[3]{\frac{1266}{0.4\times4000\times10\times20}}$$

$m_n \geq 0.175\,cm = 1.75\,mm$

Say $m_n = 0.2\,cm$ (or) $2\,mm$

8. Calculation of z_1 & z_2 :- PSG 8.22

$$z_1 = \frac{2a\cos\beta}{m_n(i+1)} = \frac{2\times9.8\times\cos 15}{0.2\,(5+1)} = 16$$

WKT $i = \frac{z_2}{z_1}$ $\therefore z_2 = i z_1$

$= 5\times16$

$= 80$

9. Calculation of diameter (d, D) :-

PSG 8.22

pitch diameter of pinion (d) = $\frac{m_n}{\cos\beta} z_1$

$$d = \frac{0.2 \times 16}{\cos 15^\circ} = 3.3\,cm$$

pitch diameter of wheel (D) :

WKT $i = \frac{D}{d} \Rightarrow D = i\,d = 5 \times 3.3$

$= 16.5\,cm$

10. Revised centre distance (a)

$$a = \frac{D+d}{2} = \frac{16.5 + 3.3}{2} = 9.9\,cm$$

11. Check for compressive stress (σ_c) :-

PSG 8.13 (For helical gears)

$$\sigma_c = 0.7 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{i b} E [M_t]} \leq [\sigma_c]$$

(consider +ve)

$a = 9.9\,cm$

$i = 5$

PSG 8.22

$b = 5\,cm$ [$b = \psi \cdot a = 0.5 \times 9.9 = 4.95 \simeq 5\,cm$]

$E = 2.15 \times 10^6\ kgf/cm^2$

$[M_t] = 1266\ kgf.cm$

$$\sigma_c = 0.7 \frac{5+1}{9.9} \sqrt{\frac{(5+1)}{5 \times 5} \times 2.15 \times 10^6 \times 1266} \leq [\sigma_c]$$

$\sigma_c = 10643\ kgf/cm^2 \leq 11000\ kgf/cm^2$

So our design is safe.

12. Check for Bending stress (σ_b)

PSG 8.13 A (for helical gear)

$$\sigma_b = 0.7 \frac{i \pm 1}{a b m_n y_v} [M_t] \leq [\sigma_b]$$

$$= 0.7 \frac{(5+1) \times 1266}{9.9 \times 5 \times 0.2 \times 0.4} \leq [\sigma_b]$$

$$= 1425 \text{ kgf/cm}^2 < 4000 \text{ kgf/cm}^2$$

So our design is safe.

13. calculation of other parameters:-

Addendum = module m_N = 0.2 cm = 2 mm

Dedendum = $1.25 \times m_N = 1.25 \times 2 = 2.5$ mm

Tip circle diameter pinion = $d_1 + (2 \times \text{addendum}) = 33 + (2 \times 2) = 37$ mm

" " " wheel = $D + (2 \times \text{addendum}) = 165 + (2 \times 2) = 169$ mm

ROOT " " pinion = $d - (2 \times \text{dedendum}) = 33 - (2 \times 2) = 28$ mm

" " " wheel = $D - (2 \times \text{dedendum}) = 165 - (2 \times 2) = 160$ mm

Results:- Tabulate the results procedure 1 to 12

HW

4. Design a helical gear drive to transmit the power of 20 hp. Speed ratio 6, pinion speed 1200 rpm helix angle is 25°.

pinion 15 Ni2Cr1Mo15

Gear C45

Design the gear drive.

Herringbone gear:-

It is the Double helical gear, the power transmitted by two helical gears. Hence for design, we should consider only half of the power.

The remaining procedure is same for helical gears.

Hw

5. Design a Herringbone gear for the data given below

power = 40 kW

pinion speed = 1800 rpm

Gear ratio = 4

Helix angle = 25°

Material used = C 45 steel.

Soln:-

Consider Half power of herring-bone $= \frac{40\,kW}{2}$

$= 20\,kW$

The remaining procedure is same for Helical gear.

UNIT III

PART B

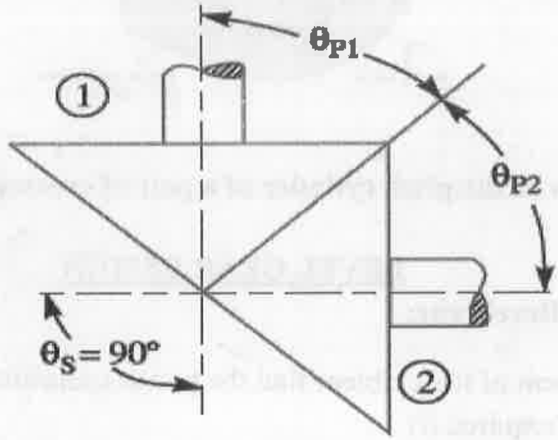
STRAIGHT BEVEL GEAR:

The bevel gears are used for transmitting power at a constant velocity ratio between two shafts whose axes intersect at a certain angle.

Classification of Bevel Gears

The bevel gears may be classified into the following types, depending upon the angles between the shafts and the pitch surfaces.

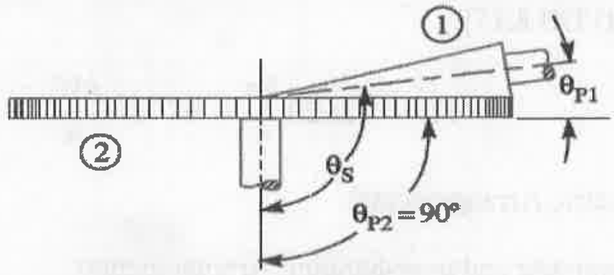
1. Mitre gears: When equal bevel gears (having equal teeth and equal pitch angles) connect two shafts whose axes intersect at right angle, as shown in Fig. then they are known as **mitre gears.**



Mitre gears.

2. Angular bevel gears: When the bevel gears connect two shafts whose axes intersect at an angle other than a right angle, then they are known as **angular bevel gears.**

3. Crown bevel gears: When the bevel gears connect two shafts whose axes intersect at an angle greater than a right angle and one of the bevel gears has a pitch angle of 90°, then it is known as a crown gear. The crown gear corresponds to a rack in spur gearing, as shown in Fig.



Crown bevel gear.

4. Internal bevel gears: When the teeth on the bevel gear are cut on the inside of the pitch cone, then they are known as **internal bevel gears.**

CROSS HELICAL GEAR

A pair of crossed-helical gears also known as spiral gears is used to connect and transmit motion between two non-parallel and non- intersecting shafts. As the contact between the mating teeth is always a point, these gears are suitable only for transmitting a small amount of power.

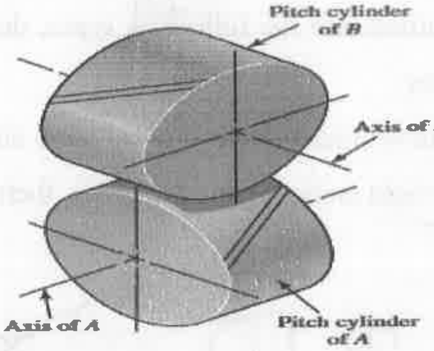


Fig View of the pitch cylinder of a pair of crossed helical gear

BEVEL GEAR DESIGN

Design Procedure for Bevel gear:

1. From the statement of the problem find the power transmitted (P), speed of driver (n_1)and speed reduction required (i)
2. Select the suitable materials and Select corresponding material related mechanical properties. [PSG DB 8.5]
3. Calculate the minimum cone distance (R). [PSG DB 8.13]

$$R \geq \Psi_y \sqrt{i^2+1}\sqrt[3]{\left(\frac{0.72}{(\Psi_y - 0.5)[\sigma_c]}\right)^2 \frac{E[M_t]}{i}}$$

Select the value of Ψ_y = R/b, based on gear ratio.[PSG DB 8.15, Table 13]

[Mt] = Mt ×k×k_d[PSG DB 8.15]

$$M_t = 71620\frac{hp}{n} = 97420\,\frac{kW}{n}$$

k.k_d = 1.3 (For symmetric Arrangements)

k.k_d = 1.5 (For unsymmetric and over-hanging Arrangements)

4. Calculate the minimum average module (m_{av}). [PSG DB 8.13A]

$$m_{av} = 1.28\sqrt[3]{\frac{[M_t]}{y_v[\sigma_b]\Psi_m Z_1}}$$

$\Psi_m = \frac{b}{m_{av}}$ =10 (Assume initially)

Assume Z_1 = 20 initially

Y_v = Form factor based on number of teeth [PSG DB 8.18]

5. Calculation of transverse module:

$m_t = m_{av} \times \frac{b}{Z}\, sin\delta$ [PSG DB 8.13A]

Standardize the transverse module. [PSG DB 8.2]

6. Correct the number of teeth of pinion [PSG DB 8.38]

$R = 0.5 \times m_t \times Z_1 \times \sqrt{i^2+1}$

$Z_1 = \frac{R}{0.5\times mt\times\sqrt{i^2+1}}$

$i = \frac{Z_2}{Z_1}$; $Z_2 = i \times Z_1$

7. Final cone distance:

$R = 0.5 \times m_t \times Z_1 \times \sqrt{i^2+1}$ [PSG DB 8.38]

8. Calculation of face width:

$b = \frac{R}{\Psi_y}$

Ψ_y = R/b . [PSG DB 8.15]

9. Check for compressive stress

$$\sigma_c = \frac{0.72}{(R-0.5b)}\sqrt{\frac{\sqrt{(i^2+1)^3}}{ib}E[M_t]} \le [\sigma_c] \quad [PSG\ DB\ 8.13]$$

10. Check for bending stress:

$$\sigma_b = \frac{R\sqrt{i^2+1}\times[M_t]}{(R-0.5b)^2\times b\times m\times y_v}\times\frac{1}{\cos\alpha}$$ [PSG DB 8.13A]

11. Other parameters of gear drives [PSG DB 8.38 & 8.39]

Tip diameters for pinion $d_{a1} = m_t(Z_1 + 2cos\delta_1)$

Tip diameters for gear $d_{a2} = m_t(Z_2 + 2cos\delta_2)$

Addendum angle $\theta_{a1} = \theta_{a2} = \tan^{-1}(\frac{m_t\times f_0}{R})$

Dedendum angle $\theta_{f1} = \theta_{f2} = \tan^{-1}(\frac{m_t \times (f_0 + c)}{R})$

Tip angle for pinion $\delta_{a1} = \delta_1 + \theta_{a1}$

Tip angle for Gear $\delta_{a2} = \delta_2 + \theta_{a2}$

Root angle for pinion $\delta_{f1} = \delta_{1'} - \theta_{f1}$

Root angle for Gear $\delta_{f2} = \delta_2 - \theta_{f2}$

Tooth height $h = h_a + h_f$

Working depth $h_w = 2m_t$

1. **Design a bevel gear drive to transmit 7 kW at 1600 rpm for the following data.**
   Gear ratio = 3
   Material for pinion and gear = C45 Steel
   Life = 10,000 hours
   (April/May 2010, R2004) (May/June 2013) (April/May 2009, R2004) (May/June 2014, 2018)

Solution:

1. **Calculation of minimum cone distance**

$$R \geq \Psi_y \sqrt{i^2 + 1} \sqrt[3]{\left(\frac{0.72}{(\Psi_y - 0.5)[\sigma_c]}\right)^2 \frac{E[M_t]}{i}}$$ [PSG DB 8.13]

Design Torque, $[M_t] = M_t \times k \times k_d$

Assume $k \times k_d = 1.3$ initially assumed

$$[M_t] = \frac{97420 \times kw \times k_d \times k}{n} = \frac{97420 \times 7 \times 1.3}{1600} = 554.07 \text{ kgf.cm}$$

For C45 steel, $E_{eq} = 2.15 \times 10^6$ kgf/cm^2 [PSG DB 8.16]

$\Psi_y = \frac{R}{b} = 3$ (Selection of Ψ_y value based on reduction ratio from PSG DB 8.15)

Design Compressive stress $[\sigma_c] = C_B\, HB\, K_{cl}$ **(OR)** $[\sigma_c] = C_R\, HRC\, K_{cl}$ [PSG DB 8.16]

$[\sigma_c]$ *For C45 steel:*

$C_R = 265$

HRC = 40 to 55 (Now Take HRC=50)

Life in number of cycles (N) = $60 n_1 T$ [PSG DB 8.17]

(Life in hours (T) = 10000 hours given)

N= 60×1600×10000 = 96× 10^7 cycles

K_{cl}= 0.585 (For steel N> 25× 10^7) [PSG DB 8.17 Table 17]

$[\sigma_c]$ $For\ Pinion\ \&\ Wheel = C_R\ HRC\ K_{cl} = 265 \times 50 \times 0.585 = \mathbf{7751.25}\boldsymbol{Kgf/cm^2}$

$$R \geq 3\sqrt{3^2+1}\sqrt[3]{\left(\frac{0.72}{(3-0.5)7751.25}\right)^2 \frac{2.15\times 10^6 \times 554.07}{3}}$$

R≥ 7.8 cm (or) 78 mm

2. Calculation of Average module

$$m_{av} = 1.28\sqrt[3]{\frac{[M_t]}{y_v[\sigma_b]\Psi_m Z_1}} \qquad \text{[PSG DB 8.13A]}$$

$\Psi_m = \frac{b}{m_{av}}$ =10 (initially assumed)

Assume Z_1 = 20 (initially assumed)

Y_v = Form factor =0.4

$$[\sigma_b] = \frac{1.4K_{bl}}{nK_\sigma}\sigma_{-1} \qquad \text{[PSG DB 8.18]}$$

$$\sigma_{-1} = 0.25\left(\sigma_u + \sigma_y\right) + 500 \qquad \text{[PSG DB 8.19 Table 19]}$$

$\sigma_{-1} = 0.25\ (6300 + 3600) + 500 = 2975\ Kgf/cm^2$ (σ_u, σ_y $from\ PSG\ DB$ 1.9)

$K_{bl} = 0.7$ → PSG 8-20

PSG 8·19 Table 21

$K_\sigma = 1.2$ n = 2.0 ← PSG 8·19 Table 20

$$[\sigma_b] = \frac{1.4 \times 0.7}{2 \times 1.2} \times 2975 = \mathbf{1214.79}\ \boldsymbol{Kgf/cm^2}$$

$$[\sigma_b] = \mathbf{1214.79}\ \boldsymbol{Kgf/cm^2}$$

$$m_{av} = 1.28\sqrt[3]{\frac{554.07}{0.4\times1214.79\times10\times20}}$$

$\boldsymbol{m_{av}}$**≥ 0.228 cm (Or) 2.28 mm**

3. **Calculation of transverse module**

$m_t = m_{av} \times \frac{b}{z}\ sin\delta\ (PSG\ 8.13A) = \qquad = m_{av} \times \frac{\psi_y}{\psi_y - 0.5}\ (PSG\ 8.38)$

PSG 8.15 Table 13 $\psi_y = 3$ for $i = 1$ to 4

$m_t = 2.28 \times \frac{3}{3-0.5} = 2.736$mm=**3mm (or) 0.3 cm** standard module [PSG DB 8.2]

4. **Correct the number of teeth of pinion** [PSG DB 8.38]

$R = 0.5 \times m_t \times Z_1 \times \sqrt{i^2+1}$

$Z_1 = \frac{R}{0.5 \times mt \times \sqrt{i^2+1}} = \frac{7.8}{0.5 \times 0.3 \times \sqrt{3^2+1}} = 16.5 = 20$ (Standard number of teeth)

$i = \frac{Z_2}{Z_1}$, $Z_2 = 3 \times 20 = 60$

5. **Final cone distance**
[PSG DB 8.38]

$R = 0.5 \times m_t \times Z_1 \times \sqrt{i^2+1}$

$R = 0.5 \times 0.3 \times 20 \times \sqrt{3^2+1}$ = **9.48 cm (OR) 94.8 mm**

Since the final cone distance is greater than initial cone distance,

Our Design is safe.

6. **Calculation of face width** (PSG DB 8.15)

$b = \frac{R}{\psi_y} = \frac{9.48}{3} = 3.16$ cm

7. **Check for compressive stress** (PSG DB 8.13)

$$\sigma_c = \frac{0.72}{(R-0.5b)}\sqrt{\frac{\sqrt{(i^2+1)^3}}{ib}E[M_t]} \leq [\sigma_c]$$

$$= \frac{0.72}{(9.48-0.5\times3.16)}\sqrt{\frac{\sqrt{(3^2+1)^3}}{3\times3.16}2.15\times10^6\times554.07}$$

$\sigma_c = 5745.16$ kgf/cm^2 < $7751.25 Kgf/cm^2$

∴ Our Design is safe.

8. **Check for bending stress**

$$\sigma_b = \frac{R\sqrt{i^2+1}\times[M_t]}{(R-0.5b)^2\times b\times m\times y_v} \times \frac{1}{\cos\alpha}[\alpha = 20\ usually\ from\ PSG\ DB\ 8.13A]$$

$$= \frac{9.48\sqrt{3^2+1}\times 554.07}{(9.48-(0.5\times 3.16))^2\times 3.16\times 0.3\times 0.4} \times \frac{1}{\cos 20^0}$$

$\sigma_b = 107\ kgf/cm^2 < 1214.79\ Kgf/cm^2$

∴ Our Design is safe.

9. **Other parameters of gear drives** [PSG DB 8.38]

$\tan\delta_2 = i = 3$ [PSG DB 8.39]

$\delta_2 = \tan^{-1}(3) = 71.56^0$

$\delta_1 = 90 - 71.56^0 = 18.44^0$ $(\delta_1 + \delta_2 = 90^0)$

Tip diameters for pinion $d_{a1} = m_t(Z_1 + 2\cos\delta_1) = 3(20 + 2\cos 18.44) = 65.69$mm

Tip diameters for gear $d_{a2} = m_t(Z_2 + 2\cos\delta_2) = 3(60 + 2\cos 71.56) = 181.89\ mm$

Addendum angle $\theta_{a1} = \theta_{a2} = \tan^{-1}(\frac{m_t\times f_0}{R}) = \tan^{-1}(\frac{3\times 1}{94.8}) = 1.81^0$

Dedendum angle $\theta_{f1} = \theta_{f2} = \tan^{-1}(\frac{m_t\times(f_0+c)}{R}) = \tan^{-1}(\frac{3\times(1+0.2)}{94.8}) = 2.17^0$

Tip angle for pinion $\delta_{a1} = \delta_1 + \theta_{a1} = 18.44 + 1.81 = 20.25^0$

Tip angle for Gear $\delta_{a2} = \delta_2 + \theta_{a2} = 71.56 + 2.17 = 73.73^0$

Root angle for pinion $\delta_{f1} = \delta_1 - \theta_{f1} = 18.44 - 2.17 = 16.27^0$

Root angle for Gear $\delta_{f2} = \delta_2 - \theta_{f2} = 71.56 - 2.17 = 69.39^0$

Addendum $h_a = m_t = 3$ mm

Dedendum $h_f = 1.1236\ m_t = 1.1236\times 3 = 3.37$ mm

Tooth height $h = h_a + h_f = 3+3.37 = 6.37$ mm

Working depth $h_w = 2m_t = 2\times 3 = 6$ mm.

2. A Pair of bevel gears is to be used to transmit 12 kW from a pinion rotating at 360 rpm to a gear mounted on a shaft which intersects the pinion shaft an angle of 70^0. Assuming that the pinion is to have an outside pitch diameter of 200 mm, a pressure angle of 20^0, a face width of 40 mm, and the gear shaft is to rotate at 120 rpm, determine the a) pitch angle of the gear b) The forces on the gear c) the torque produced about the shaft axis and d) Calculate dia. Of pinion shafts, if τ = 450 N/mm^{2}and if pinion shaft overhangs by 120 mm.

(Nov/Dec 2007)(Nov/Dec 2012)

Given Data:

P = 12 kW

Pinion speed (n_1) = 360 rpm

Angle between two shaft = 70^0

Pitch circle diameter of pinion (Outside) = d_1 = 200 mm

Pressure angle = 20^0

Facewidth b = 40 mm

Gear shaft speed = 120 rpm.

Solution:

Gear ratio $i = \frac{n_1}{n_2} = \frac{360}{120} = 3$

$i = \frac{Z_2}{Z_1}$, $Z_2 = 3 \times 20 = 60$

Pitch circle diameter (d_1) = $m_t Z_1$ = 200 (Assume Z1 = 20)

m_t= 200/20 = **10 mm**

$R = 0.5\times m_t \times Z_1 \times \sqrt{i^2+1}$ [PSG DB 8.38]

$R = 0.5\times 10 \times 20\times \sqrt{3^2+1}$ = **316.22 mm**

$\delta_1 + \delta_2 = 70^0 = \theta$

$\tan \delta 1 = \frac{sin\theta}{i+cos\theta} = \frac{sin70}{3+cos70}$**(Formula to remember)**

$\delta_1 = 15.7^0$

$15.7^0 + \delta_2 = 70^0$

$\delta_2 = 54.3^0$

a) pitch angle of the gear [PSG DBB 8.38]

Addendum angle $\theta_{a1} = \theta_{a2} = \tan^{-1}(\frac{m_t \times f_0}{R}) = \tan^{-1}(\frac{10\times1}{316.22}) = 1.48^0$

Dedendum angle $\theta_{f1} = \theta_{f2} = \tan^{-1}(\frac{m_t \times (f_0+c)}{R}) = \tan^{-1}(\frac{10\times(1+0.2)}{316.22}) = 2.17^0$

Tip angle for pinion $\delta_{a1} = \delta_1 + \theta_{a1} = 15.7 + 1.48 = 17.18^0$

Tip angle for Gear $\delta a2 = \delta 2 + \theta a2 = 54.3 + 2.17 = 56.47^0$

Root angle for pinion $\delta f1 = \delta 1 - \theta f1 = 15.7 - 2.17 = 13.53^0$

Root angle for Gear $\delta f2 = \delta 2 - \theta f2 = 54.3 - 2.17 = 52.13^0$

b) The forces on the gear **[PSG DB 8.57]**

Tangential force $(P_t)_{av} = \frac{2M_t}{d_{av}} = \frac{2M_t}{d_1(1-0.5b/R)}$

$$Mt = \frac{97420\times kw}{n} = \frac{97420\times 12}{360} = 3247.33 \text{ kgf.cm} \quad \text{[PSG DB 8.15]}$$

$$(P_t)_{av} = \frac{2\times 3247.33}{20(1-0.5\times 4/31.622)} = \mathbf{346.65\ kgf}$$

Radial Force $(P_r) = (P_t)_{av} \tan\alpha \cos\delta$

$$= 346.65 \tan 20 \cos 15.7 = \mathbf{121.46\ kgf}$$

Axial Force $(P_a) = (P_t)_{av} \tan\alpha \sin\delta$

$$= 346.65 \tan 20 \sin 15.7 = \mathbf{34.14\ kgf}$$

c) Torque produced about the shaft axis

Assume P_a and P_r induces bending moment:

$$M_1 = (P_r \times overhang) - P_a R_m$$

(Where, $R_m = \frac{d_{av}}{2} = \frac{d_1\left(1-\frac{0.5b}{R}\right)}{2} = \frac{20(1-0.5\times 4/31.622)}{2} = 9.367\ cm$)

$$M_1 = (121.46\times 12) - (34.14\times 9.367) = 1137.73\ kgf - cm$$

P_t Induce also bending moment:

$$M_2 = (P_t \times overhang) = 346.65 \times 12 = 4159.8\ kgf - cm$$

Resultant bending moment:

$$M = \sqrt{M_1^2 + M_2^2} = \sqrt{1137.73^2 + 4159.8^2} = \mathbf{4312.58\ kgf - cm}$$

Equivalent Torque on shaft:

$$T_e = \sqrt{M^2 + T^2} \quad [T = M_t = 3247.33\ kfg - cm]$$

$$T_e = \sqrt{4312.58^2 + 3247.33^2} = \mathbf{5398.47\ kgf - cm}$$

d) **Calculate diameter of pinion shaft:**

$$T_e = \frac{\pi}{16}\tau d^3$$

$$5398.47 = \frac{\pi}{16} \times 450 \times d^3$$

d = 3.93 cm ~ 4.0 cm (OR) 40 mm

Pinion shaft diameter = **40 mm**

HOME WORK

3. **A Pair of bevel gears is to be used to transmit 8 kW from a pinion rotating at 240 rpm to a gear mounted on a shaft which intersects the pinion shaft an angle of 70^0. Assuming that the pinion is to have an outside pitch diameter of 180 mm, a pressure angle of 20^0, a face width of 30 mm, and the gear shaft is to rotate at 80 rpm, determine the forces on the gears and the torque produced about the shaft axis. (Nov/Dec 2012)**

4. **Design a bevel gear drive to transmit 3.5kW.Speed ratio =4.Driving shaft speed 200 rpm. Pinion is of steel and wheel of CI. Assume a life of 25000 hrs.(Nov-2018)**

5. **Design a bevel gear drive to transmit 7.5 kW at 1440 rpm. Gear ratio 3. Pinion and gear are made of forged C45 steel. Life of gears 10,000 hours. Assume surface hardened treatment and IS quality 6.[May- 2017,Dec-2017]**

WORM GEAR DESIGN

1. From the given problem, note down the amount of power to be transmitted, speed ratio, worm speed, material required. **Usually the steel for worm and bronze for wheel are preferred.**
2. Calculation of minimum centre distance (a)

$$a \geq \left(\frac{z}{q}+1\right)\sqrt[3]{\left(\frac{540}{\frac{z}{q}[\sigma_c]}\right)^2 [M_t]} \qquad [PSG\ DB\ 8.44]$$

$[M_t] = M_t k k_d$ (For worm gear $k k_d$ = 1 initially assumed)

$$M_t = 97420 \times \frac{KW}{n_1} \times \eta \times i$$

Selection of number of teeth based on power transmission capacity. (i.e. For hp>20, z = 60 to 70; For small hp, z = 30 to 50)

$Diamerter\ factor\ (q) = \frac{d}{m_x}$ (Initial Choose q=11)

η - Efficiency of worm gear [PSG BD 8.46 Table 37]

$[\sigma_c]$ Design compressive stress which depends on sliding velocity (V_s).

(Initially V_s is assumed 3 m/s)

3. Calculation of minimum axial module (m_x)

3a. Calculation of centre distance [PSG DB 8.43]

$$a = 0.5\, M_x\, (q+z+2x)$$

$$a = 0.5 \times 10\, (11+60) = 355 \text{ mm} \quad (x=0 \text{ initially assume})$$

Since this is less than the minimum centre distance (=375 mm)

Let M_x **= 12 mm (or) 1.2 cm** (PSG DB 8.46 Table 36)

a= 0.5×12 (11+60) = 426 mm (or) 42.6 cm

Since this is greater than the minimum centre distance (=375 mm)

∴ Our design is safe

4. Calculation of actual Sliding velocity

$$v_s = \frac{v_1}{\cos\gamma} = \frac{m_x n}{19100}\sqrt{(z_1)^2 + q^2} \qquad [PSG\ DB\ 8.44]$$

$$v_s = \frac{12 \times 600}{19100}\sqrt{(3)^2 + 11^2} = 4.129 \text{m/sec}$$

Next Select corresponding design compressive stress $[\sigma_c]$ from PSG DB 8.45 Table 32.

$$[\sigma_c] = 1490\ kgf/cm^2$$

5. Check for bending

$$\sigma_b = \frac{1.9\,[M_t]}{m_x^3 \,.\, q \,.\, z \,.\, y_v} \leq [\sigma_b] \qquad [PSG\ DB\ 8.44]$$

$$\sigma_b = \frac{1.9 \times 50270}{1.2^3 \times 11 \times 60 \times 0.4} = 207\ kgf/cm^2 \leq [\sigma_b] = 550\ kgf/cm^2$$

∴ Our design is safe

6. Check for wear

$$\sigma_c = \left(\frac{540}{\frac{z}{q}}\right)\sqrt{\left(\frac{\frac{z}{q}+1}{a}\right)^3}\ [M_t] \leq [\sigma_c] \qquad [PSG\ DB\ 8.44]$$

$$\sigma_c = \left(\frac{540}{\frac{60}{11}}\right)\sqrt{\left(\frac{\frac{60}{11}+1}{42.6}\right)^3} \times 50270 = 1300 kgf/cm^2 \leq [\sigma_c] = 1490 kgf/cm^2$$

∴ Our design is safe

7. Calculation of length of the worm [PSG DB 8.48 Table 39]

Length of worm $L \geq (12.5 + 0.09z)m_x$ (Number of starts 3 or 4)

$L \geq (12.5 + 0.09 \times 60)12$

≥ 214.8 mm

$L \geq 215$ mm

$$a \geq \left(\frac{z}{q}+1\right)\sqrt[3]{\left(\frac{540}{\frac{z}{q}[\sigma_c]}\right)^2 [M_t]} \quad [PSG\ DB\ 8.44]$$

$[M_t] = M_t\, k\, k_d$ (For worm gear $k\, k_d = 1$ initially assumed)

$$[M_t] = 97420 \times \frac{KW}{n_1} \times \eta \times i$$

Z_1 = No. of starts on the worm = 3 (PSG DB 8.44)

$\eta = 0.86$ (*Assume*) Form [PSG DB 8.46 Table 37]

$$[M_t] = 97420 \times \frac{18}{600} \times 0.86 \times 20 = 50270\ kgf-cm$$

(Power = 18 kW = 24.13hp, For hp>20, z = 60 to 70)

Take **z = 60** (PSG DB 8.44)

$Diamerter\ factor\ (q) = \frac{d}{m_x}$ (Initial Choose q=11) (PSG DB 8.44)

$[\sigma_c]$ Design compressive stress which depends on sliding velocity (V_s).

(Always Initially V_s is assume 3 m/s)

$[\sigma_c]$ = 1590 kgf/cm^2 (PSG DB 8.45 Table 32)

$$a \geq \left(\frac{60}{11}+1\right)\sqrt[3]{\left(\frac{540}{\frac{60}{11}\times 1590}\right)^2 \times 50270}$$

$$a \geq 37.4\ cm$$

2. Calculation of minimum axial module (m_x)

$$M_x \geq 1.24\sqrt[3]{\frac{[M_t]}{z\, q\, y_v\, [\sigma_b]}} \quad [PSG\ DB\ 8.44]$$

$[\sigma_b]$ = 550 kgf/cm^2 [PSG DB 8.45 Table 33]

$y_v = 0.4$

$$M_x \geq 1.24\sqrt[3]{\frac{50270}{60 \times 11 \times 0.4 \times 550}}$$

$$M_x \geq 0.82\ cm\ (OR) M_x \geq 8.2\ mm$$

Take $M_x = 10\ mm$

$$M_x \geq 1.24 \sqrt[3]{\frac{[M_t]}{z\, q\, y_v\, [\sigma_b]}}$$ [*PSG DB* 8.44]

$[\sigma_b]$ = Design bending stress [PSG DB 8.45 Table 33]

4. Calculation of centre distance [PSG DB 8.43]

$$a = 0.5\, M_x\, (q+z+2x)$$

Find out the diameter factor (q) form [PSG DB 8.46 Table 36]

$x = \frac{a}{m_x} - 0.5\,(q+z)$ (x=0 initially assumed)

5. Calculation of actual Sliding velocity

$$v_s = \frac{\pi d_1 n_1}{60 \times 1000 \times \cos\gamma}$$

$$v_s = \frac{v_1}{\cos\gamma} = \frac{m_x n}{19100}\sqrt{z^2+q^2}$$ [*PSG DB* 8.44]

v_1 = Pitch line velocity of worm [PSG DB 8.15]

Next select corresponding design compressive stress σ_c from PSG DB 8.45 Table 32.

6. Check for bending

$$\sigma_b = \frac{1.9\,[M_t]}{m_x^3 \cdot q \cdot z \cdot y_v} \leq [\sigma_b]$$ [*PSG DB* 8.44]

7. Check for wear

$$\sigma_c = \left(\frac{540}{\frac{z}{q}}\right)\sqrt{\left(\frac{\frac{z}{q}+1}{a}\right)^3 [M_t]} \leq [\sigma_c]$$ [*PSG DB* 8.44]

8. Calculation of length of the worm [PSG DB 8.48 Table 39]

Length of worm $L \geq (11 + 0.06z)m_x$ (Number of starts 1 or 2)

(OR)

Length of worm $L \geq (12.5 + 0.09z)m_x$ (Number of starts 3 or 4)

For ground worm, the length L is increased to L1.

L1=L+25mm (m_x< 10 mm)

L1=L+40mm (m_x = 10 to 16 mm)

L1=L+50mm (m_x> 16 mm)

9. Calculation of Number of teeth on worm gear

$\lambda = \frac{L_1}{\pi m_x}$ Should be rounded off

Find the actual length of worm L2 = $\lambda \pi m_x$

10. Determine the Face width of the worm wheel (b) [PSG DB 8.48 Table 38]
11. Determine the Parameters of worm and worm wheel [PSG DB 8.43]

Parameters of worm:

Reference diameter $d_1 = q \times m_x$

Tip diameter $d_{a1} = d_1 + 2f_0 m_x$

Root diameter $d_{f1} = d_1 - 2f_0 m_x - 2c$ *(take f_o = 1 and c= 0.2)*

Pitch diameter $d_1' = m_x(q + 2x)$ *(assume x=0)*

Parameters of wheel

Reference diameter $d_2 = Z_2 \times m_x$

Tip diameter $da_2 = (Z_2 + 2f_0 + 2X)m_x$

Root diameter $df_2 = (Z_2 - 2f_0)m_x - 2C$

Pitch diameter $d_2' = d_2$

12. **Efficiency of worm gear drive** [PSG DB 8.49]

Efficiency of worm gear drive $\eta = \dfrac{\tan\gamma}{\tan(\gamma+\rho)}$

$\tan\rho = \mu = friction\ co\ efficient$

PROBLEMS:

1. **The input of worm gear shaft is 18 kW and 600 rpm. Speed ratio is 20. The worm is to be of hardened steel and the wheel is made of phosphor bronze. Considering wear and strength, design worm and worm wheel.**

(April/May 2012, 2018)(Nov/Dec 2012) (Nov/Dec 2015) (Nov/Dec 2010 R2004)

Given:

Power (P) = 18 kW
Speed of worm (n_1) = 600 rpm
Speed ratio (i) = 20
Material for worm = Hardened steel
Material for wheel = phosphor bronze.

Solution:

1. **Calculation of minimum centre distance (a)**

Calculation of Number of teeth on worm gear

$$\lambda = \frac{L}{\pi m_x} = \frac{215}{\pi \times 12} = 5.7 \text{ (PSG DB 8.48)}$$

Take $\lambda = 6$

Actual length of worm = $\lambda \pi m_x = 6 \times \pi \times 12 =$ **226 mm**

8. **Calculation of Face width of the worm wheel (b)** [PSG DB 8.48 Table 38]
   b = 0.75 d_1 = 0.75×($q \times m_x$) = 0.75×11×12 = 99 mm = 100 mm (say) (or) 10 cm
   where $d_1 = q \times m_x$ (PSG DB 8.43)
9. **Calculation of the Parameters of worm and worm wheel** [PSG DB 8.43]

Parameters of worm:

Reference diameter $d_1 = q \times m_x$ = 11 × 12 = 132 mm

Tip diameter $d_{a1} = d_1 + 2 f_0 m_x$ = 132 + (2×1×12) = 156 mm (f_0=1 PSG 8.43)

Root diameter $d_{f1} = d_1 - 2 f_0 m_x - 2c$ = 132 – (2×1×12) – (2×0.2×12) = 103.2 mm

(C = 0.2 m_x) PSG DB 8.43

Pitch diameter $d_1' = m_x(q + 2x)$ = 12 (11+0) = 132 mm

Parameters of wheel

Reference diameter $d_2 = Z_2 \times m_x$ = 60 × 12 = 720 mm

Tip diameter $da_2 = (Z_2 + 2f_0 + 2X)m_x$ = (60 + 2+0)12 = 744 mm

Root diameter $df_2 = (Z_2 - 2f_0)m_x - 2C$ = (60 – 2)12 – (2×0.2×12) = 691.2 mm

Pitch diameter d_2' = d_2 = 720 mm

10. Efficiency of worm gear drive [PSG DB 8.49]

Efficiency of worm gear drive $\eta = \dfrac{\tan\gamma}{\tan(\gamma+\rho)}$

$\gamma = \tan^{-1}(\frac{Z_1}{q}) = \tan^{-1}(\frac{3}{11}) = 15.25^0$ [PSG DB 8.43]

$\tan\rho = \mu$ (PSG DB 8.49)

friction co efficient

(Take μ value from friction co efficient graph PSG DB 8.49)

$\tan\rho = 0.03 \quad \rho = \tan^{-1} 0.03 = 1.72$

$$\eta = \frac{\tan 15.25}{\tan(15.25+1.72)} = 0.893 = \mathbf{89.3\ \%}$$

2. A 2 kW power is applied to a worm shaft at 720 rpm. The worm is of quadruple start type with 50 mm as pitch circle diameter. The worm gear has 40 teeth with 5 mm module. The pressure in the diametral plane is 20^0. Determine (i) the lead angle of the worm, (ii) velocity ratio, and (iii) centre distance. Also calculate efficiency of worm gear drive, and power lost in friction.

(May/June 2008, May/June 2014)

Given Data:

Power (P) = 2 kW

n_1 = 720 rpm

Z_1 = No. of starts on the worm = 4 (The worm is quadruple star type)

Pitch circle diameter (d_1) = 50 mm

Z_2 = 40

Module (m_x) = 5 mm

Solution:

Material Selection:

Material for worm = Hardened steel

Material for wheel = phosphor bronze.

[σ_c] Design compressive stress which depends on sliding velocity (V_s). (Initially V_s is assume 3 m/s)

$[\sigma_c]$ = 1590 kgf/cm^2 [PSG DB 8.45 Table 33]

$[\sigma_b]$ = 550 kgf/cm^2 [PSG DB 8.45 Table 33]

(i) The Lead angle of the worm(γ = *lead or helix angle*) :

$\gamma = \tan^{-1}(\frac{Z_1}{q})$ [PSG DB 8.44]

$$Diamerter\ factor\ (q) = \frac{d_1}{m_x} = \frac{50}{5} = 10$$

$$\gamma = \tan^{-1}(\frac{4}{10}) = 21.8^0$$

$$\gamma = \mathbf{21.8^0}$$

(ii) Velocity ratio (i)

Velocity ratio (i) $= \frac{Z_2}{Z_1} = \frac{40}{4} = \mathbf{10}$

(iii) centre distance (a) [PSG DB 8.43]

a= 0.5 M_x (q+z+2x)

a= 0.5×5 (10+40) = **125 mm** (x=0 initially assume)

(iv) Efficiency of worm gear drive [PSG DB 8.49]

Efficiency of worm gear drive $\eta = \frac{\tan\gamma}{\tan(\gamma+\rho)}$

$\tan\rho = \ \mu = friction\ co\ efficient$

(Take μ value taken from friction co efficient graph PSG DB 8.49)

$$v_s = \frac{v_1}{\cos\gamma} = \frac{m_x n}{1910}\sqrt{(z_1)^2 + q^2} \quad [PSG\ DB\ 8.44]$$

$$v_s = \frac{5\times720}{19100}\sqrt{(4)^2 + 10^2} = 2.03\text{m/sec}$$

$\tan\rho = 0.045$

$\rho = \tan^{-1} 0.045 = 2.57^0$

$$\eta = \frac{\tan 21.8}{\tan(21.8+2.57)} = 0.8829 = \mathbf{88.29\ \%}$$

(v) Power lost in friction (Q)

Q = P (1-η) = 2 (1-0.8829) = 0.234 kW

Q = 0.234 kW

HOME WORK

3. Design worm and gear speed reducer to transmit 22.5 kW at a speed of 1440 rpm. The desired velocity ratio is 24:1. An efficiency of at least 85% is desired.The temperature rise should be restricted to 40^0 C.Determine the required cooling rate.(Dec-2009,2017)

Given:

Power (P) = 22.5 kW

Speed of worm (n_1) = 1440 rpm

Velocity ratio (i) = 24:1

Efficiency = 85 %

Material for worm = **Hardened steel (Assume)**

Material for wheel = **phosphor bronze.(Assume)**

4. A hardened steel worm rotates at 1440rpm and transmits 12kW to a phosphor bronze gear. The speed of the worm gear should be 60 rpm. Design the worm gear drive if an efficiency of at least 82% is desired. (Nov-2018) (By AGMA using PSG design Data)

Given data:

5.**Design a worm gear drive to transmit 20 hp from a worm at 1440 rpm to a worm wheel. Assume the bronze is sand chill cast. The speed of the wheel should be 40 ± 2 rpm, initial sliding velocity can be assumed as 3 m/s and efficiency as 80%.[May– 2017]**

Given:

Power transmitted, P = 20 hp=20 x 736 = 14720 W

Worm speed, n = 1440 rpm

Worm wheel speed, n_1 =40 ± 2 rpm

Sliding velocity.Vs = 3 m/s

Efficiency, = 80 % = 0.8 [As the method is not given, we can select any method]

Solution:

Steps:

1. **Speed ratio, i** $i = \frac{n}{n_1} = \frac{1440}{40} = 36$

2. **Materials: Number of thread on worm, Z; number of teeth on worm wheel, z:**

Worm – steel ;Worm wheel – bronze (assumed)

Z= 3 (assumed) z=iZ = 36 x 3 = 108 teeth

3. **Design worm wheel torque, $[M_t]$:**

PSG DATA BOOK PAGE No: 8.44

$$[M_t] = M_t \times k \times k_d$$

$$k = 1 \; and \; k_d = 1$$

Torque transmitted $M_t = \frac{P\times 60}{2\pi n_1}\eta$

Table 37, PSG DATA BOOK PAGE No: 8.46

For Z = 3 threads, η = 0.8 (assumed)

$$M_t = \frac{60 \times 14720}{2 \times \pi \times 40} \times 0.8 = 2811.31 \text{ N}-\text{m}$$

$$[M_t] = 2811.31 \times 1 \times 1 = 2811.31 \text{ N}-\text{m}$$

4. **Design bending stress $[\sigma_b]$; design surface compressive stress $[\sigma_c]$:**

PSG DATA BOOK PAGE No: 8.45

Table 32: $[\sigma_c]$= 1590 kgf/cm^2=156 x 10^6 N/m^2

Table 33:$[\sigma_b]$= 550 kgf/cm^2=53.955 x 10^6 N/m^2

5. Minimum centre distance, a:

PSG DATA BOOK PAGE No: 8.44

$$a = \left(\frac{z}{q}+1\right)\sqrt[3]{\left[\frac{540}{\frac{z}{q}\times[\sigma_c]}\right]^2 \times \{[M_t]\times 10^5\}}$$

Diameter factor, q = 11 (assumed)

$$a = \left(\frac{108}{11}+1\right)\sqrt[3]{\left[\frac{540}{\frac{108}{11}\times 156\times 10^6}\right]^2 \times \{2811.31\times 10^5\}}$$

$$a = 0.3537\,m = 353.7\,mm$$

6. Axial module, m_x:

$$m_x = 1.24\sqrt[3]{\left[\frac{[M_t]}{z\,.\,q.\,y_v\times[\sigma_b]}\right]}$$

PSG DATA BOOK PAGE No: 8.43

$$\tan\gamma = \frac{Z}{q} = \frac{3}{11}\quad lead\ angle\ ,\gamma = \tan^{-1}\left(\frac{3}{11}\right) = 15.255^o$$

Virtual no. of teeth on worm wheel,

$$z_{v1} = \frac{z_1}{cos^3\gamma} = \frac{108}{cos^3 15.255} = 120.27$$

PSG DATA BOOK PAGE No: 8.18, Table 18:

For zv = 80; y_v = 0.499 at X = 0 (approx.)

$$m_x = 1.24\sqrt[3]{\left[\frac{2811.31}{108\times 11\times 0.509\times 53.955\times 10^6}\right]} = 5.48\times 10^{-3}m = 5.48\,mm$$

PSG DATA BOOK PAGE No: 8.2 Table 1:

Taking next higher standard module: mx = 8 mm

7. Revision of centre distance, a:

PSG page 8.45:

a = 0.5 mx (q+z+2x) = 0.5x(mx(q+z) = 0.5x6(11+108) = 357 mm

8. Pitch diameters; d, pitch line velocity of worm, v_1 and sliding velocity, v_s:

PSG DATA BOOK PAGE No: 8.43:

Pitch diameter – worm, d1 = q.mx = 11x6=66 = 0.066 m.

Pitch diameter – worm wheel, d2 = z.mx = 108x6 = 648 mm = 0.648 m

Pitch line velocity of worm,

$$Pitch\ line\ velocity, v_1 = \frac{\pi d_1 n}{\cos\gamma} = \frac{\pi \times 0.066 \times 1440}{\cos 15.255} = 4.98\, m/s$$

9. Revision of [σc]:

The sliding velocity, vs> 3 m/s

The design stress [σc] has to be revised & reduced.

Table 32 , PSG DATA BOOK PAGE No: 8.45:

For steel – bronze combination, vs = 5.16 m/s

Assume [σc]:1490kgf/cm^2 (approx.) = 146.2 x 10^6 N/m^2

10. Revision of design torque, [M_t]:

$$[M_t] = M_t \times k \times k_d$$

$$k = 1\ and\ k_d = 1$$

$$[M_t] = 2811.31 \times 1 \times 1 = 2546.5\ N - m$$

11. Check for induced bending stress, σ_b:

PSG DATA BOOK PAGE No: 8.44:

$$\sigma_b = \frac{1.9[\mathrm{M_t}]}{m_x^3 . q . z . y_v} . \le [\sigma_b]$$

$$\sigma_b = \frac{1.9 \times 2811.31}{0.006^3 \times 11 \times 108 \times 0.509} = 40.9 \times 10^6\ N/m^2$$

Which is less than the design bending stress, $[\sigma_b] = 53.955 \times 10^6\ N/m^2$

The design is safe.

12. Check for induced surface compressive stress, σ_c:

$$\sigma_c = \frac{540}{\left(\frac{z}{q}\right)} \sqrt{\left[\frac{\left(\frac{z}{q}+1\right)}{a}\right]^3} .\{[M_t] \times 10^5\} \leq [\sigma_c]$$

$$\sigma_c = \frac{540}{\left(\frac{108}{11}\right)} \sqrt{\left[\frac{\left(\frac{108}{11}+1\right)}{0.357}\right]^3} .\{2811.31 \times 10^5\} \leq [\sigma_c]$$

$\sigma_c = 153.83 \times 10^6 N/m^2$ *which is less than the design surface compressive stress*

$$[\sigma_c] = 146.64 \times 10^6 N/m^2$$

The design is safe.

13. Check for efficiency, η, if required:

PSG DATA BOOK PAGE No: 8.49:

$$efficienty, \eta = \frac{\tan\gamma}{\tan(\gamma+\rho)}$$

For bronze worm wheel and sliding velocity, vs = 6.02 m/s

Using extrapolation; $\mu = \tan\rho = 1.1476$ $(approx)$

Friction angle $\rho = \tan^{-1} 0.02 = 1.14576^o$

$$\eta = \frac{\tan(15.255)}{\tan(15.255 + 1.14576)} = 0.9263 \; or = 92.63 \;\%$$

This is satisfactory.

UNIT – IV

GEAR BOX

	Arrangement of speeds
1. 6 Speed gear box	$6 = 1 \times 2 \times 3$
2. 9 Speed gear box	$9 = 1 \times 3 \times 3$
3. 12 Speed gear box	$12 = 1 \times 2 \times 2 \times 3$
4. 18 Speed gear box	$18 = 1 \times 2 \times 3 \times 3$

X

Note :-
If Speed reduction by belt in any speed gear box means

example : 9 Speed = $1 \times 1 \times 3 \times 3$
for belt reduction.

problems

1. Design a six speed gear box for the following data

Motor power = 5 kW at 1440 rpm

Maximum output speed = 1600 rpm

Minimum output speed = 460 rpm

Given data:-

power, $P = 5\,kW$

input (Motor) speed, $N = 1440$ rpm

Max. output speed, $N_{max} = 1600$ rpm

Min. output speed, $N_{min} = 460$ rpm

Number of speeds $r = 6$

Solution:-

1. progression ratio $\phi = \left(\dfrac{N_{max}}{N_{min}}\right)^{\frac{1}{(r-1)}}$ (or) $\dfrac{N_{max}}{N_{min}} = \phi^{r-1}$

$$\phi = \left(\frac{1600}{460}\right)^{\frac{1}{(6-1)}} = 1.283$$

In PSG 7.20 Basic Series of preffered numbers

$\phi = 1.283$ is not available.

So Now required output speeds are

$N_1 = N_{min} = 460$ rpm

$N_2 = N_1 \times \phi = 460 \times 1.283 = 590 = 600$ rpm (Say)

$N_3 = N_2 \times \phi = 600 \times 1.283 = 770 = 750$ rpm (Say)

$N_4 = N_3 \times \phi = 750 \times 1.283 = 962 = 950$ rpm (Say)

$N_5 = N_4 \times \phi = 950 \times 1.283 = 1219 = 1250$ rpm (Say)

$N_6 = N_5 \times \phi = 1250 \times 1.283 = 1604 = 1600$ rpm (Say)

∴ Six out put speeds are

460, 600, 750, 950, 1250 and 1600 rpm.

∴ Let us consider the arrangement of speeds

$6 = 1 \times 2 \times 3$

∴ Let us consider all the gear in the gear box are SPUR GEAR (Assume)

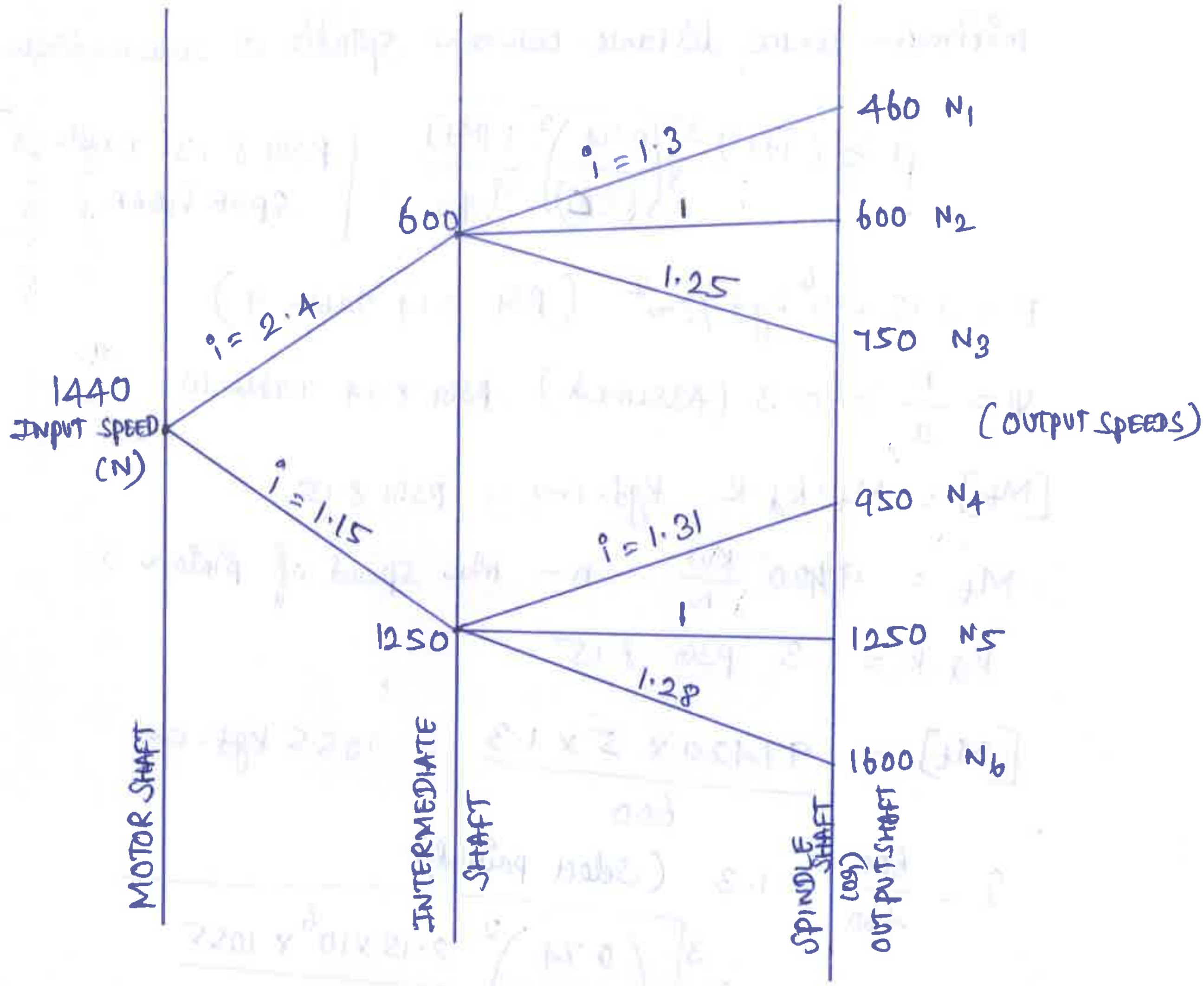
∴ Material for all SPUR GEAR in the gear box } = C45 Steel (Assume)

PSG DB 8.5

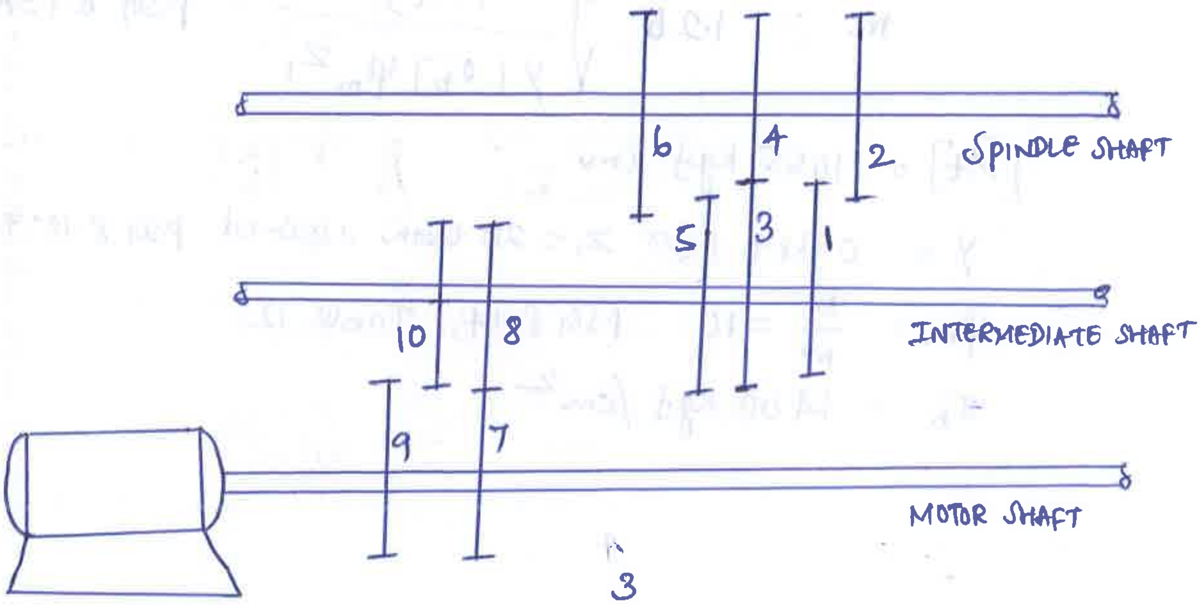
Compressive stress $[\sigma_c] = 5000$ kgf/cm^2

Bending stress $[\sigma_b] = 1400$ kgf/cm^2

2. SPEED DIAGRAM (OR) RAY DIAGRAM :- (1×2×3)



3. KINEMATIC ARRANGEMENT



4. Design of gears between spindle shaft and Intermediate shaft :-

minimum centre distance between spindle & Intermediate shaft :

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.74}{(\sigma_c)}\right)^2 \frac{E[M_t]}{i\,\psi}}$$ [PSG 8.13 Table 8 SPUR GEAR]

$E = 2.15 \times 10^6\ kgf/cm^2$ (PSG 8.14 Table 9)

$\psi = \frac{b}{a} = 0.3$ (Assumed) PSG 8.14 Table 10

$[M_t] = M_t \cdot K_d \cdot K$ kgf. cm PSG 8.15

$M_t = 97420 \frac{kW}{n_1}$ n – min speed of pinion

$K_d \cdot K = 1.3$ PSG 8.15

$$[M_t] = \frac{97420 \times 5 \times 1.3}{600} = 1055\ kgf \cdot cm$$

$i = \frac{600}{460} = 1.3$ (Select pair i, 82)

$$a \geq (1.3+1)\sqrt[3]{\left(\frac{0.74}{5000}\right)^2 \frac{2.15 \times 10^6 \times 1055}{1.3 \times 0.3}}$$

$a \geq 11.5\ cm$

Minimum Module

$$m \geq 1.26\sqrt[3]{\frac{[M_t]}{y\,[\sigma_b]\,\psi_m Z_1}}$$ PSG 8.13A Table 8

$[M_t] = 1055$ kgf. cm

$y = 0.389$ (for $Z_1 = 20$ teeth assumed PSG 8.18 Table (8))

$\psi_m = \frac{b}{m} = 10$ PSG 8.14 Table 12

$\sigma_b = 1400\ kgf/cm^2$

$$m \geq 1.26 \sqrt[3]{\frac{1055}{0.389 \times 1400 \times 10 \times 20}}$$

$m \geq 0.27$ cm (or) 2.7 mm.

Next standard module (PSG 8.2 Table 1)

$m \geq 0.3$ cm (or) 3 mm (say)

Now number of teeth of pinion

$$z_1 = \frac{2a}{m(i+1)} = \frac{2 \times 11.5}{0.3(1.3+1)} = 34$$

WKT $i = \frac{z_2}{z_1} \Rightarrow z_2 = i z_1 = 1.3 \times 34 = 44$

Correct centre distance

$$a = \frac{m}{2}(z_1+z_2) \quad \text{PSG 8.22}$$

$$a = \frac{0.3}{2}(34+44) = 11.7 \text{ cm (OK)}$$

It is satisfied.

Also $z_1 + z_2 = 78$ teeth $\therefore (34 + 44 = 78$ teeth$)$

Since the centre distance is same for other engaging pair are equal.

$$z_1 + z_2 = z_3 + z_4 = z_5 + z_6 = 34 + 44 = 78$$

WKT gear ratio for pair 3 & 4 is 1

$$\therefore z_3 = z_4 = \frac{78}{2} = 39 \text{ teeth}$$

$\therefore z_3 = 39$ teeth, $\therefore z_4 = 39$ teeth

IIly

$$\frac{z_5}{z_6} = 1.25 \; ; \; \therefore z_5 = 43 \; ; \; z_6 = 35$$

5. Design of gears between Intermediate Shaft and Motor Shaft :-

Minimum centre distance

$$a \geq (i+1) \sqrt[3]{\left(\frac{0.74}{(\sigma_c)}\right)^2 \frac{E[M_t]}{i\psi}}$$ PSG 8.13 Table 8

$$[M_t] = \frac{97420 \times 5 \times 1.3}{1440} = 440 \text{ kgf.cm}$$

$$i = \frac{1440}{60} = 2.4$$ (Select pair 7 & 8)

$$a \geq (2.4+1) \sqrt[3]{\left(\frac{0.74}{5000}\right)^2 \frac{2.15 \times 10^6 \times 440}{2.4 \times 0.3}}$$

$$a \geq 10 \text{ cm.}$$

Minimum Module PSG 8.13A Table 8

$$m \geq 1.26 \sqrt[3]{\frac{[M_t]}{y[\sigma_b]\psi_m z_1}}$$ [Here $z_1 = z_7$] but assume $z_1 = 20$

$$m \geq 1.26 \sqrt[3]{\frac{440}{0.389 \times 1400 \times 10 \times 20}}$$

$m \geq 0.24$ cm (or) 0.3 cm Say (Next standard module Refer PSG 8.2 Table 1)

$\therefore m \geq 0.3$ cm (or) 3 mm

$$\therefore z_7 = \frac{2a}{m(i+1)} = \frac{2 \times 10}{0.3(2.4+1)} = 21$$

WKT $i = \frac{z_8}{z_7} \Rightarrow z_8 = i z_7 = 2.4 \times 21 = 51$ teeth

Correct centre distance

$$a = \frac{m}{2}(z_7 + z_8)$$ PSG 8.22

$$a = \frac{0.3}{2}(21+51) = 10.8 \text{ cm}$$ (OK)

It is Satisfied

$\therefore Z_7 + Z_8 = Z_9 + Z_{10} = 72 \ (21 + 51)$

WKT

Gear ratio for pair 9 & 10 is 1.15

11ly $\frac{Z_{10}}{Z_9} = 1.15$; $Z_9 = 33$; $Z_{10} = 39$ teeth

Results:-

Centre distance

Spindle to intermediate shaft = 11.7 cm

Intermediate shaft to Motor = 10.8 cm

Module of gears = 0.3 cm (or) 3 mm

No. of Teeth of gears (Z)	Gear ratio (i)
$Z_1 = 34$; $Z_2 = 44$	1.3
$Z_3 = 39$; $Z_4 = 39$	1
$Z_5 = 43$; $Z_6 = 35$	1.25
$Z_7 = 21$; $Z_8 = 51$	2.4
$Z_9 = 33$; $Z_{10} = 39$	1.15

A machine tool gear box is to have 9 speeds gear box is driven by an electrical motor whose shaft rotational speed is 1400 rpm. The gear box is connected to the motor by a belt drive. The maximum and minimum speeds required at the gear box output are 1000 rpm and 200 rpm respectively. Suitable speed reduction can also be provided in the belt drive. What is the step ratio and what are the values of 9 speeds ?. Sketch the gear arrangement obtain No. of teeth on each gear and also the actual output speeds.

Given Data:

No. of speed $(r) = 9$

Motor speed $(N) = 1400$ rpm

Maximum speed $(N_{max}) = 1000$ rpm.

Minimum speed $(N_{min}) = 200$ rpm.

Solution:

$$\phi^{r-1} = \frac{N_{max}}{N_{min}}$$

$$\phi^{9-1} = \frac{1000}{200} \Rightarrow \phi = (5)^{1/8}$$

$$\phi = 1.22 \simeq 1.25$$

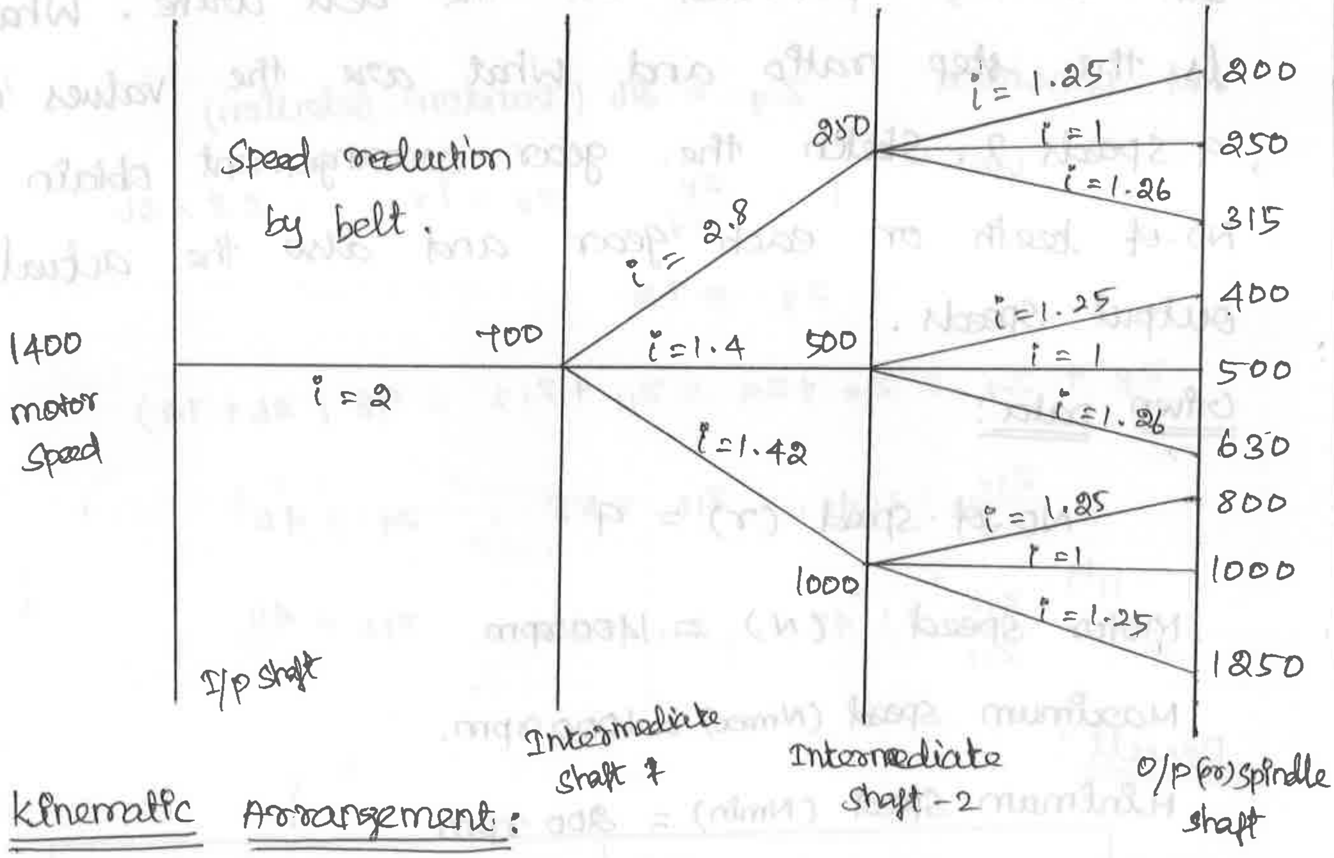
9 output speeds are, (From PSG 7.20)

200, 250, 315, 400, 500, 630, 800, 1000 & 1250.

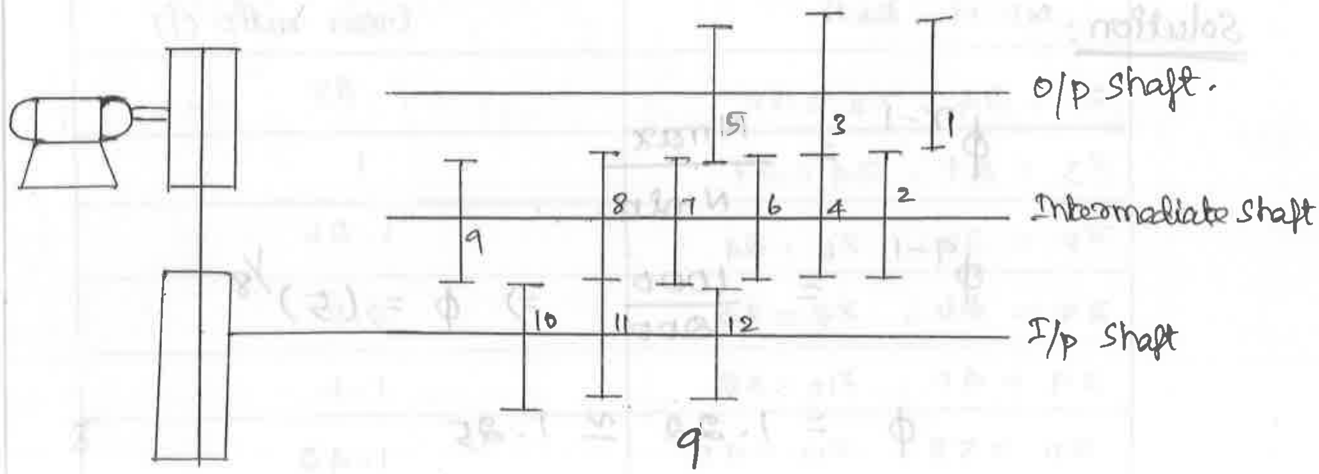
Speed arrangement of 9 speed gear box.

$$1 \times 1 \times 3 \times 3 = 9$$

<u>Speed (or) Ray diagram</u>: $(1 \times 1 \times 3 \times 3)$



<u>Kinematic Arrangement</u>:



Let us consider,

$$Z_1 = 24 \quad (\text{Random selection})$$

WKT,
$$i = \frac{Z_2}{Z_1} \Rightarrow Z_2 = iZ_1 = 1.25 \times 24$$

$$Z_2 = 30.$$

$$Z_1 + Z_2 = Z_3 + Z_4 = Z_5 + Z_6 = 54 \ (24+30).$$

$$\frac{Z_4}{Z_3} = 1, \quad Z_4 = 27, \quad Z_3 = 27$$

^{11}ly

$$\frac{Z_6}{Z_5} = 1.26, \quad Z_5 = 30, \quad Z_6 = 24$$

Let consider, $Z_7 = 26$ (Random selection)

WKT,
$$i = \frac{Z_8}{Z_7}, \quad Z_8 = iZ_7 = 2.8 \times 26$$

$$Z_8 = 72$$

$$Z_8 + Z_7 = Z_{10} + Z_9 = Z_{11} + Z_{12} = 98 \ (26 + 72)$$

$$\frac{Z_{10}}{Z_9} = 1.4, \quad Z_{10} = 58, \quad Z_9 = 40$$

^{11}ly

$$\frac{Z_{12}}{Z_{11}} = 1.42, \quad Z_{11} = 58, \quad Z_{12} = 40.$$

Result:

No. of teeth	Gear ratio (i).
$Z_1 = 24$, $Z_2 = 30$	1.25
$Z_3 = 27$, $Z_4 = 27$	1
$Z_5 = 30$, $Z_6 = 24$	1.26
$Z_7 = 26$, $Z_8 = 72$	2.8
$Z_9 = 40$, $Z_{10} = 58$	1.4
$Z_{11} = 58$, $Z_{12} = 40$	1.42

(6) Design the layout of 12 speed gear box for a lathe the minimum and maximum speeds are 100 and 1200 rpm power is 5 kW from 1440 rpm reduction motor. Construct the speed diagram using standard ratio. Calculate the No. of teeth in each gear wheel and sketch the arrangement of gear box.

Given Data:

No. of. speed $(r) = 12$

Motor speed $(N) = 1440$ rpm

power $(P) = 5$ kW

Minimum Speed $(N_{min}) = 100$ rpm.

Maximum Speed $(N_{max}) = 1200$ rpm.

Solution:

$$\phi^{r-1} = \frac{N_{max}}{N_{min}}$$

$$\phi^{12-1} = \frac{1200}{100} \Rightarrow \phi = (11)^{1/11}$$

$$\phi = 1.25$$

12 output speeds are, (From PSG 7.20)

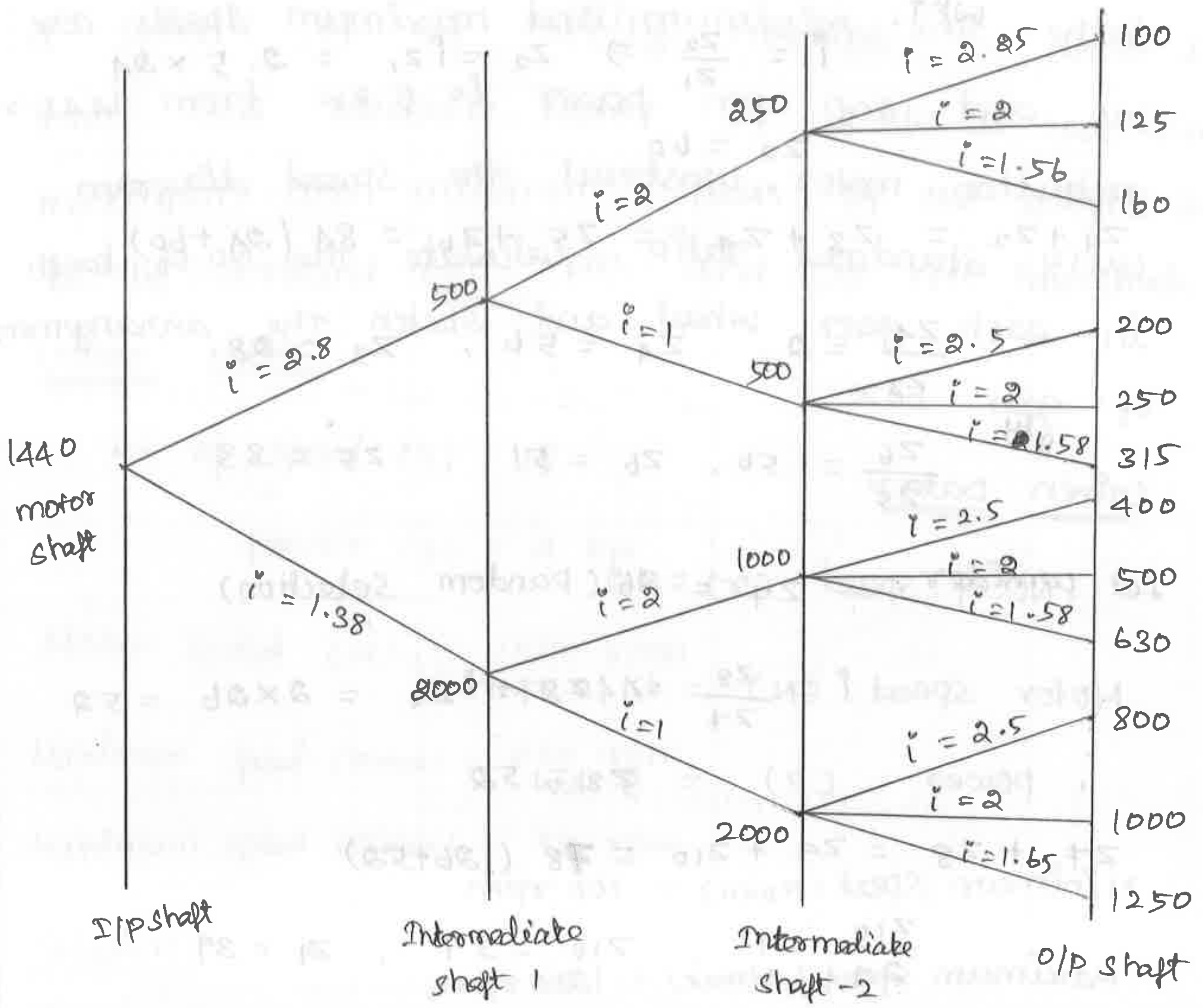
100, 125, 160, 200, 250, 315, 400, 500
630, 800, 1000 & 1250

Speed arrangement of 12 Speed gear box,

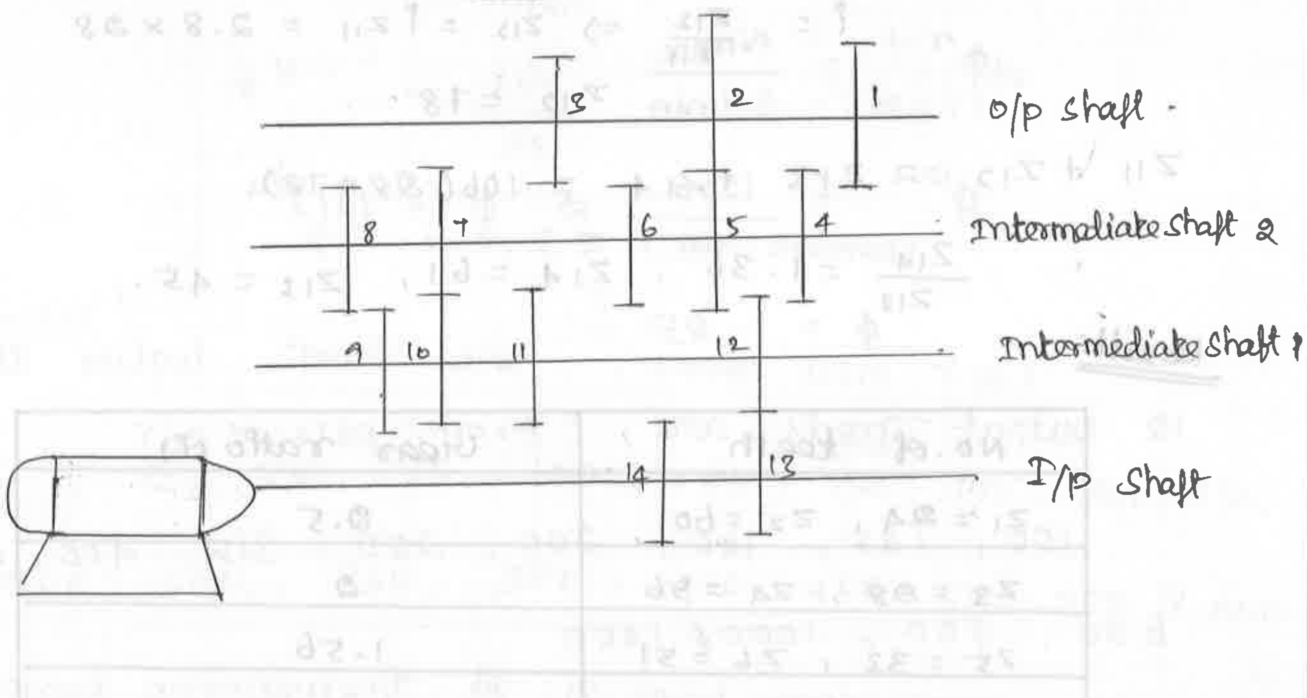
$$1 \times 2 \times 2 \times 3 = 12$$

11

Speed (or) Ray diagram : $(1 \times 2 \times 2 \times 3)$



Kinematic Arrangement :



Let consider, $Z_1 = 24$ (Random selection)

WKT,

$$i = \frac{Z_2}{Z_1} \Rightarrow Z_2 = i Z_1 = 2.5 \times 24$$

$$Z_2 = 60$$

$$Z_1 + Z_2 = Z_3 + Z_4 = Z_5 + Z_6 = 84\ (24+60)$$

$$\frac{Z_4}{Z_3} = 2, \quad Z_4 = 56, \quad Z_3 = 28$$

11^{ly}

$$\frac{Z_6}{Z_5} = 1.56, \quad Z_6 = 51, \quad Z_5 = 33$$

Let consider, $Z_7 = 26$ (Random selection)

$$i = \frac{Z_8}{Z_7} \Rightarrow Z_8 = i Z_7 = 2 \times 26 = 52$$

$$Z_8 = 52$$

$$Z_7 + Z_8 = Z_9 + Z_{10} = 78\ (26+52)$$

$$\frac{Z_{10}}{Z_9} = 1, \quad Z_{10} = 39, \quad Z_9 = 39$$

Let consider, $Z_{11} = 28$ (Random selection)

$$i = \frac{Z_{12}}{Z_{11}} \Rightarrow Z_{12} = i Z_{11} = 2.8 \times 28$$

$$Z_{12} = 78$$

$$Z_{11} + Z_{12} = Z_{13} + Z_{14} = 106\ (28+78).$$

$$\frac{Z_{14}}{Z_{13}} = 1.36, \quad Z_{14} = 61, \quad Z_{13} = 45.$$

Result:

No. of. teeth	Gear ratio (i)
$Z_1 = 24$, $Z_2 = 60$	2.5
$Z_3 = 28$, $Z_4 = 56$	2
$Z_5 = 33$, $Z_6 = 51$	1.56
$Z_7 = 26$, $Z_8 = 52$	2
$Z_9 = 39$, $Z_{10} = 39$	1
$Z_{11} = 28$, $Z_{12} = 78$	2.8
$Z_{13} = 45$, $Z_{14} = 61$	1.36

Design a gear box to give 18 speeds for a spindle of a milling machine. The drive is from an electric Motor of 4 kW at 1000 rpm. maximum and minimum speeds of the spindle are to be around 650 rpm and 35 rpm respectively.

Given Data:

No. of. speed $(r) = 18$

power $(p) = 4$ kW

Motor speed $(N) = 1000$ rpm

Maximum speed $(N_{max}) = 650$ rpm

Minimum speed $(N_{min}) = 35$ rpm.

Solution:

$$\phi^{r-1} = \frac{N_{max}}{N_{min}}$$

$$\phi^{18-1} = \frac{650}{35} \Rightarrow \phi = (18.57)^{1/17}$$

$$\phi = 1.18, \ \phi \approx 1.06 \text{ (assumed)}$$

18 output speeds are, (From PSG T.20).

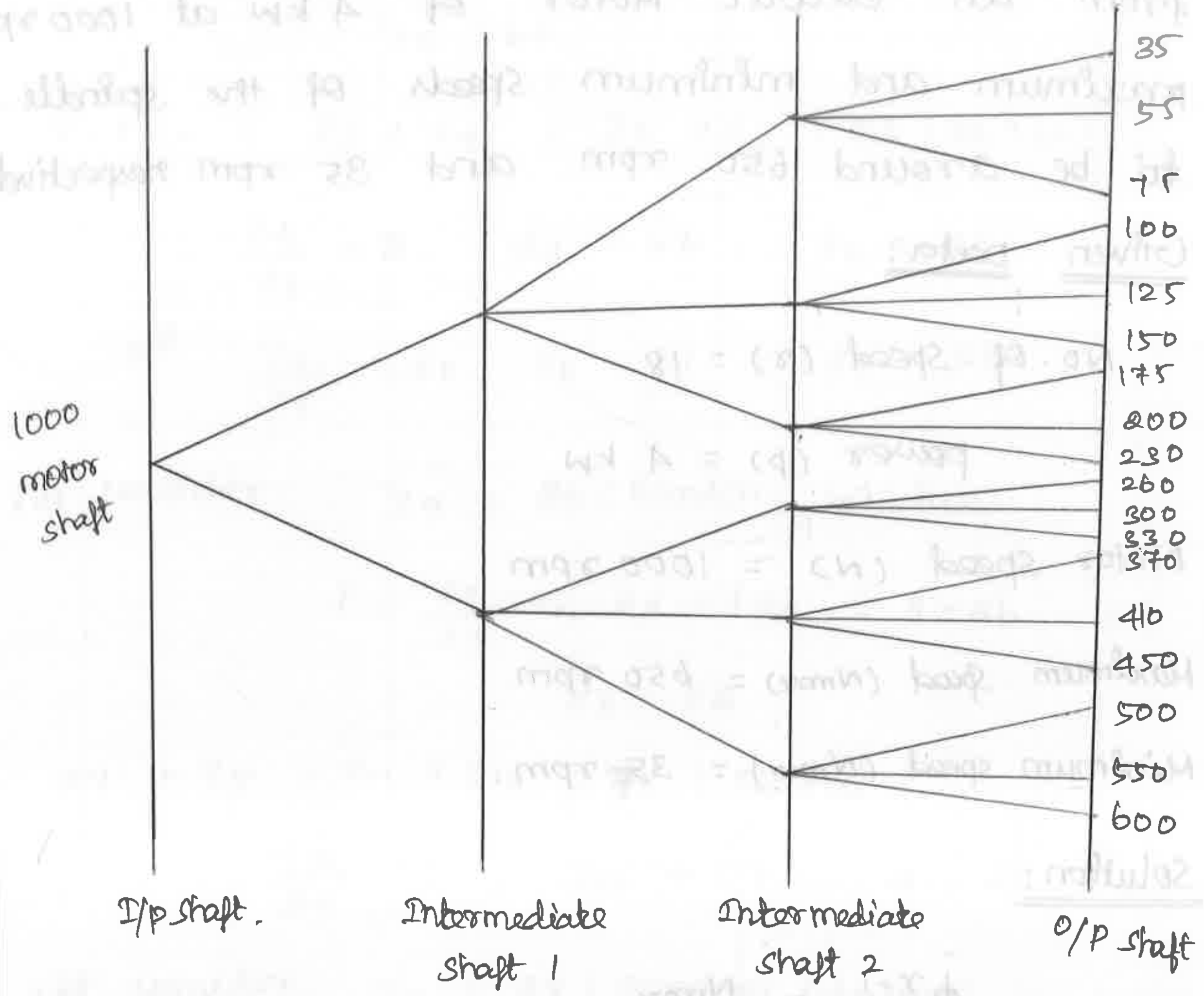
35, 55, 75, 100, 125, 150, 175, 200, 230, 260, 300, 330, 370, 410, 450, 500, 550 & 600

speed arrangement of 18 speed gear box.

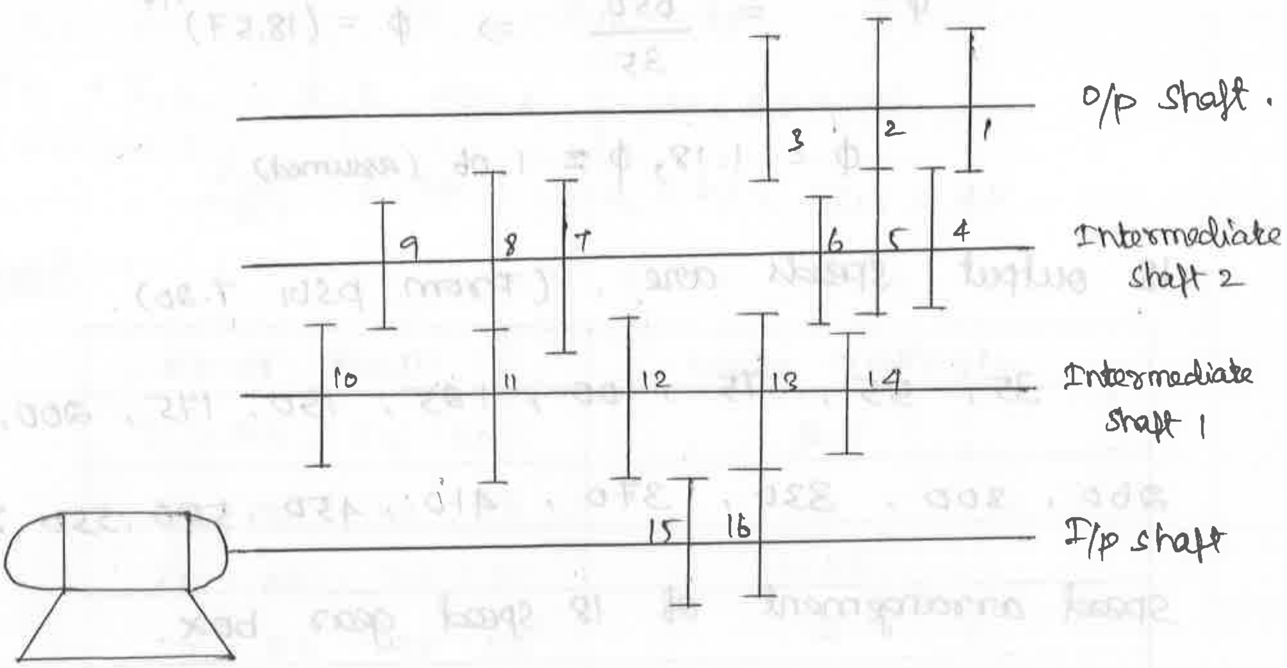
$$1 \times 2 \times 3 \times 3 = 18$$

(8)

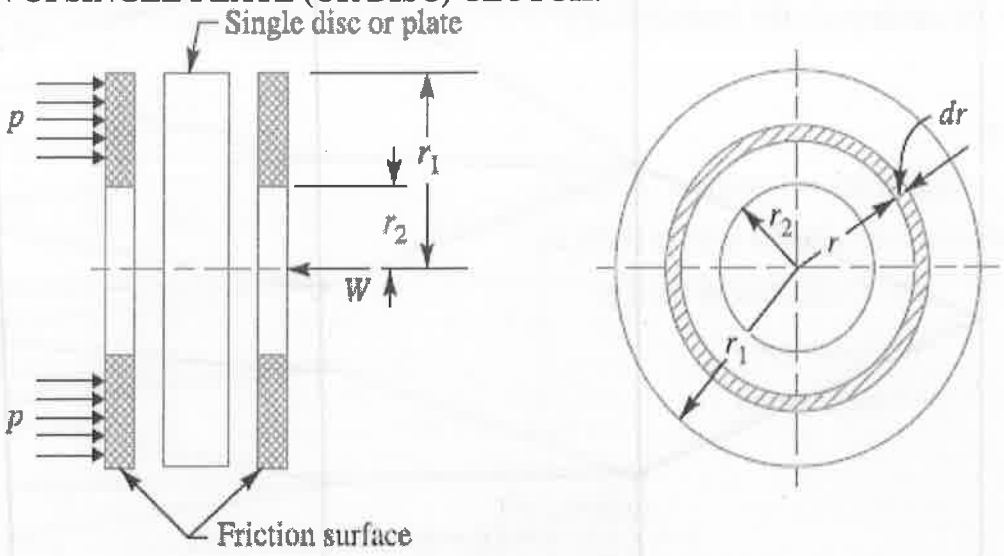
Speed (or) Ray diagram : $(1 \times 2 \times 2 \times 3)$



Kinematic Arrangement:



DESIGN OFSINGLE PLATE (OR DISC) CLUTCH:



$$r_1 = r_o;\ r_2 = r_i$$

(i) Uniform pressure theory

Intensity of pressure $P = \frac{W}{\pi(r_o^2 - r_i^2)}$

Total Frictional torque $T_f = \frac{2}{3}\,\mu\, W \left[\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right]$

$$T = \mu W r$$

Where r = mean radius of the friction surface

$$r = \frac{2}{3}\left[\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right]$$

Maximum Intensity of pressure $P_{max} = \frac{W}{\pi\,({r_o}^2 - {r_i}^2)}$ $for\ uniform\ pressure$

(ii) Uniform wear theory

- The intensity of pressure is maximum at inner radius r_i

$$P_{max} r_i = C$$

- The intensity of pressure is minimum at outer radius r_o

$$P_{max} r_o = C$$

Maximum Intensity of pressure $P_{max} = \frac{W}{2\pi\, r_o (r_o^2 - r_i)}$ $for\ uniform\ wear$

Total Frictional torque $T = \mu W r$

Where r = mean radius of the friction surface

$$r = \frac{r_o + r_i}{2}$$

Axial thrust = W= $2\pi C\ (r_o - r_i)$

In general

Total Frictional torque $T = n\,\mu W r$

Where n= no. of pairs contact surfaces

$\mu =$ Coefficient of friction

W= axial load

r = mean radius

$$r = \frac{2}{3}\left[\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right] For\ uniform\ pressure$$

$$r = \frac{r_o + r_i}{2} \quad For\ uniform\ wear$$

For a single plate clutch, n= 2 (both sides of the plate are effective)

For a multi disc clutch n = $n_1 + n_2 - 1$

Where $n_1 = no. of\ discs\ on\ driving\ shaft$

$n_2 = no. of\ discs\ on\ driven\ shaft$

where $n_1 = \frac{n}{2}$

$n_2 = n_1 + 1$

1. A single dry plate clutch is to be designed to transmit 10 HP at 900 rpm. Find (a) diameter of the shaft (b) mean radius and face width of the friction lining assuming the ratio of the mean radius to the face width as 4 (c) outer and inner radii of the clutch plate. Take σ_y for the shaft material as 38 kg/mm^2 (Nov/Dec 2009)

Given:

Power = 10 HP = 10×736 = 7.36 kW

Speed = 900 rpm

$\frac{Maen\ Radius}{face\ width} = \frac{r_m}{b} = 4$

$\sigma_y = 38\ kg/mm^2$

Solution:

(a) To find diameter of clutch shaft (d)

Diameter of clutch shaft $d = \sqrt[3]{\frac{495000 \times kW \times k_w}{n[\tau]}}$ [*PSG DB* 7.89]

$$k_w = k_1 + k_2 + k_3 + k_4$$

$$k_1 = 0.5\,, k_2 = 1.25, k_3 = 0.32, k_4 = 0.75$$

$$k_w = 2.82$$

Assume material of shaft as C50

$$\sigma_y = 38\ \text{kg}/mm^2 \quad \text{FOS} = 2$$

$$[\tau] = \frac{\sigma_y}{2FOS} = \frac{38}{2 \times 2} = 9.5\ kgf/mm^2$$

$$[\tau] = 950\ kgf/cm^2$$

$$d = \sqrt[3]{\frac{495000 \times 7.36 \times 2.82}{900 \times 950}}$$

d = 2.3 cm (or) 23 mm

(b) To find the face width and mean radius

$$i_{min} = \frac{[M_t]}{2\pi p_a b \mu r_m^2} \qquad [PSG\ DB\ 7.89]$$

$$i_{min} = 2\ (For\ single\ plate\ clutch)$$

$$Allowable\ Pressure\ (p_a) = k.P_b\ , Assume\ k = 1$$

$$Basic\ Pressure\ (P_{b)} = 2.5\ to\ 3\ kgf/cm^2\ (For\ steel\ plate\ dry\ running)$$

Take $P_b = 3\ kgf/cm^2$

$$[M_t] = M_t k_w$$

$$M_t = \frac{97400\ kW}{n} = \frac{97400 \times 7.36}{900}$$

$$M_t = 795.77\ kgf - cm$$ ∴ $M_t = M_t\ k_w = 796 \times 2.82$
$= 2242$

$$2 = \frac{2242}{2\pi \times 3 \times \frac{r_m}{4} \times 0.25 \times r_m^2}$$

$$r_m = 9.8\ cm\ (or)\ \mathbf{98\ mm}$$

Face width b = $\frac{r_m}{4} = \frac{98}{4} = \mathbf{24.5mm}$

(c) To find the inner and outer radius

b = face width of the plate = $r_o - r_i = \frac{r_m}{4} = 24.5$ − − − − − − − − − − (1)

$\frac{r_m}{b} = 4$ (given) ∴ So $b = \frac{r_m}{4}$

$$r_m = \frac{r_o + r_i}{2} \qquad \text{PSG 7.89}$$

$$r_o + r_i = 196 - - - - - - - - \quad (2)$$

By Solving equ (1) and (2) we get $\boldsymbol{r_o = 110.25\ mm r_i = 85.75\ mm}$

2. Find the torque that a two surface, dry disk clutch can transmit if the outside and inside lining diameters are 120 mm and 70 mm, respectively, and the applied axial force is 10 kN. Assume uniform wear and $\mu = 0.4$. Is the pressure on the lining acceptable? What lining material would be suitable? (May/June 2010)

Given:

Outer diameter (d_0) = 120mm

Inner diameter (d_i) = 70mm

Co-efficient of friction (μ) = 0.4

Axial force (F) = 10KN

No. of surfaces (n) = 2

To Find:

(i) Torque capacity (T)

(ii) Maximum lining pressure (P_{max})

(iii) Lining material

Solution:

Outer radius

$r_0 = 0.5\ d_0$

$r_0 = 0.5 \times 120$

$r_0 = 60$ mm

Inner radius

$r_i = 0.5\ d_i$

$r_i = 0.5 \times 70$

$r_i = 35$ mm

Calculate torque capacity:

$T = n\mu F\left(\frac{r_0 + r_i}{2}\right)$

n = Number of surfaces (Given = 2)

$$T = 2\times0.4\times10000\times(\frac{60+35}{2})$$

$$T = 380\text{ N-m}$$

Calculate the maximum lining pressure (P_{max}):

$$P_{max} = \frac{F}{2\pi r_i(r_0-r_i)}$$

$$P_{max} = 1.819\text{KPa (or) KN/mm}^2$$

Properties of common clutch, brake lining materials the obtained P_{max} = 1819KPa falls in the range. Therefore, the suitable lining material are **molded and sintered metal**

3. **A single plate clutch effective on both sides, is required to transmit 25 KW at 3000 rpm. Determine the outer and inner diameter of frictional surfaces if the coefficient of friction is 0.25, ratio of diameter is 1.25 and the maximum pressure is not to exceed 0.1 N/mm^2. Determine (i) the face width required and (ii) the axial spring force necessary to engage the clutch. Assume uniform wear condition. (Nov/Dec 2009 R2001).**

Given:

Power (P) = 25 KW

Speed (N) = 3000 rpm

Co-efficient of friction (μ) = 0.25

Maximum pressure (P_{max}) = 0.1 N/mm^2

Ratio of diameter ($\frac{R_0}{R_I}$) =1.25

Single plate clutch (n) = 2

To Find:

(i) The face width required

(ii) The axial spring force necessary to engage the clutch.

Solution:

WKT $P = \frac{2\pi N T}{60}$

Torque to be transmitted, $T = \frac{60P}{2\pi N} = \frac{60\times25000}{2\pi\times3000} = 79.577\ N-m = 79.577\times10^3\ N-mm$

Also we know that,

$T = F\mu\ R_f\ n_p$

$F_a = 2\pi P_{max} R_i(R_0 - R_i)$

$= 2\pi \times 0.1 \times R_i(1.25R_i - R_i)$

$= 0.157\ R^2_i$

$R_f = \frac{R_0 + R_i}{2} = 1.125\ R_i$ $\left(\text{ie}\ \frac{1.25R_i + R_i}{2} = 1.125R_i\right)$

$T = F\mu\ R_f\ n_p$

$79.577\ \times 10^3 = 0.25 \times 0.157\ R^2_i \times 1.125\ R_i \times 2$

$R_i = 96.58 = 97$ mm say; $D_i = 193$ mm

$R_o = 1.25 \times 97 = 121$ mm; $D_0 = 242$ mm

$F_a = 0.157\ R^2_i = 0.157 \times (97)^2$

$F_a = 1477.213$ N

face width $b = r_o - r_i$

$= 242 - 193$

$= 49$ mm

4. A single plate clutch transmits 25kW at 900 rpm. The maximum pressure intensity between the plates is 85 kPa. The ratio of raii is 1.25. Both the sides of the plates are effective and the coefficient of friction is 0.25.Determine (i) the inner diameter of the plate, and (ii) the axial force to engage the clutch. Assume theory of uniform wear.(Nov-2018)

Given:

Power (P) = 25 KW

Speed (N) = 900 rpm

Co-efficient of friction (μ) = 0.25

Maximum pressure (P_{max}) = 85kPa = 85 x10^3 N/m^2

Ratio of diameter ($\frac{R_0}{R_I}$) =1.25

To Find:

(i) The face width required

(ii) The axial spring force necessary to engage the clutch.

Solution:

(i) Inner diameter of plate

Torque to be transmitted, $T = \frac{60P}{2\pi N} = \frac{60 \times 25000}{2\pi \times 900} = 265.26\ N - m = 265.26\ \times 10^3\ N - mm$

Also we know that,

$T = F\mu\ R_f\ n_p$

$F_a = 2\pi P_{max} R_i(R_0 - R_i)$

$= 2\pi \times 85X10^3 \times R_i(1.25R_i - R_i)$

$= 133517.68\ R^2_i$

$R_f = \frac{R_0 + R_i}{2} = 1.125\ R_i$

$T = F\mu\ R_f\ n_p$

$265.26 = 133517.68\ R^2_i \times 0.25X\ 1.125\ R_i \times 2$

$R_i = 0.1523m = 152.3$ mm say;

Inner diameter (D_i)= 304.6 mm

(ii) Axial force at engage the clutch

$F_a = 133517.68\ R^2_i = 133517.68 \times (0.1523)^2$

$F_a = 3097$ N

5. A multi plate clutch with both sides effective transmits 30 kW at 360 rpm. Inner and outer radii of the clutch discs are 100 and 200 mm respectively. The effective coefficient of friction is 0.25. An axial load of 600 N is applied. Assuming uniform wear conditions find the number of discs required and the maximum intensity of pressure developed.

Given:

Power (P) = 30 kW

N = 360 rpm.

Axial Load (W) = 600 N

$r_o = 200\ mm$

$r_i = 100\ mm$

$\mu = 0.25$

Solution:

Torque to be transmitted, $T = \frac{60P}{2\pi N} = \frac{60 \times 30000}{2\pi \times 360} = 796\ N - m = 796\ \times 10^3\ N - mm$

(a) To find the number of discs required:

Total Frictional torque $T = n\ \mu W r$

Where n= no. of pairs contact surfaces

μ = Coefficient of friction

W= axial load

r = mean radius

$$r = \frac{r_o + r_i}{2} \quad For\ uniform\ wear$$

$$r = \frac{200+100}{2} = 100\,mm$$

$$796 \times 10^3 = n \times 0.25 \times 600 \times 100$$

n = 36 $\therefore$ n = 35

For a multi disc clutch n = $n_1 + n_2 - 1$

Where $n_1 = no. of\ discs\ on\ driving\ shaft$

$n_2 = no. of\ discs\ on\ driven\ shaft$

$$n_1 = \frac{n}{2} = \frac{36}{2} = \mathbf{18};\ n_2 = 18 + 1 = \mathbf{19}$$

(b) Maximum Intensity of pressure (P_{Max})

Maximum Intensity of pressure $P_{max} = \frac{W}{2\pi\, r_o(r_o^2 - r_i^2)}$

$$= \frac{600}{2\pi \times 100(200-100)} = \mathbf{0.0095\ N/mm^2}$$

6. **A multi disk clutch consists of five steel plates and four bronze plates. The inner and outer diameters of friction disks are 75mm and 150mm respectively. The coefficient of friction is 0.1 and the intensity of pressure is limited to 0.3. N/mm2. Assuming the uniform wear theory, calculate (i) the required operating force, and (ii) power transmitting capacity at 750 rpm. (April/May 2008)**

GIVEN:

Outer diameter (D_0) = 150mm,

Inner diameter (D_i) = 75 mm

Intensity of pressure (P_{max}) = 0.3 N/mm^2

Power transmitting capacity (N) = 750 rpm

Number of driving plate m_1 = 5

Number of driven plate m_2 =4

To Find:

(i) The required operating force

(ii) power transmitting capacity at 750 rpm

Solution:

(i) The required operating force

For uniform wear condition

$$P_{max} \times R_i = C$$

$$C = 0.3 \times 37.5 = 11.25\ N/mm$$

Hence required operating force

$$F = W = 2\pi C\ (R_0 - R_i)$$

$$F = 2650.7\ N$$

(ii) Power transmitting capacity at 750 rpm

Nominal torque = $T = F\mu\ R_f\ n_p$

We know that

$R_f = \frac{R_0 + R_i}{2}$ and,

$n_p = m_1 + m_2 - 1 = 5 + 4 - 1$ (PSG 7.89)

$n_p = 8$

$$T = F\mu\ R_f\ n_p$$

$$= 2650.7 \times 0.1 \times \left(\frac{75 + 37.5}{2}\right) \times 8$$

$$= 119.2\ N\text{-}m$$

$$P = \frac{2\pi NT}{60} = 9361.94\ W = 9.36\ KW$$

7. **A single disc clutch having one pair of contacting surface is required to transmit 10kW at 720 rpm under normal operating condition. Due to space limitation the outer diameter should be limited to250mm.The co efficient of friction is 0.25 and the permissible intensity of pressure is 0.5N/mm^2.Use(a) uniform pressure theory and(b) uniform wear theory and determine the clutch dimensions.(May-2018)**

Given:

Power (P) =10kW; Speed (N)=720rpm; Pressure =0.5N/mm^2;Outer diameter (D_o)= 250mm; Co-efficient of friction(μ) =0.25

Solution

Let D_i ,D_o be the inner and outer diameter of the disc respectively, and R_i, R_o the corresponding radii in mm

For uniform wear

$$\therefore R_0 = 125mm$$

Now

Torque to be transmitted, $T = \frac{60\,P}{2\pi N} = \frac{60X10X10^3}{2X\pi X720} = 132.62\text{N.m}$

Assuming ratio of the diameter is 1.5

$\frac{R_0}{R_i}$ =1.5

$$\therefore R_i = 83.33mm$$

We know that

$T = n\,\mu F_0 R_m$

$Where$

$n = Number\ of\ friction\ surfaces$

$n = 2\ and\ \mu = 0.25$

$\therefore\ F_o = Axial\ thrust\ supplied\ by\ springs$

$$= 2\pi P_{max} R_i (R_0 - R_i)$$

$$= 2\pi X500X10^3 X0.083(0.125 - 0.083)$$

$$= 10951\ \ \text{Newtons}$$

For uniform pressure

Intensity of pressure $P = \frac{W}{\pi(r_o^2 - r_i^2)}$

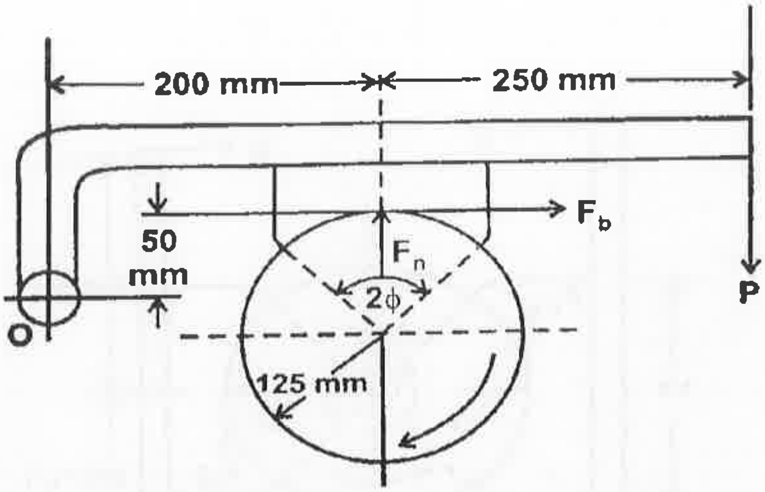
$W = PX\pi\left(r_o^2 - r_i^2\right)$

$= 500X10^3 X\pi(0.125^2 - 0.083^2)$

Axial thrust=13722 Newtons

DESIGN OF SHOE OR BLOCK BRAKE:

8. A single block brakes as shown in figure has the drum diameter 250 mm. the angle of contact is 90^0 and the coefficient of friction between the drum and the lining is 0.35. if the torque transmitted by the brakes is 80 N-m, find the force required to operate the brake.(May-2018)



Given data:

Brake drum diameter, D=250 mm

Angle of contact, $2\varphi = 90^0$

Coefficient of friction $\mu = 0.35$

Torque transmitted, $T_b = 80\ N - m$

Solution:

Equivalent coefficient of friction, $\mu' = \frac{4\mu \sin\varphi}{2\varphi + \sin 2\varphi}$

$$\mu' = \frac{4 \times 0.35 \sin 45}{1.57 + \sin 90} = 0.385$$

Now take moment about the fulcrum O, We get

$(P \times 450) + (F_b \times 50) = F_n \times 200$------------------------------------(1)

Where, $F_b = $ Braking force (i. e tangential force) $= \frac{Torque}{Drum\ radius} = \frac{T_b}{R} = \frac{80000}{125} = 640\ N$

$$Normal\ reaction\ force\ (F_n) = \frac{F_b}{\mu'} = \frac{640}{0.385} = 1662\ N$$

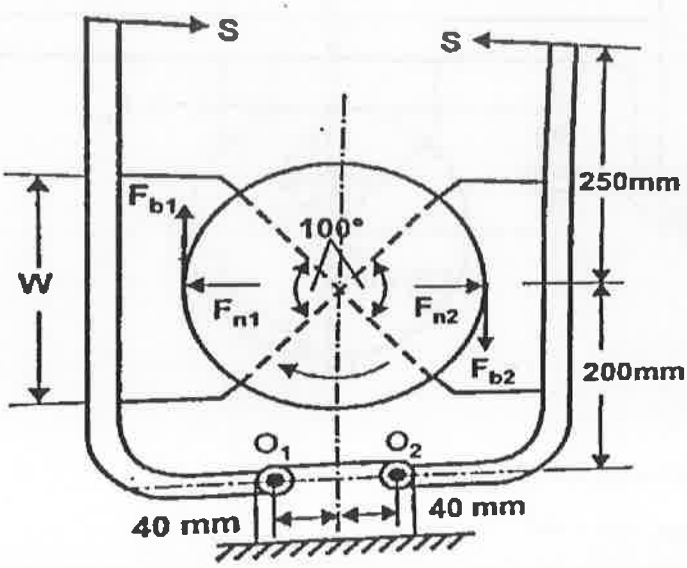
Substitute the above values in equation (1), we get

$$(P \times 450) + (640 \times 50) = 1662 \times 200$$

P = 668 N

The force required to operate the brake is not to be less than 668N. i.e.$P \geq 668\ N$

9. A double shoe brake as shown in figure is capable of absorbing a torque of 1500 N-m. The diameter of the brake drum is 400 mm and the angle of contact for each shoe is 100^0. If the coefficient of friction between the brake drum and lining is 0.4, find (a) the spring force necessary to set the brake and (b) the width of the brake shoe, if the bearing pressure on the lining material is not to exceed 0.3 N/mm^2.



Given data:

Brake drum diameter, D=400 mm

Angle of contact, $2\varphi = 100^0$

Coefficient of friction $\mu = 0.4$

Braking Torque, $T_b = 1500\ N-m$

Bearing pressure $P_b = 0.3\ N/mm^2$

Solution:

(a) the spring force necessary to set the brake:

Let S = spring force necessary to set the brake

$$F_{n1} \& F_{b1} = Normal\ reaction\ and\ braking\ force\ for\ the\ left\ hand\ side$$

$$F_{n2} \& F_{b2} = Normal\ reaction\ and\ braking\ force\ for\ the\ right\ hand\ side$$

Equivalent coefficient of friction, $\mu' = \frac{4\mu \sin\varphi}{2\varphi + \sin 2\varphi}$

$$\mu' = \frac{4 \times 0.4 \sin 50}{1.74 + \sin 100} = 0.45$$

Now taking moment about the fulcrum O_1, we get

$$(S \times 450) + F_{b1}(200 - 40) = F_{n1} \times 200 \qquad (F_{n1} = \frac{F_{b1}}{\mu'})$$

$$(S \times 450) + F_{b1}(200 - 40) = \frac{F_{b1}}{\mu'} \times 200$$

$$F_{b1} = 1.58\, S$$

Similarly taking moment about O_2, we get,

$$(S \times 450) = F_{b2}(200 - 40) + F_{n2} \times 200$$

$$F_{b2} = 0.75\, S$$

For the double shoe brake, the braking torque.

$$T_b = (F_{b1} + F_{b2})\, R$$

$1500 \times 10^3 = (1.58S + 0.75S)\ 200$

S = 3219N

(b) the width of the brake shoe

Let w = width of brake shoe

The bearing pressure for the brake shoe is given by,

$$P_b = \frac{F_n}{2\, R\, w \sin \varphi}$$

Now, $F_{n1} = \frac{F_{b1}}{\mu'} = \frac{(1.58\, S)}{0.45} = \frac{(1.58 \times 3219)}{0.45} = 11302\ N$

$$F_{n2} = \frac{F_{b2}}{\mu'} = \frac{(0.75\, S)}{0.45} = \frac{(0.75 \times 3219)}{0.45} = 5365\ N$$

In this double shoe brake, since the left hand side shoe is facing maximum normal reaction, the width is designed based on this maximum normal reaction.

$$0.3 = \frac{11302}{2 \times 200 \times w \times \sin 50}$$

w = 123 mm

10.A single block brake, the diameter of drum is 250mm and the angle of contact is 90^0. The operating force of 700N is applied at the end of lever which is at 250mm from the centre of the brake block. Determine the torque that may be transmitted. Fulcrum is at 200mm from the centre of brake block with an offset of 50mm from the surface of contact. The coefficient of friction is 0.35 (May/June 2010)

Solution:

(a) Consider clockwise direction:

Taking moments at "O"

$$P \times 450 + F_t \times 50 = R_N \times 200$$

But P = 700N,

$$700 \times 450 + F_t \times 50 = R_N \times 200$$

$$700 \times 450 + (\mu R_N) \times 50 = R_N \times 200$$

$$R_N = 1726 \text{ N}$$

$$F_t = \mu R_N$$

$$F_t = 604.109 \text{ N}$$

Braking Torque, $T_B = F_t \times r$

$$T_B = 604.109 \times \frac{250}{2}$$

$$= 75513.6 \text{ N-mm}$$

$$= 75.513 \text{ N-m}$$

(b) Consider anti-clockwise direction:

Taking moments at "O"

$$P \times 450 = F_t \times 50 + R_N \times 200$$

But P = 700N,

$$700 \times 450 = F_t \times 50 + R_N \times 200$$

$$700 \times 450 = (\mu R_N) \times 50 + R_N \times 200$$

$$R_N = 1448.27 \text{ N}$$

$$F_t = \mu R_N$$

$$F_t = 506.89 \text{ N}$$

Braking Torque, $T_B = F_t \times r$

$$T_B = 506.89 \times \frac{250}{2}$$

$$= 75513.6 \text{ N-mm}$$

$$= 63.36 \text{ N-m}$$

DESIGN OF BAND BRAKE

11. Design a differential band brake for a winch lifting a load of 20 KN through a steel wire rope wound round a barrel of 600 mm diameter. The brake drum, keyed to the barrel shaft, is of 800 mm diameter and the angle of lap of the band over the drum is about 250^0.

Operating arms of the brake are 50 mm and 250 mm. the length of operating lever is 1.6 m.

Given:

Load to be lifted = 20 KN = 20000 N

Barrel diameter $D' = 600$ mm

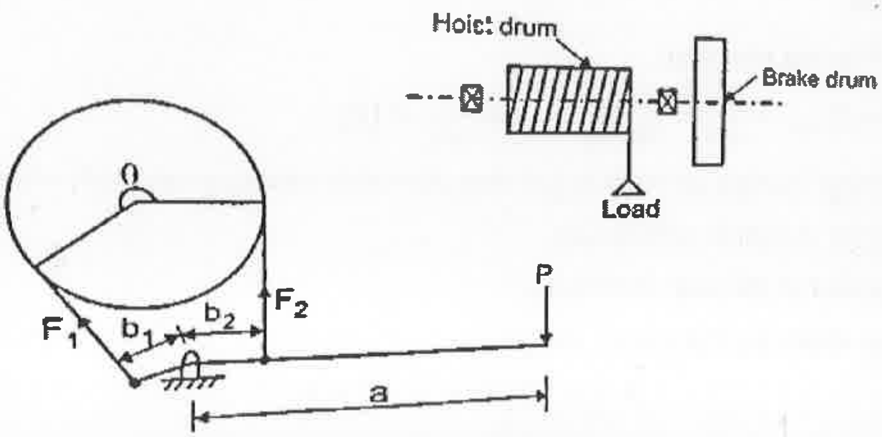
Brake drum diameter, D = 800 mm

Angle of lap, $\theta = 240^0$

Length of arms, $b_1 = 50\ mm;\ b_2 = 250\ mm$

Length of lever, a = 1.6 m = 1600 mm

Solution:



Braking torque, $T_b = load\ to\ be\ lifted\ \times barrel\ radius = 20000 \times 0.3 = \ 2000\ N - m$

This torque is absorbed by the band brake during braking.

Let F_1 and F_2 be the tensions in the tight side and slack side of the band.

$$F_1 - F_2 = F_t \qquad [PSG\ DB\ 7.98]$$

$$F_1 - F_2 = F_t = \frac{Torque}{drum\ radius} = \frac{6000}{0.4} = 15000N$$

$F_1 - F_2 = 15000\ N$ ---------------------------------- (1)

$$\frac{F_1}{F_2} = e^{\mu\theta} \qquad [PSG\ DB\ 7.98]$$

$\frac{F_1}{F_2} = e^{0.3\times(240\times\pi/180)} = 3.514$ ---------------------------------- (2)

Solve the equ (1) & (2), we get

$$F_1 = 20968\ N$$

$$F_2 = 5967\ N$$

Calculation of band width and thickness:

Material for band brake = steel (Allowable tensile stress (σ_t) = 60 N/mm^2)

The Maximum induced tensile stress in the band $\sigma_t = \frac{Tension\ on\ tight\ side}{Area\ of\ cross\ section\ of\ band}$

$$\sigma_t = \frac{F_1}{w\,.\,t}$$

Now take, t = 0.005D = 4 mm

$$69 = \frac{20968}{w\,.4}$$

w = 87.4 mm

Take **w = 90 mm**

Checking for bearing pressure:

Bearing pressure, $P_{max} = \frac{F_1}{w.R} = \frac{20968}{90\times400} = 0.58\frac{N}{mm^2} < [P]$

Since the developed bearing pressure is less than allowable bearing pressure [P] = 1.5 N/mm^2 (PSG DB 7.98) our design is satisfactory.

Force to be applied at the end of the lever:

Taking moments about the fulcrum O, we get

$$P \times a + F_1 \times b_1 = F_2 \times b_2$$

$$P \times 1600 + 20968 \times 50 = 5967 \times 250$$

P = 277 N

Specifications:

Material for band brake = steel

Width of band = 90 mm

Thickness of band = 4 mm

Diameter of barrel = 600 mm

Diameter of brake drum = 800 mm

Force applied at the end of the lever = 277 N

12 .Two way band brake shown in figure. Has the following data

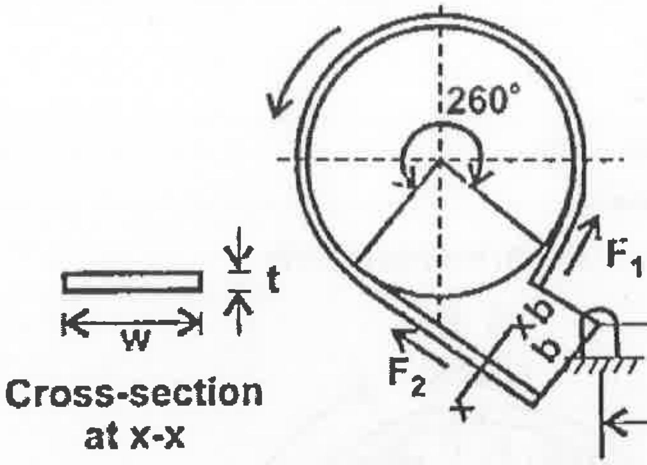
Drum diameter – 480 mm

Coefficient of friction – 0.25

Contact angle - 260^0

Band width – 90 mm

Calculate (i) the torque sustained by force P of 750 N and (ii) thickness of band when allowable stress for band is 70 N/mm². Take moment arm length for tensions as 90 mm.



Solution:

Let F_1, F_2 = Tensions at tight side and slack side respectively

t = thickness of band.

Now, the torque T = $(F_1 - F_2)\,R = (F_1 - F_2)\,200$ -------------------- (1)

$\frac{F_1}{F_2} = e^{\mu\theta} = e^{0.25\times260\times\pi/180} = 3.1$-- (2)

Taking moment about the fulcrum, we get

$$P \times a = F_1 \times b + F_2 \times b = b(F_1 + F_2)$$

$F_1 + F_2 = 6333$ ---------------- (3)

Solve Equ (2) & (3)

$$F_1 = 4788\,N$$

$$F_2 = 1545\,N$$

Substitute the above value in equ (1), we get

$$T = (F_1 - F_2)\,200$$

T = (4788-1545)200

T = 778.32 N-m

Maximum tension, $F_1 = Allowable\ stress\ \times area\ of\ cross\ section\ of\ band$

i.e4788 = $\sigma_t \times$ w.t

$$t = \frac{4788}{70 \times 90} = \mathbf{0.76\ mm}$$

For safety, take thickness of band as 1 mm.

INTERNAL EXPANDING SHOE BRAKE:

13.The internal expanding shoe brake is shown in figure having following particulars.

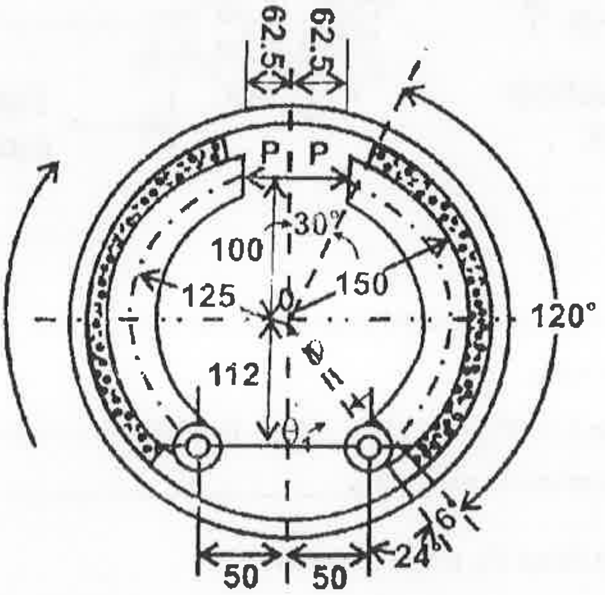
Drum inner diameter – 300 mm

Face width of the shoes – 32 mm

Coefficient of friction for linings – 0.32

Maximum pressure – 1.05 N/mm^2

Determine (a) The actuating force and (b) braking capacity



Solution:

(a) The actuating force:

For the right shoe $P_1 = \frac{M_n - M_f}{C}$ [PSG DB 7.99]

For the left shoe $P_2 = \frac{M_n' + M_f'}{C}$

Where M_n, M_n' - Moment of the normal force on the right and left shoe

M_f, M_f' - Moment of the frictional force on the right and left shoe

C – Moment arm of the actuating force

$$M_f = \frac{\mu P_m br}{\sin\theta_m}\int_{\theta_1}^{\theta_2} \sin\theta\,(r - a\cos\theta)d\theta \text{ [PSG DB 7.99]}$$

$$M_f = \frac{\mu P_m br}{\sin\theta_m}\left[r\,(\cos\theta_1 - \cos\theta_2) + \frac{a}{4}(cos2\theta_2 - cos2\theta_1)\right]$$

$\theta_1 = 6^0 \theta_2 = 126^0$, $P_m = 1.05\ N/mm^2$, r = 150 mm, $\theta_m = 90^0, a = 125\ mm, c = 112 + 100 = 212\ mm$ (Ref figure)

$$M_f = \frac{0.32 \times 1.05 \times 32 \times 125}{\sin 90}\left[150\,(\cos 6 - \cos 126) + \frac{a}{4}(cos252 - cos12)\right]$$

$$\boldsymbol{M_f = 318 \times 10^3\ N - m}$$

$$M_n = \frac{P_m bra}{\sin\theta_m}\int_{\theta_1}^{\theta_2} \sin^2\theta\, d\theta \quad \text{[PSG DB 7.99]}$$

$$M_n = \frac{P_m bra}{2\sin\theta_m}\left[(\theta_2 - \theta_1) - \frac{1}{2}\,(sin2\theta_2 - sin2\theta_1)\right]$$

$$M_n = \frac{1.05 \times 32 \times 150 \times 125}{2\sin 90}\left[(126 - 6)(\pi/180) - \frac{1}{2}\,(sin252 - sin12)\right]$$

$$\boldsymbol{M_n = 842.3 \times 10^3 N - m}$$

$$P_1 = \frac{842.3 \times 10^3 - 318 \times 10^3}{212} = 2475\ N$$

$$P_2 = \frac{842.3 \times 10^3 + 318 \times 10^3}{212} = 5475\ N$$

Even though $P_1 < P_2$, the total actuating force may be taken as $2P_2$ for safety. i.e Total actuating force P = $2P_2 = 2 \times 5475 = 10950\ N$

(b) **Braking capacity:**

Braking torque by the right shoe,

$$M_t = \frac{\mu P_m br^2}{\sin\theta_m}(\cos\theta_1 - \cos\theta_2)$$

$$M_t = \frac{0.32 \times 1.05 \times 32 \times 150^2}{\sin 90}(\cos 6 - \cos 126) = \boldsymbol{383\ N - m}$$

Torque applied on the left side shoe is also same as the torque applied on right side shoe.

Hence, the total torque (i.e., braking capacity) is

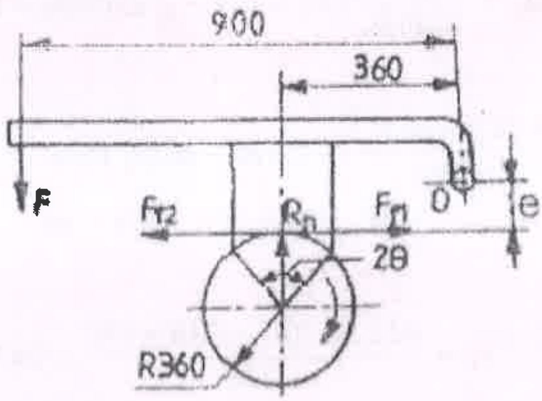
$T = 2M_t = 2 \times 382 = \mathbf{766\ N\text{-}m}$

14. Describe with the help of a neat sketch the principle of an internal expanding shoe. Also deduce the expression for the braking torque.(Nov-2018)

Refer PSG 7.99 Draw the drawing and write the formulas

<u>HOME WORK PROBLEMS:</u>

1. A 360 mm radius Brake drum contacts a single shoe as shown in figure and resists a torque of 250 Nm at 500 rpm. The co-efficient of friction is 0.3. Determine (i) The normal reaction on the shoe, (ii) The force to be applied at the lever end for counter clockwise rotation of the drum if e= 0(iii) The force to be applied at the lever end for clockwise rotation of the drum if e= 40 mm.(iv) The force to be applied at the lever end for counter clockwise rotation of the drum if e = 40 mm. (May/June 2009)



DEPARTMENT OF MECHANICAL ENGINEERING

ME8651– DESIGN OF TRANSMISSION SYSTEMS

PART – A (2 MARKS)

UNIT – I

DESIGN OF TRANSMISSION SYSTEM FOR FLEXIBLE ELEMENTS

1. State the -Law of Belting' (May/June 2007)

2. Explain the term crowning of pulleys. (April/May2008) (Nov/Dec 19)

3. Distinguish between open drive and cross drive of a belt drive. (May-11, Nov-04)

4. What are the types of belts? (Nov 18)

5. What are the materials used for belt? (May/june 2013)

6. Indicate some merits and demerits of belt-drive,

7. What is meant by the ply of belt? (Apr 19)

8. Mention the different types of joints employed for joining flat-belts.

9. What is belt rating?

10. Specify the purpose of crowning of belts.

11. Explain creep in belts.

12. How is a V-belt designated?

13. How is wire-ropes designated?

14. Give the relationship of ratio of tensions in a V-belt drive.(April/May 2008)

15. Define maximum tension in a belt? (April/May 2008)

16. What is silent chain? In what situations, silent chains are preferred? (Nov/Dec 2007)

17. Give any three applications of chain drive. What are their limitations? (Apr -11,18, Nov 19)

18. In what ways the timing belts are superior to ordinary V-belts? (May-15) (Nov 17)

19. Mention the losses in belt drives. (Nov-14)

20. What is meant by “chordal action of chain”? Also name a company that produces driving chains. (May -15)

21. Briefiy explain about friction and its applications (Apr 18)
22.When do you use stepped pulley drive? When do you use fast and loose pulley drive (Nov 18)

UNIT – II

SPUR GEARS AND PARALLEL AXIS HELICAL GEARS

1. Specify the types of gears-failures. (May/june 20013) (Apr 18)

2. How are the following terms defined? ((May/june 2013)

3. What factors influence backlash? (April/May 2008)

4. Write short notes on backlash of gears. (April/May 2008)

5. What is a herringbone gears? (Nov/Dec 2009)

6. What is the advantage of helical gear over spur gear? (May -08) (Apr 18) (Nov 19)

7. What are the common forms of gear tooth profile? (May -10)

8. How does failure pitting happen in gears? (Nov-11, May-04)

9. Differentiate between circular pitch and diametral pitch. (Nov-13, Nov-08, May-11)

10. What are the materials used for gear manufacturing? (May- 11, May-09)

11. Where do we use spiral gears? (Nov-13)

12. Why is gear tooth subjected to dynamic load? (Nov-14, May-09)

13. Laws of gearing (Apr 19)

UNIT – III

BEVEL, WORM AND HELICAL GEARS

1. In which gear-drive, self-locking is available? (Apr 18)

2. When do we use worm-gears? (May/June 2013) (Nov 19)

3. What are the forces acting on the bevel gears? (May / June 2013) (Nov/Dec 2009, 2018)

4. What is the effect of increasing the pressure angle in gears? (Nov-11)

5. What is working depth of a gear-tooth? (May-11)

.6. Name few gear materials. (May-11, May-12)

7. Mention the characteristics of hypoid gear. (May -10)

8. Calculate the angle between the shafts of a crossed helical gears made of two right handed helical gears of 15° helix angle each. (May-09)

9. When is bevel gear preferred? (May-09, May-11) (Apr 18)

10. How can you specify a pair of worm gears? (May -08, May-09)

11. Give the advantage of worm gear drive in weight lifting machines.(May-08) (Nov 17)

12. State the advantages of herringbone gear.(May -15, Nov-08)

13. What is a zerol bevel gear? (May -15, Nov-07)

14. What is virtual number of teeth in bevel gears? (May-14, Nov-14)

15. Define the following terms: a. Cone distance, b. Face angle. (May-14)

16. What is the difference between an angular gear and a miter gear? (Nov-13, 17) (Apr 19)

UNIT IV

DESIGN OF GEAR BOXES

1. What purpose does the housing of gear-box serve? (Apr 18)

2. What is step ratio in a gear box? (May-12, Nov-09)

3. What is step ratio? Name the series in which speeds of multi-speed gear box are arranged. (May-14)

4. What are preferred numbers? (May/June 2013)

5. What are the possible arrangements to achieve 12 speeds from a gear box? (May 2013)

6. Differentiate ray diagram and structural diagram? (Nov/Dec 2009) (Apr 18) (Nov 19)

7. Specify four types of gear boxes. (Nov-14, May-11) (Nov 19)

8. List the ways by which the number of intermediate steps may be arranged in a gear box.

(May-10, May-12)

9. Which type of gear is used in constant mesh gear box? Justify. (Nov-09, May-11)

10. What is speed reducer? (Nov-08, May-10)

11. What are the methods of lubrication in speed reducers? (Nov-11)

12. What is the function of spacers in a gear box? (May-12, Nov-04)

13. Define progression ratio. [Nov 18]

14. What is torque converter? (Apr 19)

UNIT V

DESIGN OF CAM CLUTCHES AND BRAKES

1. State about the profile of cam that gives no jerk and mention how jerk is eliminated. (May-12)

2. Why is it necessary to dissipate the heat generated during clutch operation? (May-13, Nov-11)

3. What is self-locking in a brake? (Nov-11, May-13) (Nov 18)

4. What are the factors upon which the torque capacity of a clutch depends? (May-11, Nov-10)

5. When do we use multiple disk clutches? (May -10)

6. What is the disadvantage of block brake with one short shoe? What is the remedy? (May-11)

7. Under what condition of a clutch, uniform rate of wear assumption is more valid? (May-09)

8. Name four profiles normally used in cams. (May-10)

9. How the "uniform rate of wear" assumption is valid for clutches? (May-08)

10. What are the significances of pressure angle in cam design? (Nov-07)

11. If a multidisc clutch has 6 discs in driving shaft and 7 discs in driven shaft then how many number of contact surfaces it will have? (May -15)

12. Why in automobiles, braking action when traveling in reverse is not as effective as when moving forward? (May -15)

13. Differentiate between uniform pressure and uniform wear theories adopted in the design of Clutches. (Nov-14)

14. Classify clutches based on the coupling methods. (May-14)

15. What is meant by a self-energizing brake? (May-13) (Nov 17)

16. Define pitch point in cam. (Nov-13)

17. Differentiate between clutch and a brake. (Nov-13)

18 .In what ways, the clutches are different from brakes? (Nov/Dec 2011) (Apr 18, 19)

19. Differentiate brakes and dynamometer.[A/M -2017]
20. Distinguish between wet and dry operation of clutches. (Nov 17)
21. Why are cone clutches better than disc clutches. (Apr 19, Nov 19)

PART-B & C

UNIT-I

Flat Belts

1. A leather belt 9mm x 250mm is used to drive a CI pulley 900mm in diameter at 336rpm. If the active arc on the smaller pulley is 120^{o} and stress in tight side is 2Mpa, find the power capacity of the belt. The density of the leather may be taken as 980 kg/m^3 and coefficient of friction of leather on CI is 0.35.

2. Design a FLAT belt drive to transmit 10 KW at 400 rpm. The speed ratio is 3. The distance between the pulley centres is 600 mm. the drive is for a crusher.

3. Design a flat belt drive to transmit 10KW @1000rpm. The centre distance is 2m and the speed ratio is 3.

4. It is required to design a leather crossed belt drive to connect 7.5 KW, 1440rpm electric motor to a compressor running at 480rpm. The distance between the centers of the pulley is twice the diameter of the larger pulley. The belt should operate at 20m/s and its thickness is 5mm. Density of leather is $950kg/m^3$ and permissible stress is 5.6MPa.

V- Belts

1. A V- belt drive consists of three V- belts in parallel on grooved pulleys of the same size. The angle of groove is 30° and the coefficient of friction 0.12. The cross sectional area of each belt is 800 mm^2 and the permissible safe stress in the belt material is 3MPa. Calculate the power that can be transmitted between two pulleys 400mm in diameter rotating at 960rpm.

2. (i) Select a suitable V-belt drive to connect a 7.5Kw, 1440 rpm induction motor to run a fan at a approximately 480 rpm for a service of hr per day. The space available for center distance is 1m. (ii) Enlist the merits and demerits of V -belt over flat belt.

3. A V-belt drive is to transmit 40KW in a heavy duty saw mill which works in two shifts of 8hours each. The speed of motor shaft is 1440 rpm with the approximate speed reduction of 3 in the machine shaft. Design the drive and calculate the average stress induced in the belt.

4. Design a V-belt drive and calculate the actual belt tension and average stress for the following data. Driven pulley diameter, D= 500 mm, driver pulley diameter, d=150 mm, center distance C=925 mm, speed n_1 = 1000 rpm, n_2 = 300 rpm and power, P = 7.5 kW.

5. Design a V-belt drive to transmit 50KW at 1440 rpm from an electric motor to a textile machine running 24 hours a day. The speed of the machine shaft is 480 rpm.

6. A V-belt drive is to transmit 50KW in a heavy duty saw mill which works in two shifts of 8hours each. The speed of motor shaft is 1440 rpm with the approximate speed reduction of 2 in the machine shaft. The peripheral speed of the belt should not exceed 24m/s. Design the drive and calculate the average stress induced in the belt.

7. Two shafts whose centers are 1 meter apart are connected by a V – belt drive. The driving pulley is supplied with 95 kW power and has an effective diameter of 300mm. It runs at 1000 rpm, while the driven pulley runs at 375 rpm. The angle of groove on the pulleys is 40^{o}. Permissible tension in 400 mm^2 cross– sectional area belt is 2.1MPa. The material of the belt has density of 1100 kg/ mm^3. The driven pulley is overhung, the distance of the centre from the nearest bearing being 200 mm. The coefficient of friction between belt and pulley rim is 0.28. Estimate the number of belts required.

Chain Drives

1. The reduction of speed from 360 rpm to 120 rpm is desired by the use of chain drive. The driving sprocket has 10 teeth. Find the number of teeth on the driven sprocket. If the radius of driven sprocket is 250mm and the center to center distance between the two sprockets is 400mm, find the pitch and length of the chain.

2. Design a CHAIN drive to connect at 15 KW, 1440 rpm electric motor to a transmission shaft running at 350 rpm. The operation involves, moderate shocks

3. A roller chain drive is used between a driver shaft running at 1440rpm and a driven shaft running approximately at 720rpm. The power transmitted is 15KW. The drive is to be used for 2 shifts/day with 8hours/shift. The centre distance is approximately 1000mm and the chain tension can be adjusted by moving the motor in the rails. Design the drive.

4. Design a chain drive to run a compressor from an 11 KW electric motor running at 970 rpm, the compressor speed being 330 rpm. The compressor operates 16 hr per day. The center distance should be approximately 500mm. The chain tension can be adjusted by shifting the motor on slides.

5. Design a chain drive to actuate a compressor from a 12 kW electric motor at 900 rpm, the compressor runs at 250 rpm. Minimum centre distance should be 500 mm; the chain tension may be adjusted by shifting the motor on rails. The compressor is to work 8 hour/day.

6. A blower is to run at 600 rpm. Power to the blower is available from a motor rated 8kW at 1500 rpm. Design a chain drive for the system if the centre distance is to be 800mm.

7. Design a chain drive to actuate a compressor from 15kW electric motor running at 1,000 rpm, the compressor speed being 350 rpm. The minimum centre distance is 500 mm. The compressor operates 15 hours per day. The chain tension may be adjusted by shifting the motor.

Rope Drives

1. A crane is lifting a load of 18 KN through a wire rope and a hook. The weight of the hook etc. is 10kN. The load is to be lifted with an acceleration of 1m/s^2.Calculate the diameter of the wire rope. The rope diameter may be taken as 30 times the diameter of the rope. Take a factor of safety of 6 and Young's modulus for the wire rope 0.8 x 10^5 N/mm^2. The ultimate stress may be

taken as 1800 N/mm^2. The cross-sectional area of the wire rope may be taken as 0.38 times the square of the wire rope diameter.

2. A crane is used to lift a load of 32KN through wire rope. Weight of crane hook is 6KN. The load is to be lifted with an acceleration of 1.2 m/s^2. Design the drive.

UNIT-II
PART-B
Spur Gear drive

1. Design a pair of straight SPUR gears to transmit 15 KW at 1440 rpm. Speed reduction is 3. State clearly all assumptions made. Check for compressive and bending stresses. Also check for plastic deformation of teeth. Tabulate the results neatly.

2. Design a spur gear pair to transmit 5KW at 1440 rpm from an electric motor to an air compressor running at 720rpm. Take working life as 10,000 hrs.

3. Design a straight spur gear drive to transmit 8KW. The pinion speed is 720rpm and the speed ratio is 2. Both the gears are made of the same surface hardened carbon steel with 55RC and core hardness less than 350BHN. Ultimate strength is 720 N/mm^2 and yield strength is 360 N/mm^2.

4. A 27.5 kW power is transmitted at 450 rpm to a shaft running at approximately 112 rpm through a spur gear drive. The load is steady and continuous. Design the gear drive and check the design. Assume the following materials: Pinion-heat treated cast steel; Gear-High grade cast iron. [AUC: Dec 2010]

5. A motor shaft rotating at 1440 rpm has to transmit 15 KW power to a low speed shaft at 500 rpm. A 20° pressure angle involute tooth gear pinion is used. The pinion has 25 teeth. Both gear and pinion are made of cost iron having allowable strength of 55 N/ mm^2. Design a suitable gear drive.

6. Design a spur gear which is required to transmit 10KW power. The speed of the driving motor and the driven machine are 400 rpm and 200 rpm, respectively. The approximate center distance may be taken as 600 rpm. The teeth have 20° full depth involute profile. Assuming that the gear is made of cost iron FG200 , Having allowable strength 75 N/mm^2 and 180 BHN
Core hardness.

7. A motor shaft rotating at 1500 rpm has to transmit 15 kW to a low speed shaft with a speed reduction of 3:1. Assume starting torque to be 25% higher than the running torque. The teeth are 20 degree involute with 25 teeth on pinion. Both the pinion and gear are made of C45 steel. Design a spur gear drive to suit the above conditions and check for compressive and bending stresses and plastic deformation. Also sketch the spur gear drive.

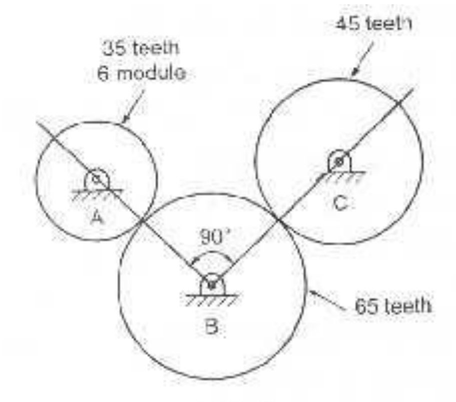
8. Design a straight spur gear drive to transmit 8 kW. The pinion speed is 720 rpm and the speed ratio is 2. Both the gears are made of the same surface hardened carbon steel with 55RC and core hardness less than 350 BHN. Ultimate strength is720 N/mm^2 and yield strength is 360 N/ mm^2.

9. Design a spur gear pair to transmit 1.5KW at 1440 rpm from an electric motor to an air compressor running at 720rpm. Take working life as 10,000 hrs.

10. An electric motor is to be connected to a reciprocating pump through a gear pair. The gears are overhanging in their shafts. Motor speed = 1440 rpm. Speed reduction ratio = 5. Motor power = 36.8 kW. The gears are to have 20° pressure angles. Design a spur gear drive.

11. Design a gear drive to transmit 22 kW @ 1000rpm. Speed reduction is 2.5. The centre distance between the shafts is 350mm. The materials are: pinion-C45, gear wheel: CI grade 30. Design the drive using Lewis and Buckingham equations.

12. Referring figure: 1, spur gear A receives 3KW @ 600rpm through its shaft and rotates clockwise. Gear B is an idler and C is the driven gear. The teeth are 20 degree full depth. Determine: (i) the torque each shaft must transmit (ii) The tooth load for which each gear must be designed (iii) The force applied to the idler shaft
as a result of the gear tooth load.



Helical Gear drive

1. Design a pair of helical gears to transmit 25KW at a speed reduction ratio of 5:1. The input shaft runs at 2000rpm.

2. A helical gear with 30 degree helix angle has to transmit 35kW at 1500 rpm with a speed reduction ratio 2:5. If the pinion has 24 teeth determine the necessary module, pitch diameter and face width for 20 degree full depth teeth. Assume 15Ni 2Cr 1 Mo15 material for both pinion and wheel.

3. Design a HELICAL gear drive to transmit 5 KW at 1440 rpm. Desired speed ratio is 2.5. Take helix angle as 15°. Use C45 steel for the gears. Check for strength of materials under different modes of failure. Make a clear sketch showing important values of parameters. [AUT-Dec2010]

4. A pair of helical gear subjected to moderate shock loading is to transmit 20KW at 1500 rpm of the pinion. The speed reduction ratio is 4 and the helix angle is 20°.
The service is continuous and the teeth are 20°full depth in the normal plane. For the gear life of 10,000 hours, design the gear drive.

5. Design a pair of helical gears to transmit 30KW power at a speed reduction ratio of 4:1. The input shaft rotates at 2000 rpm. Take helix angle and normal pressure angles equal 25° and 20° respectively. Both pinion and gear are made of steel .The number of teeth on the pinion may be taken as 30.
Name of the part Permissible stress BHN
Pinion 55MPa 340
Gear 40MPa 300

6. A pair of helical gears with 23° helix angel is to transmit 2.5 KW at 1000 rpm of the pinion. The velocity ratio is 4: 1. The pinion is to be forged steel and the driven gear is to be cast steel. The gears are of 20° full depth involute form and the pinion is to have 24 teeth. Deign the gear drive
.
7. A pair of helical gears is used to transmit 5 KW at 720 rpm of the pinion. Gears are of made of C45 steel. The speed reduction ratio is 2. Number of teeth on pinion is 20. Normal pressure angle is 20°. Normal module is 5mm. helix angle is 30°. Design the gear drive. (Use Lewis and Buckingham's equation)

8. A helical gear with 30° helix angle has to transmit 35kW at 1500 rpm. With a speed reduction ratio 2.5. If the pinion has 24 teeth, determine the necessary module, pitch diameter and face width for 20° full depths the teeth. Assume 15Ni 2Cr 1 Mo 15 material for both pinion and wheel.

9. Design a pair of helical gears to transmit 30KW at a speed reduction ratio of 4:1. The input shaft runs at 2000rpm. Both pinion and gear are 15Ni2Cr1 Mo 15 under carburized condition.

UNIT-III

PART-B
Bevel Gear Drive

1. Design the teeth of a pair of bevel gears to transmit 18.75 kW at 600 rpm of the pinion. The velocity ratio should be about 3 and the pinion should have about 20 teeth which are full depth 20_o involutes. Find the module, face width, diameter of the gears and pitch cone angle for both gears. [AUC-2017]

2. Design a pair of bevel gears for two shafts whose axes are at right angles to transmit 20KW @ 1000 rpm. The speed of gear is 250rpm.

3. Design a BEVEL gear drive to transmit 4 KW. Speed ratio = 4. Driving shaft speed 225 rpm. The drive is non-reversible. Assume a life of 25000 hours.

4. A Pair of bevel gears is to be used to transmit 14KW from a pinion rotating at 400rpm to a gear mounted on shaft running at 200rpm. The axes of the two shafts are at 90°. Design the pair of bevel gears. .

5. Design a pair of bevel gears for two shafts whose axes are at right angles to transmit 10KW @ 1440 rpm. The speed of gear is 720rpm. Use Lewis & Buckingham's equation.

6. Design a pair of bevel gears to transmit 10 KW at a pinion speed of 1440 rpm. Required transmission ratio is 4. Materials for gears is 15 Ni 2Cr 1 Mo 15.

7. Design a cast iron bevel gear drive for a pillar drilling machine to transmit 1.5KW at 800 rpm to a spindle at 400 rpm. The gear is to work for 40 hours per week for 3 years. Pressure angle is 20°. Check the design and calculate the basic dimensions.

8. A pair of straight tooth bevel gears has a velocity ratio of 4/3. The pitch diameter of the pinion is 150 mm. The face width is 50mm. The pinion rotates at 240 rev/min. The teeth are 5mm module, 14 10 involutes. If 6 kW is transmitted, determine (i) the tangential force at the Mean radius (ii) the pinion thrust force (iii) the gear thrust force. Draw the free body diagrams indicating the forces.

9. Design a straight bevel gear drive between two shafts at right angles to each other. Speed of the pinion shaft is 360 rpm and speed of the gear wheel shaft is 120 rpm. Pinion is of steel and wheel of cast iron. Each gear is expected to work 2 hours/ day for 10 years. The drive transmits 9.37 KW.

10. A 1 kW motor running at 1200 rpm drives a compressor at 780 rpm through a 90_o bevel gearing arrangement. The pinion has 30 teeth. The pressure angle of the teeth is 20_o. Both the pinion and gear are made of heat treated cast iron grade 35. Determine the cone distance, average module and face width of the gears.

Worm and Wheel Drive

11. Design worm gear drive to transmit 50KW @ 1440 rpm. Velocity ratio is 24.

12. A hardened steel WORM rotates at 1440 rpm and transmits 12 KW to a phosphor bronze gear with gear ratio of 16. Design the worm gear drive and determine the power loss by heat generation.

13. Design a worm gear drive to transmit 15 KW form a worm at 1440 rpm to the worm wheel. The speed of the worm wheel should be 40±2% rpm.

14. A hardened steel worm rotates at 1440 rpm and transmits 12KW to a phosphor bronze gear. The speed of the worm wheel should be 60±3%rpm. Design the worm gear drive if an efficiency of at least 82% is desired

15. Design a worm gear drive to transmit a power of 22.5 KW. The worm speed is 1440 rpm and the speed of the wheel is 60 rpm. The drive should have a minimum efficiency of 80% and above. Select suitable materials for the worm and the wheel and decide upon the dimensions of the drive.

16. A 2 kW power is applied to a worm shaft at 720 rpm. The worm is of quadruple start with 50mm as pitch circle diameter. The worm is of quadruple start type with 50mm as pitch circle diameter. The worm gear has 40 teeth with 5mm module. The pressure angle in the diametral plane is 20°. Determine (i) the lead angle of the worm, (ii) velocity ratio, and (ii) centre distance. Also, calculate efficiency of the worm gear drive, and power lost in friction.

17. Design a worm gear drive to transmit 22.5 kW at a worm speed of 1440 rpm. Velocity ratio is 24:1. An efficiency of at least 85% is desired. The temperature rise should be restricted to 40_{o} C. Determine the required cooling area

UNIT-IV

PART-B

1. A six speed gear box is required to provide output speeds in the range of 125 to 400 rpm with a step ratio of 1.25 and transmit a power of 5 kW at 710 rpm. Draw the speed diagram and kinematics diagram. Determine the number of teeth module and face width of all gears, assuming suitable materials for the gears.

2. Design a 9 speed gear box for the following data. Minimum speed: 100rpm, step ratio: 1.25. The input is from a 4KW, 1440rpm motor. Draw the speed diagram, kinematic diagram and indicate the number of teeth on each gear.

3. Design a nine – speed gear box for a machine to provide speeds ranging from 100 to 1500 rpm. The input is from a motor of 5 kW at 1440 rpm. Assume any alloy steel for the gear.

4. A machine tool gear box is to have 9 speeds. The gear box is driven by an electric motor whose shaft rotational speed is 1400 rpm. The gear box is connected to the motor by a belt drive. The maximum and minimum speeds required at the gear box output are 1000 rpm and 200 rpm respectively. Suitable speed reduction can also be provided in the belt drive. What is the step ratio and what are the values of 9 speeds? Sketch the arrangement. Obtain the number of teeth on each gear and also the actual output speeds.

5. Select speeds for a 12 speed gear box for a minimum speed of 16 rpm and a maximum speed of 900rpm. Draw the speed diagram, kinematic diagram and indicate the number of teeth on each gear.

6. Design the layout of a 12 speed gear box for a milling machine having an output of speeds ranging from 180 to 2000 rpm. Power is applied to the gear box from a 6 kW induction motor at 1440 rpm. Choose standard step ratio and construct the speed diagram. Decide upon the various reduction ratios and number of teeth on each gear wheel sketch the arrangement of the gear box. [AUC-May2008]

7. Design the headstock gear box of a lathe having nine spindle speeds ranging from 25 to 1000 rpm. The power of the machine may be taken as 6 kW and speed of the motor is 1450 rpm. Minimum number of teeth on the gear is to be 25. a) Draw the speed diagram b) Sketch the layout of the gear box. c) Calculate the number of teeth on all gears.

8. A gear box is to be designed for the following specifications:
Power to be transmitted = 5.5KW
Number of speeds = 9
Minimum speed = 280 rpm
Maximum speed = 1800 rpm
Input motor speed = 1440 rpm
Draw the kinematic layout diagram and the speed diagram. Determine the number of teeth on all gears.

9. Draw the ray diagram and kinematic lay out of a gear box for an all geared headstock of a lathe. The maximum and minimum speeds are to be 600 and 23 rpm respectively. The number of steps is 12 and drive is from a 3 kW electric motor running at 1440rpm.

10. Select speeds for a 12 speed GEAR BOX for a minimum speed of 112 rpm and maximum speed of 1400 rpm. Drive speed is 1400 rpm. Draw speed diagram and a kinematic arrangement of the gear box showing the number of teeth in all the gears.

11. The spindle of a pillar drill is to run at 12 different speeds in the range of 100rpm and 135 rpm. Design a three stage gear box with a standard step ratio. The gear box receives 5KW from an electric motor running at 360rpm. Sketch the layout of the gear box, indicating the number of teeth on each gear. Also sketch the speed diagram.

12. Design a 16 speed gear box for the following data. Minimum speed: 100rpm, step ratio: 1.25. The input is from a 5KW, 1000rpm motor. Draw the speed diagram, kinematic diagram and indicate the number of teeth on each gear

13. In a milling machine, 18 different speeds in the range of 35 rpm and 650 rpm are required. Design a three stage gear box with a standard step ratio. Sketch the layout of the gear box, indicating the number of teeth on each gear. The gear box receives 3.6 kW from an electric motor running at 1440 rpm. Sketch also the speed diagram.

14. For a load lifting arrangement transmitting 7.5 KW with electric motor running at 1440 rpm, constant mesh type SPEED REDUCER is required with reduction ratio 16. Design a suitable arrangement and make a neat sketch.

UNIT-V
PART-B
Plate Clutches

1. A single plate sketch, effective on both sides, is required to transmit 25 KW at 3000 rpm. Determine the outer and inner diameter of frictional surfaces if the coefficient of friction is 0.25, ratio of diameter is 1.25 and the maximum pressure is not to exceed 0.1 N/mm2. Determine (i) the face width required and (ii) the axial spring force necessary to engage the clutch.

2. A dry single plate clutch is to be designed for an automotive vehicle whose engine is rated to give 100KW at 2400 rpm and maximum torque 500 N-m. The outer radius of the friction plate is

25% more than the inner radius. The intensity of pressure between the plates is not to exceed 0.07 N/mm^2. The coefficient of friction may be assumed to be equal to 0.3. The helical springs are required by this clutch to provide axial force necessary to engage the clutch are 8. If each spring has a stiffness of 40N/mm, determine the dimensions of the friction plate and initial compression in the springs.

3. A plate clutch with maximum diameter 60mm has maximum lining pressure of 0.35 MPa. The power to be transmitted at 400 rpm is 135 KW and μ= 0.3. Find inside diameter and spring force required to engage the clutch. Springs with spring index 6 and material spring steel with safe shear stress 600 MPa are used. Find the diameters if 6 springs are used.

4. A single plate clutch, both side being effective is required to connect a machine shaft to a driver shaft which runs at 500rpm .The moment of inertia of the rotating parts of the machine is $1Kgm^2$.The inner and the outer radii of the friction discs are 50mm&100mm respectively .Assuming uniform pressure of $0.1N/mm^2$ and μ =0.25, determine the time taken for the machine to reach full speed when the clutch is suddenly engaged. Also determine the power transmitted by the clutch, the energy dissipated during the clutch slip and the energy supplied to the machine during engagement.

Multi-plate Clutch

1. A multi disk clutch consists of five steel plates and four bronze plates. The inner and outer diameters of friction disks are 75mm and 150mm respectively. The coefficient of friction is 0.1 and the intensity of pressure is limited to 0.3. N/mm^2. Assuming the uniform wear theory, calculate (i) the required operating force, and (ii) power transmitting capacity at 750 rpm.

2. A plate clutch has 3 discs on the driving shaft and 2 discs on the driven shaft, providing 4 pairs of contact surfaces. The OD of contact surface is 240mm and ID is 120mm. Assuming uniform pressure and μ=0.3, find the total spring load for pressing the plates together to transmit 25KW @ 1575 rpm. If there are 6 springs each of stiffness 13KN/m and each of contact surfaces have worn away by 1.25mm, find the power that can be transmitted, assuming uniform wear.

3. A multi disc wet clutch is to be designed for a machine tool driven by an electric motor of 12.5 KW running at 1440 rpm. Space restrictions limit the outside disc diameter to 100mm. Determine the appropriate value of inside diameter, total number of discs and clamping force.

4. A hydraulically operated clutch is to be designed for an automatic lathe. Determine the number of plates and operating force required for the clutch to transmit 35 Nm. The clutch is to be designed to slip under 300% of rated torsional moment to protect the gears and other part of the drive. The limits for the diameter of friction surfaces due to space limitation are 100mm and 62.5mm. This clutch is to operate in an oily atmosphere.

Cone Clutches

1. An engine developing 45kW at 1000 rpm id fitted with a cone clutch built inside the fly wheel. The cone has a face angle of 12.5 degree and a maximum mean diameter of 500 mm. The coefficient of friction is 0.2. The normal pressure on the clutch face is not exceeded $0.1N/mm^2$. Determine (i) The face width required (ii) the axial spring force necessary to engage the clutch.

2. A cone clutch has a cone angle of 11.5^o, a mean frictional diameter of 320 mm, and face width of 60 mm. The clutch is to transmit a torque of 200 Nm. The coefficient of friction is 0.26. Find the activating force and pressure using the assumption of uniform pressure.

3. A power of 20 KW is to be transmitted through a cone clutch at 500rpm. For uniform wear condition, find the main dimensions of the clutch and shaft. Also determine the axial force required to engage the clutch. Assume the coefficient of friction as 0.25, the maximum normal pressure on the friction surface is not to exceed 0.08 MPa and take the design stress for the shaft material as 40MPa.

4. A cone clutch is to transmit 7.5KW at 900rpm. The cone has a face angle of 12^o. The width of the face is half of the mean radius and the normal pressure between the contact faces is not to exceed 0.9 N/mm^2. Assuming uniform wear and the coefficient of friction between contact faces as 0.2, find the main dimensions of the clutch and the axial force required to engage the clutch.

5. A leather faced conical clutch has cone angle of 30 degree. The pressure between the contact surfaces is limited to $0.35N/mm^2$ and breadth of the conical surface is not to exceed 1/3 of the mean radius. Find the dimensions of the contact surfaces to transmit 22KW at 2000rpm. Also calculate the force required to engage the clutch. Take coefficient of friction as 0.15.

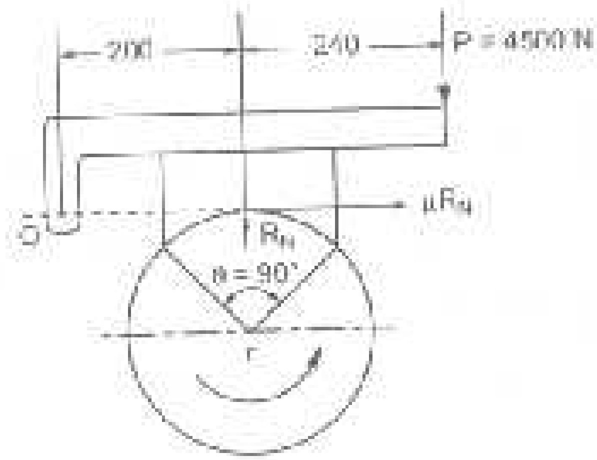
6. A power of 20 KW is to be transmitted through a cone clutch at 500 rpm. For uniform wear condition find the main dimensions of the clutch and shaft. Also determine the axial force required to engage the clutch. Assume the coefficient of friction as 0.25, the maximum normal pressure on the friction surface is not to exceed 0.08 MPa and take the design stress for the shaft material as 40 MPa.

Brakes

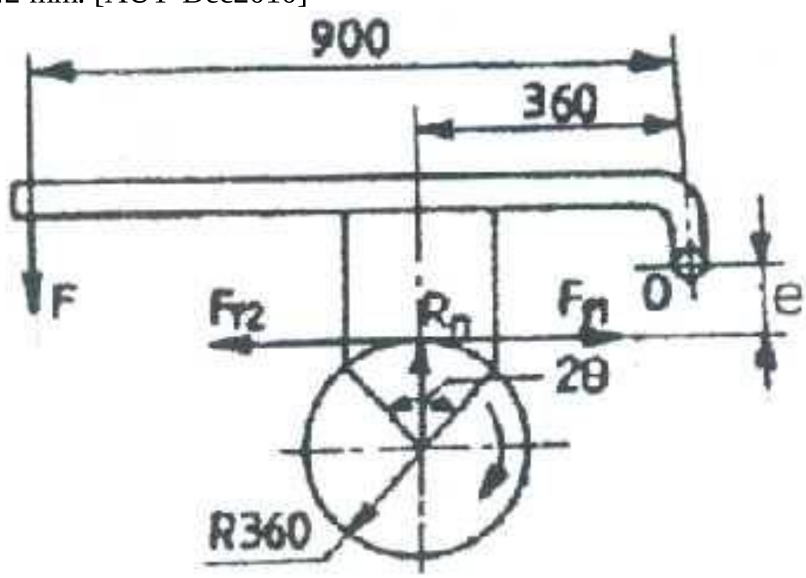
Single block brake

1. A single block brake, the diameter of drum is 250mm and the angle of contact is 90^o. The operating force of 700N is applied at the end of lever which is at 250mm from the centre of the brake block. Determine the torque that may be transmitted.Fulcrum is at 200mm from the centre of brake block with an offset of 50mm from the surface of contact. The coefficient of friction is 0.35

2. Calculate the average bearing pressure and the initial and average braking powers for the block shoe shown in fig.B1. The diameter of the drum is 400 mm and it rotates at 200 rpm. Coefficient of friction is 0.2 and drum width is 75 mm.



3. A 360 mm radius Brake drum contacts a single shoe as shown in figure (B2) and resists a torque of 250 Nm at 500 rpm. The co-efficient of friction is 0.3.
determine
(i) The normal reaction on the shoe,
(ii) The force to be applied at the lever end for counter clockwise rotation of the drum if e= 0
(iii) The force to be applied at the lever end for clockwise rotation of the drum if e= 42 mm.
(iv) The force to be applied at the lever end for counter clockwise rotation of the drum if e = 42 mm. [AUT-Dec2010]



Double block brake

4. A rope drum of an elevator having 650mm diameter is fitted with a brake drum of 1m diameter. The brake drum is provided with 4 cast iron brake shoes each subtending an angle of 45^{o} .The mass of elevator when loaded is 200Kg and moves with a speed of 2.5m/s. The brake has sufficient capacity to stop the elevator in 2.75m. Assuming the coefficient of friction as 0.2, find (i) Width of shoe if the allowable pressure on the brake shoe is limited to 0.3 N/mm^2. (ii) Heat generated in stopping the elevator.

5. The double shoe brake shown in figure: B3 must provide a braking torque of 280.5Nm @ 720rpm. Assuming coefficient of friction 0.3 for brake lining and pV=1100 KPa m/s, determine
(a) Spring force P required to set the brake
(b) Width of shoes (c) Which shoe will have greater rate of wear?

6. The layout of a double block brake is shown in figure (B4). The brake is rated at 250N-m @ 650rpm. The drum diameter is 250mm.assuming the co-efficient of friction as 0.3 and for conditions of service a pV value of 1000(Kpa) m/s may be assumed. Determine (i) The spring force "S" required to set the brake (ii) Width of shoes (iii) Which shoe will have greater rate of wear?

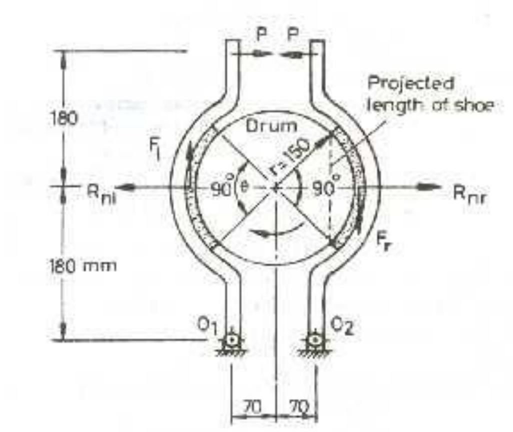


Figure: B3

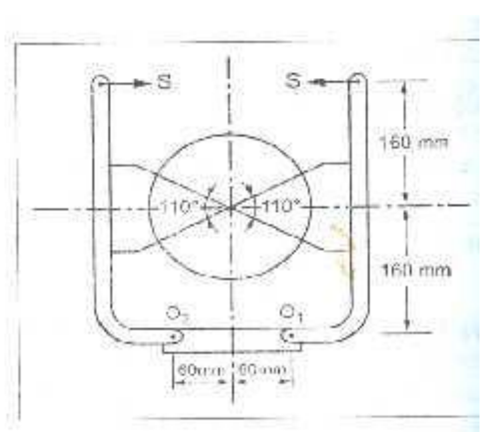


Figure: B4

Internal Expanding Brake

1. An internal expanding shoe brake has the following dimensions:
Diameter of the drum = 300 mm, distance between the fulcrum centers = 80 mm , distance of fulcrum centers and that of cam axis, both from the drum centre= 100 mm, distance of the line of action of braking force from the cam axis = 90 mm, distance between the points where the cam acts on the two brake shoes = 30 mm. Each shoe subtends an angle of 90° at the drum centre. If the braking force is 750 N and the coefficient of friction is 0.3, find the braking torque on the drum. Assume the reaction between the brake shoes and the drum passes through the point bisects the contact angle. Also assume that forces exerted by the cam ends on the two shoes are equal.

2. The figure: B4 show the arrangement of two brake shoes which act on the internal surface of the brake drum. The width of brake lining is 35mm. The intensity of pressure at any point is 0.4 $\sin\theta$ N/mm^2. The coefficient of friction is 0.4. Determine the braking torque and the magnitude of F_1 & F_2.

Band Brake

1. In a band brake the drum diameter is 800 mm and the band thickness is 5 mm. The brake facing has a coefficient of friction of 0.25. The arc of contact is 250^o. This brake drum is attached to a hoisting drum that sustains a rope load of 8 kN. The operating force has a moment arm of 1.5 m and the band is attached 150 mm from the pivot point. Determine a) the force required to just support the load b) the required force when the direction is reversed, and c) the width of steel band, limiting its tensile strength to 50 N/mm^2.

2. Design a differential band brake for a winch lifting a load of 20KN through a steel rope wound round a barrel of 600mm diameter. The brake drum, keyed to the barrel shaft, is of 800mm diameter and the angle of lap of the band over the drum is about 240^o. Operating arm of the brake are 50mm and 250mm. Length of operating lever is 1.6m. Also calculate the effort applied. [AUC-May 2006]

3. Design a differential band brake for a winch lifting a load of 20KN through a steel rope wound round a barrel of 600mm diameter. The brake drum, keyed to the barrel shaft, is of 800mm diameter and the angle of lap of the band over the drum is about 240^o. Operating arm of the brake are 50mm and 250mm. Length of operating lever is 1.5m. Also calculate the effort applied.

4. A differential band brake is to be designed for a winch lifting a load of 45KN through a rope wound round a barrel of 500mm diameter. The brake drum, (to be keyed to the barrel shaft, is to be 600mm diameter and the angle of lap of the band over the drum is 250^o. Determine the width and thickness of band. Operating arm of the brake are 40mm and 200mm. Length of operating lever is 1.5m. Also calculate the effort applied.

Band & Block Brake

5. Determine the maximum braking torque for a band and block brake having 12 blocks, each of which subtends an angle of 16_o at the centre. The brake is applied to a rotating drum of diameter 600mm. The blocks are 75mm thick. The two ends of bands are attached to a pin on the opposite sides of the brake fulcrum at distances 40mm and 150mm from the fulcrum. A force of 250N is applies at a distance of 900mm from the fulcrum.

Design of Cam

6. Draw the displace time , velocity time and the acceleration time curves for the follower in order to satisfy the following conditions (1) Stroke of the follower 25mm (2) Outstroke takes place with SHM during 90_0 of cam rotation (3) Return stroke takes with SHM during 75_0 of cam rotation (4) Cam rotates with a uniform speed of 800 rpm.

7. A cam is to give the following motion to a knife-edged follower :
Outstroke during 60_o of cam rotation;
Dwell for the next 30_o of cam rotation;
Return stroke during next 60_o of cam rotation and
Dwell for the remaining 210_o of cam rotation. The stroke of the follower is 40mm and the minimum radius of the cam is 50mm. the follower moves with uniform velocity during both the outstroke and return strokes. Draw the profile of the cam when the axis of the follower passes through the axis of the cam shaft.

8. A radial cam rotates at 1200 rpm with the follower rising 20mm with SHM in 1500 of the cam rotation .The roller is 32mm in diameter and the prime circle is 80mm in diameter. Check whether undercutting will occur.

9. A cycloidal cam with a central roller follower has a rise of 25mm in cam angle of 70_o. Base circle radius is 90mm and the follower roller radius is 20mm. Speed of cam is 5000rpm. Mass of follower is 0.5Kg. Find the maximum value of acceleration of the follower, corresponding pressure angle, stiffness of the spring used with the follower and maximum cam force. The friction between the follower and the guide may be ignored.

10. Design a cam for operating the exhaust valve of an oil engine. It is required to give equal uniform acceleration and retardation during opening and closing of the valve, each of which corresponding to 60° of cam rotation. The valve should remain in the fully open position for 20° of cam rotation. The lift of the valve is 37.5 mm and the least radius of the cam is 50 mm, the follower is provided with a roller of 50 mm diameter and its line of stroke passes through the axis of the cam.

Reg. No. :

Question Paper Code : 42854

B.E./B.Tech. DEGREE EXAMINATION, APRIL/MAY 2018

Sixth Semester

Mechanical Engineering

ME 2352 – DESIGN OF TRANSMISSION SYSTEMS

(Common to Mechanical and Automation Engineering)

(Regulations 2008)

(Also common to PTME 2352 – Design of Transmission Systems for B.E. (Part-Time) Fifth Semester – Mechanical Engineering – Regulations 2009)

Time : Three Hours Maximum : 100 Marks

Usage of Approved design data book is permitted.

Answer ALL questions.

PART – A (10×2=20 Marks)

1. Mention the important characteristics of Worm gear.
2. Differentiate crown gears and mitter gears.
3. Classify brakes based on the direction of application of braking force.
4. What is a wet clutch ? State its advantage.
5. Why thin flat belt is preferred than thick narrow belts ?
6. State any two advantages of V belt drive over the flat belt drive.
7. Enlist any two main advantages and disadvantages of helical gears.
8. What are the modes of gear failure ?
9. What is ray diagram ?
10. State the main difference between sliding mesh and constant mesh gear box.

PART – B (5×16=80 Marks)

11. a) Design a gear box to provide 12 output speeds ranging from 25 to 600 rpm. The input speed of 2.25 kW motor is 1440 rpm. Assume standard step ratio and minimum number of teeth in any stage as 20. List the speeds of all shafts.

(OR)

b) Sketch the speed diagram and the kinematic layout for an 18 speed gear box for the following data : Motor speed = 1440 rpm, Maximum and minimum output speed = 800 and 16 rpm respectively. Arrangement = 2×3×3.

12. a) Design a Fort duck flat belt drive to transmit 20kW at 720 rpm to an aluminium rolling machine, the speed ratio being 3. The distance between the pulleys is 3 m. Diameter of rolling machine pulley is 1.2 m.

(OR)

b) A V-belt having a lap of 180° has a cross section area of 2.5 cm^2 and groove angle as 45°. The density of a belt is 0.0015 kg/cm^3 and maximum stress is limited to 400×10^4 N/m^2. If $\mu = 0.15$, find the power that can be transmitted, if the wheel has a mean diameter of 300 mm and runs at 1000 rpm.

13. a) Design a spur gear drive required to transmit 45 kW at a pinion speed of 800 rpm. The velocity ratio is 3.5:1. The teeth are 20° full depth involute with 18 teeth on the pinion. Both the pinion and gear are made of steel with a maximum safe static stress of 180 N/mm^2 and hardness 400 BHN. Assume carefully cut wheels, medium shock conditions (K_0=1.25) with arbitrary initial velocity as 12m/s.

(OR)

b) A compressor running at 360 rpm is driven by a 140 kW, 1440 rpm motor through a pair of 20° full depth carefully cut precision helical gears having helix angle of 25 °. The centre distance is approximately 400 mm. The motor pinion is to be forged steel (40 $Ni_2Cr_1Mo_{28}$) and the driven gear is to be cast steel Grade 1 (CS 65). Assume medium shock conditions, minimum number of teeth in any stage as 20 and hardness 400 BHN. Design the gear pair.

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

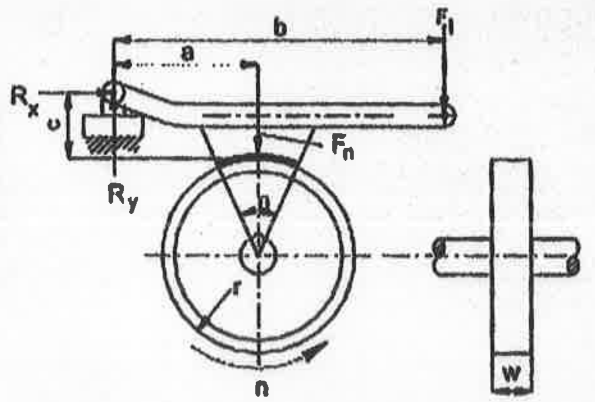
(OR)

b) 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 $14\frac{1}{2}°$ 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 4. Efficiency of the drive if the coefficient of friction is 0.05. Determine whether or not the drive is self-locking.

15. a) A multi plate disc clutch transmits 55 kW of power at 1800 rpm. Coefficient of friction for the friction surfaces is 0.1. Axial intensity at pressure is not to exceed 160 kN/m^2. The internal radius is 80 mm and is 0.7 times the external radius. Find the number of plates needed to transmit the required torque.

(OR)

b) A short shoe block brake shown in fig has a coefficient of friction 0.3, has to absorb a frictional power of 14.924 kW at 650 rpm. What is the actuating force required ? Can the brake be self-locking ?



b = 1m
r = 0.375m
a = 0.375m
c = 0.05m

Reg. No. :

Question Paper Code : 41409

B.E./B.Tech. DEGREE EXAMINATION, APRIL/MAY 2018

Sixth/Seventh Semester

Mechanical Engineering

ME6601 – DESIGN OF TRANSMISSION SYSTEMS

(Common to Mechanical Engineering (Sandwich) and Mechanical and Automation Engineering)

(Regulations 2013)

Time : Three Hours Maximum : 100 Marks

Use of PSG design data book is permitted in the examination.

Assumptions and assumed data have to be stated clearly.

Answer ALL questions

PART – A (10×2=20 Marks)

1. Define Co-efficient of friction.
2. What are the advantages of chain drives ?
3. Specify the types of gear-failures.
4. In what ways helical gears are different from spur gears ?
5. When do we use bevel gears ?
6. In which gear-drive, self-locking is available ?
7. For what purpose we are using gear-box ?
8. What is a speed diagram ?
9. What are the types of brakes used in modern vehicles ?
10. How does the function of a brake differ from that of a clutch ?

PART – B (5×13=65 Marks)

11. a) Select a flat belt to drive a mill at 250 rpm from a 10 kW, 730 rpm motor. Centre distance is to be around 2000 mm. The mill shaft pulley is of 1000 mm diameter.

(OR)

b) 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 centre distance should be 500 mm. Motor is mounted on an auxiliary bed. Compressor is to work for 8 hours/day.

12. a) Design a pair of spur gears to transmit 20 kW at a pinion speed of 1400 rpm. The transmission ratio is 4. Assume 15 Ni2Cr1Mo15 for pinion and C45 for gear.

(OR)

b) Design a helical gear drive to transmit the power of 15 kW. Speed ratio 6, pinion speed 1200 rpm, helix angle is 25°. Select 15 Ni2Cr1Mo15 for pinion and C45 for gear and design the gear pair.

13. a) Design a bevel gear drive to transmit 7 kW at 1600 rpm for the following data.

Gear ratio	= 3
Material for pinion and gear	= C45 steel
Life	= 10,000 hours

(OR)

b) The input to worm gear shaft is 18 kW and 600 rpm. Speed ratio is 20. The worm is to be of hardened steel and the wheel is made of chilled phospher bronze. Considering wear and strength, design worm and worm wheel.

14. a) Design the layout of a 12 speed gear box for a lathe. The minimum and maximum speeds are 100 and 1200 rpm. Power is 5 kW from 1440 rpm Induction motor. Construct the speed diagram using a standard speed ratio. Calculate the number of teeth in each gear wheel and sketch the arrangement of the gear box.

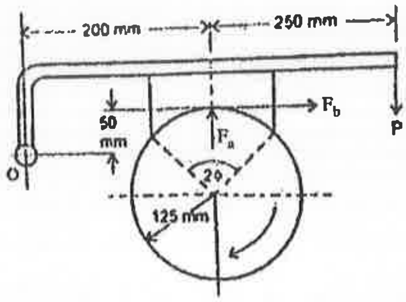
(OR)

b) Design a gear box to give 18 speeds for a spindle of a milling machine. The drive is from an electric motor of 4 kW at 1000 rpm. Maximum and minimum speeds of the spindle are to be around 650 rpm and 35 rpm respectively.

15. a) A single disk clutch having one pair of contacting surface is required to transmit 10 kW at 720 rpm under normal operating condition. Due to space limitation the outer diameter should be limited to 250 mm. The coefficient of friction is 0.25 and the permissible intensity of pressure is 0.5 N/mm^2. Use (a) uniform pressure theory and (b) uniform wear theory and determine the clutch dimensions.

(OR)

b) A single block brake as shown in fig. has the drum diameter 250 mm. The angle of contact is 90° and the coefficient of friction between the drum and the lining is 0.35. If the torque transmitted by the brake is 80,000 N-mm, find the force required to operate the brake.



PART – C **(1×15=15 Marks)**

16\. a) Select a V-belt drive for 15 kW, 1440 rpm motor, which drives a centrifugal pump running at a speed of 576 rpm for a service of 8-10 hours per day. The distance between the driver and the driven shaft is approximately 1.2 m. Service factor, $K_s = 1.1$, design factor $N_a = 1.0$, $V_R = 2.5$.

(OR)

b) A temporary elevator is assembled at the construction site to raise building materials, such as cement, to a height of 20 m. It is estimated that the maximum weight of the material to be raised is 5 kN. It is observed that the acceleration in such applications is $1 m/s^2$, 10 mm diameter, 6 × 19 construction wire ropes with fibre core are used for this application. The tensile designation of the wire is 1570 and the factor of safety should be 10 for preliminary calculations. Determine the number of wire ropes required for this application. Neglect bending stresses.

Reg. No. : | | | | | | | | | | | | |

Question Paper Code : 20817

B.E./B.Tech. DEGREE EXAMINATION, NOVEMBER/DECEMBER 2018.

Sixth/Seventh Semester

Mechanical Engineering

ME 6601 – DESIGN OF TRANSMISSION SYSTEM

(Common to: Mechanical Engineering (Sandwich)/Mechanical and Automation Engineering)

(Regulations 2013)

(Also Common to: PTME 6601 – Design of Transmission System for B.E. (Part – Time) Fifth Semester – Mechanical Engineering – Regulations – 2014)

Time : Three hours

Maximum : 100 marks

(Use of approved design data book is permitted.)

Answer ALL questions.

PART A — (10 × 2 = 20 marks)

1. Name the four types of belts used for transmission of power.

2. When do you use stepped pulley drive?

3. State the advantages of toothed gears over the other types of transmission systems.

4. Why pinion is made harder than gear?

5. List the forces acting on bevel gears.

6. What is irreversibility in worm gear?

7. Define progression ratio.

8. List out the all possible arrangements to achieve 16 speed gear box

9. Name few commonly used friction materials.

10. What do you meant by self-locking brake?

PART B — (5 × 13 = 65 marks)

11. (a) A nine speed gear box used as a headstock gear box of a turret lathe is to provide a speed range of 180 rpm to 1800 rpm. Using standard step ratio, draw the speed diagram, and the kinematic lay out. Also find and fix the number of teeth on all gears.

Or

(b) Sketch the speed diagram and the kinematic layout for an 18 speed gear box for the following data: Motor speed = 1440 rpm, minimum output speed = 16 rpm, maximum output speed = 800 rpm, arrangement 2×3×3. List the speeds of all the shafts when the output speed is 16 rpm.

12. (a) Select a High speed duck flat belt drive for a fan running at 360 rpm which is driven by 10 kW, 1440 rpm motor. The belt drive is open type and space available for a center distance of 2 m approximately. The diameter of the driven pulley is 1000 rpm.

Or

(b) A Centrifugal pump running at 340 rpm is to be driven by a 100 kW motor running at 1440 rpm. The light duty drive is to work for atleast 20 hours every day. The center distance between the motor shaft and the pump shaft is 1200 mm. Suggest a suitable multiple V belt drive for this application.

13. (a) A Compressor running at 300 rpm is driven by a 15 kW, 1200 rpm motor through a 14½° full depth spur gears. The center distance is 375 mm. The motor pinion is to be of C30 forged steel hardened (BHN 250) and tempered, and the driven gear is to be of cast iron. Assuming medium shock condition and minimum number of teeth as 18, design the gear drive completely.

Or

(b) Design a carefully cut helical gear to transmit 15 kW at 1400 rpm to the following specifications. Speed reduction is 3. Pressure angle is 20°, Helix angle is 15°. The material for both the gears is C45 steel. Allowable static stress is 180 N/mm^2, surface endurance limit 800 N/mm^2. Young's modulus of the material = 2×10^5 N/mm^2. Assume minimum number of teeth as 20 and medium shock conditions. v = 15 m/s.

14. (a) Design a pair of bevel gears to transmit 10kW at a pinion speed of 1440 rpm. Required transmission ratio is 4. Material of gears is 15 Ni 2Cr 1 Mo 15 steel (BHN = 400). The tooth profiles of the gears are of 20° composite form. Assume minimum number of teeth as 20,v = 5m/s and medium shock Conditions.

Or

(b) A hardened steel worm rotates at 1440 rpm and transmits 12 kW to a phosper bronze gear. The speed of the worm wheel Should be 60 ±3% rpm. Design the worm gear drive if an efficiency of atleast 82% is desired. Assume q = 1, medium shock conditions v = 5 m/s, pressure angle = 20°.

15. (a) A single plate clutch transmits 25 kW at 900 rpm. The maximum pressure intensity between the plates is 85 kN/m². The ratio of radii is 1.25. Both sides of the plates are effective and the co efficient of friction is 0.25. Determine (i) The inner diameter of the plate and (ii) the axial force to engage the clutch. Assume theory of uniform wear.

Or

(b) Determine the capacity and the main dimensions of a double block brake for the following data: The brake sheave is mounted on the cast iron drum shaft. The hoist with its load weighs 45 kN and moves downwards with a velocity of 1.15 m/s. The pitch diameter of the hoist drum is 1.25 m. The hoist must be stopped within a distance of 3.25 m. The kinetic energy of the drum may be neglected. Assume sintered metal block shoe, equal friction force on each shoe, Continuous service and poor heat condition.

PART C — (1 × 15 = 15 marks)

16. (a) The transporter of a heat treatment furnace is driven by a 4.5 kW, 1440 rpm induction motor through a chain drive with a speed reduction ratio of 2.4. The transmission is horizontal with bath type of lubrication. Rating is Continuous with 3 shifts per day. Design the complete chain drive. Assume center distance as 500mm and service factor as 1.5.

Or

(b) A workshop crane is lifting a load of 25 kN through a wire rope and hook. The weight of the hook etc., is 15 kN. The rope drum diameter may be taken as 30 times the diameter of the rope. The load is to be lifted with an acceleration of 1m/s². Calculate the diameter of the wire rope. Take a factor of safety of 6 and E for the wire is 80 kN/mm². The ultimate stress may be taken as 1800 MPa. The cross sectional area of the wire rope may be taken as 0.38 times the square of the wire rope diameter.

Reg. No. : | | | | | | | | | | | |

Question Paper Code : 53313

B.E./B.Tech. DEGREE EXAMINATIONS, APRIL/MAY 2019.

Sixth/Seventh Semester

Mechanical Engineering

ME 6601 — DESIGN OF TRANSMISSION SYSTEMS

(Common to Mechanical Engineering (Sandwich)/
Mechanical and Automation Engineering)

(Regulation 2013)

(Also common to PTME 6601 — Design of Transmission System for B.E. Part Time – Fifth Semester – Mechanical Engineering – Regulation 2014)

Time : Three hours | Maximum : 100 marks

(Usage of approved design data book is permitted)

Answer ALL questions.

PART A — (10 × 2 = 20 marks)

1. Which side of the belt should be on the bottom side of the pulley and why?

2. What are the various stresses induced in wire ropes?

3. State the law of gearing.

4. What is meant by virtual number of teeth?

5. What is crown and miter gear?

6. Define pitch and lead of worm gears.

7. What is a torque converter?

8. Draw the kinematic layout for the 6-speed gearbox.

9. How does the function of a brake differ from that of a clutch?

10. Why are cone clutches better than disc clutches?

PART B — (5 × 13 = 65 marks)

11. (a) A motor driven blower is to run at 650 rpm driven by an electric motor of 7.5 kW at 1800 rpm. Design a suitable V-belt drive.

Or

(b) 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 0.5m. Motor is mounted on an auxiliary bed. Compressor is to work for 8 hours/day.

12. (a) Design a spur gear drive to transmit 22 kW at 900 rpm, speed reduction is 2.5. Materials for pinion and wheel are C15 steel and Cast-Iron grade 30 respectively. Take pressure angle of 20° and working life of the gears as 10,000 hours.

Or

(b) A pair of helical gears is to be designed to transmit 30 kW at a pinion speed of 1500 rpm. The velocity ratio is 3. Selecting 15 Ni2Cr1 Mo15 steel as the material, determine the dimensions of the gears.

13. (a) Design a bevel gear drive to transmit 7 kW at 1600 rpm for the following data.

Gear ratio = 3

Material for pinion and gear = C45 steel

Life = 10,000 hours.

Or

(b) A hardened steel worm rotates at 1440 rpm and transmits 12 kW to a phosphor bronze gear. The speed of the worm gear should be 60rpm. Design the worm gear drive if an efficiency of atleast 82% is desired by using AGMA method.

14. (a) Design a gearbox with 12 speeds from a source of motor with a speed of 1600 rpm. The required range is from 160 rpm to 2000 rpm.

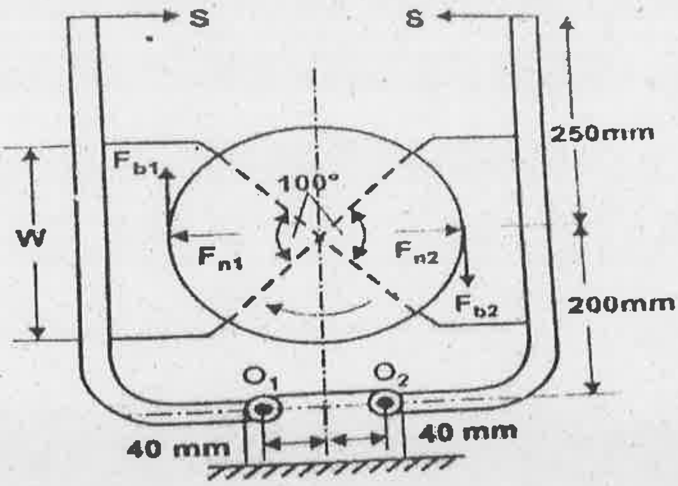
Or

(b) Design a gearbox with 9 speed output from a single speed input. The required speed range is from 180 rpm to 1800 rpm.

15. (a) A multiple disc wet clutch is to be designed for a machine tool driven by an electric motor of 12.5 kW running at 1440 rpm. The frequency of clutch engagement is 6/hr and the machine tool is to operate continuously 8hrs/day. Determine the appropriate values for disc inside diameter, outside diameter, total number of discs and clamping force.

Or

(b) A double shoe brake as shown in the figure is capable of absorbing a torque of 1500 N-m. The diameter of the brake drum is 400 mm and the angle of contact for each shoe is 100°. If the coefficient of friction between the brake drum and lining is 0.4, find (i) the spring force necessary to set the brake and (ii) the width of the brake shoe, if the bearing pressure on the lining material is not to exceed 0.3 N/mm².



PART C — (1 × 15 = 15 marks)

16. (a) Design an 18-speed gearbox from a source of 1000 rpm. Maximum and minimum speeds are to be around 650 rpm and 35 rpm respectively.

Or

(b) Select a suitable wire rope for a mini hoist carrying a load of 2 tonnes to be lifted from a depth of 100m. A rope speed of 10m/s must be attained in 10 seconds. Assume minimum factor of safety as 10.

Reg. No. :

Question Paper Code : 91848

B.E./B.Tech. DEGREE EXAMINATIONS, NOVEMBER/DECEMBER 2019

Sixth/Seventh Semester

Mechanical Engineering

ME 6601 – DESIGN OF TRANSMISSION SYSTEMS

(Common to Mechanical Engineering (Sandwich)/Mechanical and Automation Engineering)

(Regulations 2013)

(Also Common to PTME 6601 – Design of Transmission Systems for B.E. (Part-Time) – Fifth Semester – Mechanical Engineering – Regulations 2014)

Time : Three Hours Maximum : 100 Marks

Use of Design Data book is permitted.

Answer ALL questions.

PART – A (10×2=20 Marks)

1. What do you mean by crowning of pulleys ?
2. State the advantages of chain drives.
3. What are the advantages of toothed gears over the other types of transmission systems ?
4. Why do you prefer helical gears than spur gears ?
5. Differentiate a straight bevel gear and a spiral bevel gear.
6. For transmitting large power, worm reductions gears are not generally preferred. Why ?
7. What does the ray diagram of gear box indicate ?
8. Specify four types of gear boxes.
9. Define base circle and pitch circle with respect to cam.
10. Why are cone clutches better than disc clutches ?

PART – B (5×13=65 Marks)

11. a) Design a V-belt drive to run a centrifugal pump at 340 rpm is to be driven by a 100 kW motor running at 1440 rpm. The drive is to work for atleast 20 hours per day. The centre distance between the motor shaft and the pump shaft is 1200 mm. (13)

(OR)

b) Select a wire rope for a vertical mine hoist to lift a load of 20 kN from a depth of 500 metres. A rope speed of 3 m/s is to be attained in 10 seconds. (13)

12. a) Design a spur gear drive to transmit a power of 8 kW. Pinion speed is 764 rpm. Speed ratio is 2. The gears are to be made of C45. Life is to be 10,000 hours. (13)

(OR)

b) A pair of helical gears subjected to moderate shock loading is to transmit 37.5 kW at 1750 rpm of the pinion. The speed reduction ratio is 4.25 and the helix angle is 15°. The service is continuous and the teeth are 20° full depth in the normal plane. Design the gears, assuming a life of 10,000 hours. (13)

13. a) Design a bevel gear drive to transmit 7 kW at 1600 rpm for the following data. Gear ratio = 3, Material for pinion and gear = C45 Steel, Life = 10,000 hours. (13)

(OR)

b) The input to worm gear shaft is 18 kW and 600 rpm. Speed of worm wheel is 30 rpm. The worm is to be hardened steel and the wheel is made of chilled phosphor bronze. Considering wear and strength, design worm and worm wheel. (13)

14. a) Design the layout of a 12 speed gear box for a lathe. The minimum and maximum speeds are around to be 30 rpm and 1400 rpm respectively. Construct the speed diagram using a standard speed ratio and sketch the arrangement of the gear box. (13)

(OR)

b) Design a 16 speed gear box, the minimum speed is 100 rpm and maximum speed is 560 rpm. Construct the speed diagram using a standard speed ratio and sketch the arrangement of the gear box. (13)

15. a) A single plate clutch, effective on both sides, is required to transmit 25 kW at 1500 rpm. Determine the inner and outer diameter of friction surface if the coefficient of friction is 0.25, ratio of diameter is 1.5 and the maximum pressure is not to exceed 2 N/mm^2. Also, determine the axial thrust to be provided by springs. Assume the theory of uniform wear and max. pressure theory. (13)

(OR)

b) A double shoe brake as shown in fig 15 (b) is capable of absorbing a torque of 1500 N-m. The diameter of the brake drum is 400 mm and the angle of contact for each shoe is 100°. If the coefficient of friction between the brake drum and lining is 0.4, find (i) the spring force necessary to set the brake and (ii) the width of the brake shoe, if the bearing pressure on the lining material is not to exceed 0.3 N/mm^2. **(13)**

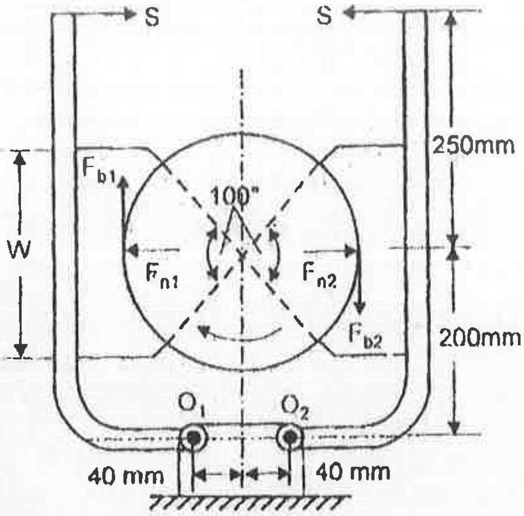


Fig 15 (b)

PART – C **(1×15=15 Marks)**

16. a) i) State the condition and approaches to be followed to avoid under cutting in cam. **(8)**

ii) Discuss the force analysis and derive equation to determine the torque transmitted by it. **(7)**

(OR)

b) i) 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 centre distance should be 0.5 m. Motor is mounted on an auxiliary bed. Compressor is to work for 8 hours per day. **(7)**

ii) Variable speed drive is to be design to vary speed from 300 rpm to 600 rpm. Suggest a variable speed drive system and explain its kinematic arrangement. **(8)**

Reg. No. :

Question Paper Code : 90866

B.E./B.Tech. DEGREE EXAMINATIONS, NOVEMBER/DECEMBER 2022.

Sixth Semester

Mechanical Engineering

ME 8651 – DESIGN OF TRANSMISSION SYSTEMS

(Common to: Mechanical Engineering (Sandwich) Mechanical and Automation Engineering)

(Regulations 2017)

Time : Three hours

Maximum : 100 marks

Answer ALL questions.

PART A — ($10 \times 2 = 20$ marks)

1. Why are belt drives called 'flexible' drives?

2. What are the advantages of chain drives?

3. State two reasons for adopting involute curve for Gear tooth Profile.

4. Draw a double helical gear.

5. List the types of bevel gears.

6. Write two drawbacks of worm gears.

7. Enumerate the types of gearbox.

8. How does a hydraulic fluid coupling work?

9. Name four profiles used in cams.

10. Differentiate the clutch and the brake.

PART B — (5 × 13 = 65 marks)

11. (a) A 6 × 19 wire rope with fiber core and ultimate tensile strength of 1570 N/mm^2 is used to raise the load of 20 kN as shown in figure 1. The nominal diameter of the wire rope is 12 mm and the sheave has 500 mm pitch diameter. Determine the expected life of the rope assuming 500 bends per week.

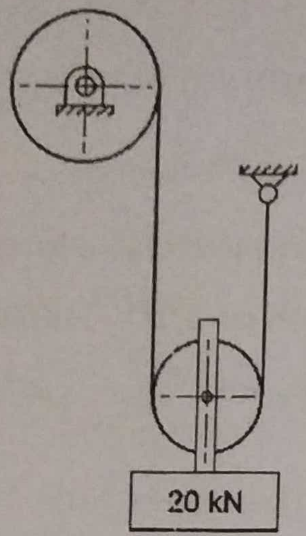


Figure 1

Or

(b) Design a chain drive to connect 5kW, 1400 rpm electric motor to a drilling machine. The speed reduction is 3:1. The centre distance should be approximately 500 mm. (i) Select a proper roller chain for the drive. (ii) Determine the number of chain links. (iii) Specify the correct centre distance between the axes of sprockets.

12. (a) A pair of spur gears with 20° full-depth involute teeth consists of a 20 teeth pinion meshing with a 41 teeth gear. The module is 3 mm while the face width is 40 mm. The material for pinion as well as gear is steel with an ultimate tensile strength of 600 N/mm^2. The gears are heat treated to a surface hardness of 400 BHN. The pinion rotates at 1450 rpm and the service factor for the application is 1.75. Assume that velocity factor accounts for the dynamic load and the factor of safety is 1.5. Determine the rated power that the gears can transmit.

Or

(b) A pair of parallel helical gears consists of a 20 teeth pinion meshing with a 100 teeth gear. The pinion rotates at 720 rpm. 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 the gear is made of steel 4OC8(Sut = 600 N/mm^2) and heat treated to a surface hardness of 300 BHN. The service factor and the 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 gears.

13. (a) A pair of bevel gears transmitting 7.5 kW at 300 rpm is shown in figure 2. The pressure angle is 20°. Determine the components of the resultant gear tooth force and draw a free-body diagram of forces acting on the pinion and the gear.

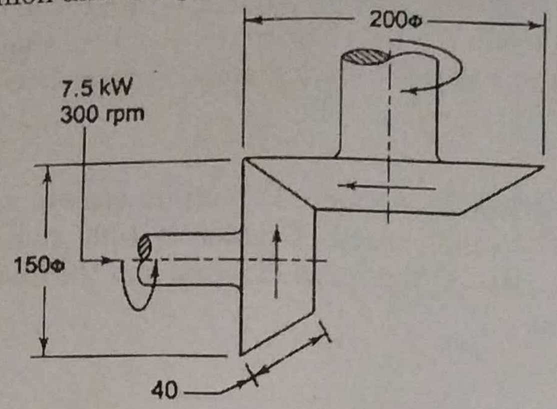


Figure 2

Or

(b) 1 kW power at 720 rpm is supplied to the worm shaft. The number of starts for threads of the worm is four with a 50 mm pitch – circle diameter. The worm wheel has 30 teeth with 5 mm module. The normal pressure angle is 20°. Calculate the efficiency of the worm gear drive and the power lost in friction.

14. (a) Design a sliding mesh nine speed gear box for a machine tool with speed ranging from 36 rpm to 550 rpm. Draw the speed diagram and kinematic arrangement showing number of teeth in all gears.

Or

(b) Explain about torque converters in automotive industries.

15. (a) A plate clutch consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 100 and 200 mm respectively. The coefficient of friction is 0.2 and the permissible intensity of pressure is 1 N/mm². Assuming uniform-wear theory, calculate the power-transmitting capacity of the clutch at 750 rpm.

Or

(b) A four-wheeled automobile car has a total mass of 1000 kg. The moment of inertia of each wheel about a transverse axis through its centre of gravity is 0.5 kg-m². The rolling radius of the wheel is 0.35 m. The rotating and reciprocating parts of the engine and the transmission system are equivalent to a moment of inertia of 2.5 kg-m², which rotates at five times the road-wheel speed. The car is traveling at a speed of 100 km/h on a plane road. When the brakes are applied, the car decelerates at 0.5 g. There are brakes on all four wheels. Calculate the energy absorbed by each brake and the torque capacity of each brake.

PART C — ($1 \times 15 = 15$ marks)

16. (a) Design V-belt drive to connect a 15kW, 2880 rpm normal torque AC. motor to a centrifugal pump, running at approximately 2400 rpm, for a service of 18 hours per day. The centre distance should be approximately 400 mm. Assume that the pitch diameter of the driving pulley is 125 mm.

Or

(b) Design a gearbox to provide 12 output speeds ranging from 160 to 2000 rpm. The input speed of motor is 1600 rpm. Choose a standard speed ratio: construct the speed diagram and the kinematic arrangement.

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Reg. No. : | | | | | | | | | | | | |

Question Paper Code : 91848

B.E./B.Tech. DEGREE EXAMINATIONS, NOVEMBER/DECEMBER 2019
Sixth/Seventh Semester
Mechanical Engineering
ME 6601 – DESIGN OF TRANSMISSION SYSTEMS
(Common to Mechanical Engineering (Sandwich)/Mechanical and Automation Engineering)
(Regulations 2013)
(Also Common to PTME 6601 – Design of Transmission Systems for B.E. (Part-Time) – Fifth Semester – Mechanical Engineering – Regulations 2014)

Time : Three Hours   Maximum : 100 Marks

Use of Design Data book is permitted.
Answer ALL questions.

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1. What do you mean by crowning of pulleys ?

2. State the advantages of chain drives.

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(OR)

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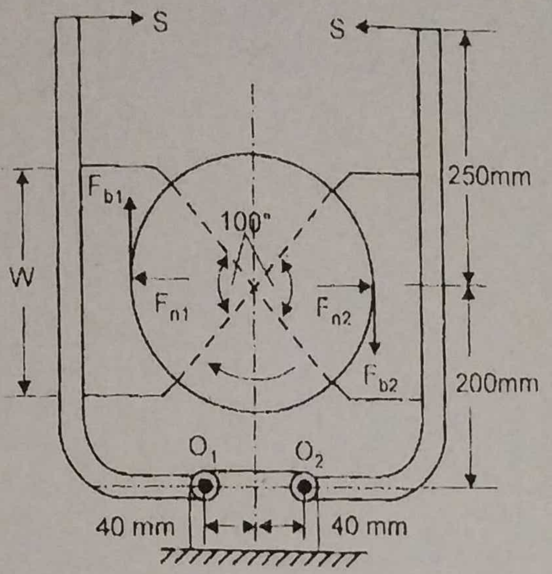

Fig 15 (b)

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